Active Lateral Secondary Suspension in a High-Speed Train to Improve Ride Comfort

by

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Licentiate thesis

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Preface

This thesis summarises the work carried out during my licentiate studies. The project is part of the Swedish research and development programme Gröna Tåget (Green Train), financed by Banverket (Swedish Rail Administration). The work has mainly been carried out at the Royal Institute of Technology (KTH) in Stockholm, in close co-operation with Bombardier Transportation, Västerås.

The original vehicle model built in SIMPACK was received from Bombardier Transportation in Västerås. Björn Roos provided good support when I modified the model according to my needs. General questions regarding the simulation tool have been answered by Homan Seyedin and Christoph Weidemann at Intec in Munich, Germany.

The actuator model in Simulink was provided by Liebherr in Lindenberg, Germany, where Philipp Kegel was very helpful when I integrated 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 received from Interfleet Technology. Paul Sundvall has made valuable comments and suggestions regarding the simulations with $H_\infty$ control.

I am grateful for the support and assistance from my supervisors Sebastian Stichel and Rickard Persson. Thanks also to all the colleagues at the Division of Rail Vehicles for a nice working environment.

Last, but not least, I would like to thank my dear Lena for her moral support as well as linguistic advice on this thesis.

Stockholm, April 2009

Anneli Orvnäs
The Swedish Licentiate degree may need an explanation for readers outside of Sweden. An intermediate academic degree called Licentiate of Technology can be obtained halfway between an MSc and a PhD. The examination for this degree is less formal than for a PhD but it requires the completion of a thesis and a public seminar.
Abstract

Active secondary suspension in trains has been studied for a number of years, showing promising improvements in ride comfort. However, due to relatively high implementation and maintenance costs, active technology is not being used in service operation to a large extent. The objective of this study is to develop an active lateral secondary suspension concept that offers good ride comfort improvements and enables centring of the carbody above the bogies when negotiating curves at unbalanced speed. Simultaneously, the active suspension concept should be a cost-effective solution for future series production. The thesis consists of an introductory part and three appended papers.

The introductory part describes the concept of active secondary suspension together with different actuator types and control methods. Further, the present simulation model and applied comfort evaluation methods are presented. The introductory part also comprises a summary of the appended papers, an evaluation of track forces and suggestions for further work.

Paper A presents the initial development of an active lateral secondary suspension concept based on sky-hook damping in order to improve vehicle dynamic performance, particularly on straight tracks. Furthermore, a Hold-Off-Device (HOD) function has been included in the suspension concept in order to centre the carbody above the bogies in curves and hence avoid bumpstop contact. Preparatory simulations as well as the subsequent on-track tests in the summer of 2007 showed that the active suspension provides improved passenger ride comfort and has significant potential to be a cost-effective solution for future implementation.

In Paper B, measurement results from on-track tests performed in 2008 are presented. 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 in order to enable detection of transition curves and to switch off the dynamic damping in these sections. Ride comfort in the actively suspended carbody was significantly improved compared to that in the passively suspended car. The satisfactory results led to implementation of the active suspension system in long-term tests in service operation in the beginning of 2009.

In Paper C, a quarter-car model in MATLAB has been used to investigate a more advanced control algorithm: $H_{\infty}$ instead of sky-hook. $H_{\infty}$ control provides more flexibility in the design process due to the possibility to control several parameters. In particular, this is done by applying weight functions to selected signals in the system. When comparing the two control strategies through simulations, the results show that $H_{\infty}$ control generates similar carbody accelerations at the same control force as sky-hook; however, the relative displacement is somewhat lower.

Keywords: railway, active suspension, ride comfort, sky-hook, $H_{\infty}$, multi-body simulation, on-track tests
Outline of thesis

The scope of this thesis is development of an active lateral secondary suspension concept for a high-speed rail vehicle. The thesis includes an introduction, which also summarises a literature survey [33], and the following appended papers:

**Paper A**  

**Paper B**  

**Paper C**  

Planning of the preparatory simulation work 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.
Thesis contribution

This thesis presents an active secondary suspension solution that provides good ride comfort improvements, but still to a reasonable cost. The final goal is to offer a good solution for series production.

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

- A literature survey in the introductory part and in [33] has been compiled, which covers various active suspension concepts, actuator types and control strategies.

- An existing vehicle model, received from Bombardier Transportation, has been modified for this study and validated against measurement results. The agreement between measurements and simulations is good up to approximately 10 Hz.

- The complete control strategy needed for running on straight track, in curves and in transition curves is developed and optimised through simulations, using properties of a real existing actuator.

- Comparisons of the performance of a single-variable sky-hook control compared to a multivariable $H_\infty$ control for an active lateral suspension in trains are carried out.

- Ride comfort improvements have been proved by multi-body simulations as well as on-track tests.
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1 Introduction

1.1 Background

It is difficult to further improve running behaviour of a rail vehicle, especially ride comfort, with conventional passive suspension when vehicle speed tends to increase. By means of implementing active technology in the secondary suspension system, the potential of improving ride comfort increases. The concept of active technology in rail vehicles has been analysed theoretically and experimentally since the 1970s, but has not yet made its convincing breakthrough in operational use (except for tilting train technology), as has been experienced in, for example, aircraft and automotive industry. The likely reason for the non-success of implementing and maintaining active technology in rail vehicles is the high costs. 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 utilised 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 worse,

d) increase carbody width.

If ride comfort is already good, further improvements at unchanged vehicle speed and track conditions are generally not justified due to high costs of implementing active suspension. However, goal b) allows 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 saved. Further, an increased carbody width allowing more seats can be economically very interesting for an operator.

Active technology, in general, is based on the idea of controlling a certain signal with the signal itself, i.e. by means of a closed loop. In order to achieve this control loop in the suspension of a rail vehicle, actuators, sensors and a controller must be added to the mechanical system. The actuators replace conventional passive dampers, for example between carbody and bogies. They should actively generate a required force demand according to a control law specified in the controller. The control law can, for example, use acceleration signals measured by sensors in order to calculate the required force to the actuator. The accelerations, in turn, depend on the generated actuator force. Hence, the control loop is closed. The principle of an active suspension system is shown in Figure 1-1.
Introduction

Figure 1-1 The concept of an active suspension system

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

1.2 The present thesis

Since 2005, a Swedish research and development programme, Gröna Tåget (Green Train) [21], is being carried out. Its main goal is to develop the next generation of high-speed trains for Nordic conditions, increasing vehicle speed from today’s 200 up to 250–300 km/h. This thesis is one part of the Gröna Tåget programme, focusing on active lateral secondary suspension in a high-speed rail vehicle. The active suspension is developed in co-operation with Bombardier Transportation, Västerås (Sweden).

The concept of active secondary suspension is described in Chapter 2. In Chapter 3 the function, advantages and drawbacks of various actuator types are described. Chapter 4 presents different control methods that are commonly used for active technology in rail vehicles. A description of the present simulation model is given in Chapter 5. The applied ride comfort evaluation methods are introduced in Chapter 6. A summary of the appended papers along with an evaluation of track forces is given in Chapter 7. Finally, conclusions and further work are presented in Chapter 8.
2 Active secondary suspension

The main goals with active technology in rail vehicles can be divided into two categories: improving running stability and wheelset guidance (mainly controlled through the primary suspension), and improving passenger ride comfort (controlled through various modifications of the secondary suspension). This thesis focuses on active suspension to improve ride comfort. Improving running stability and wheelset guidance by means of active suspension has briefly been mentioned in the literature survey compiled by the author [33].

Previous extensive surveys that include different suspension concepts of active technology in rail vehicles origin from Hedrick and Wormley in 1975 [18], Goodall and Kortüm in 1983 [14], Goodall in 1997 [15] and Bruni, Goodall, Mei and Tsunashima in 2007 [6].

2.1 Active suspension to improve ride comfort

Compared to passive damping, active control of the secondary suspension provides better isolation of the carbody from excitations transmitted from track irregularities. Hence, passenger comfort is improved. The secondary suspension is normally controlled in the lateral direction, including the yaw mode, or in the vertical direction, including the pitch mode. Active control of the roll mode of the secondary suspension belongs to the tilting concept.

There are various alternatives how the actuators can be implemented in the secondary suspension. Firstly, the actuators can be fitted in the bogie environment in combination with the existing passive components, either in series or in parallel. Fitting the actuator in parallel with a passive spring enables reduced actuator size, since the spring principally can be 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. The solution with actuators in combination with passive components is particularly used when the actuators are considered not to be able to handle possible failure modes. Hence, the passive components act as a back-up in case of actuator failure. Active secondary suspension implemented in this way can, therefore, probably be regarded as non-safety critical, which makes the acceptance for this technology much easier. The second alternative is when the passive components are completely replaced by actuators. This requires reliable actuators that secure an ability to work in passive mode in case of actuator failure.
**Dynamic control**

The obvious application of active secondary suspension is to control oscillations of the carbody due to imperfections in the track. The actuator should be able to control vibrations between approximately 0.5 Hz and 4–10 Hz, depending on application. The control task is usually to minimise accelerations in the carbody.

**Hold-Off-Device**

When travelling in a curve at high speed, causing large lateral accelerations, a quasi-static displacement between carbody and bogies arises. The carbody moves laterally outwards in the curve and there is a risk of hitting the bumpstops. When bumpstop contact occurs it has a negative impact on ride comfort, since track irregularities are transmitted up to the carbody via the bumpstops without the normal filtering ability of the secondary suspension. Therefore, it is desirable to avoid bumpstop contact. The gap between bogie frame and bumpstop (connected to the carbody) is 46 mm on the considered vehicle in the present study (Figure 2-1). The bumpstop is a progressive rubber component with significant stiffness and after 65 mm there is a metallic stop. If the quasi-static displacement of the carbody can be minimised, not only can good ride comfort be maintained, but a wider carbody is also possible, since the play between carbody and bogie does not have to be as large as before. Furthermore, if the position of the bumpstops is changed, and hence the play between carbody and bogie is decreased, cross wind stability can be improved.

![Figure 2-1 Bumpstop connected to the carbody with a gap of 46 mm to the bogie frame](image-url)
In order to achieve a centred position of the carbody above the bogies, a so-called Hold-Off-Device (HOD) can be used. This application is also called low-bandwidth control, since it detects low frequencies of deterministic track inputs, i.e. curves, in order to minimise the lateral displacement of the carbody in relation to the bogies.

The HOD concept was introduced in the early 1990s by Allen [1] when a hydraulic actuator prototype was designed and tested. In comparison with a conventional passive solution the active HOD prototype showed significantly reduced dynamic lateral acceleration and hence improved ride comfort. However, the concept of low-bandwidth control was mentioned already 1983 by Pollard [34]. Further research has been performed by Stribersky, Steidl, Müller and Rath [43], and Stribersky, Müller and Rath [44], where the benefit of active lateral positioning has been proven through comparisons between simulated and measured results of passive and active solutions, respectively.

2.2 Degree of control

Conventional passive suspension has a rather simple design and is cost-effective compared to active suspension. On the other hand, possibilities of further improvements are restricted, for example regarding passenger comfort. Therefore, implementation of active technology in rail vehicles will probably become more common. There are two general concepts of active suspension – fully-active and semi-active – basically governed by the required amount of external power, as described by Jalili [24].

So-called fully-active suspension offers high performance control and gives the best response in a wide frequency bandwidth. In a diagram with actuator force versus actuator velocity it works in all four quadrants, 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 in [28].

Between the passive and the fully-active solutions there is the semi-active suspension system (sometimes referred to as controlled damping). It adapts the advantages of the passive suspension as being safe and cost-effective together with a rather good performance. In comparison with the fully-active suspension it is less complex and does not require an external power supply. However, energy can not be transferred to the system, but only dissipated from it, which follows from the possibility to work only in the first and third quadrant of the force-velocity diagram, shown in Figure 2-2. The actuator force depends on relative damper velocity, i.e. velocity difference between the two bodies where the actuator is situated. Large actuator forces cannot be generated at low velocities. Furthermore, the actuator in a semi-active system is not able to develop a force in the opposite direction as the relative damper velocity. With opposite signs a zero-force is applied, which is described by Roth and Lizell [38]. Despite this drawback, the semi-active suspension can still work in passive mode if failure of the control system occurs.

The concepts of semi-active and fully-active suspensions are schematically shown in Figure 2-3.
**Figure 2-2**  Force-velocity diagram for semi-active damping, Goodall and Mei [16]

**Figure 2-3**  Concepts of semi-active and fully-active suspension control, Goodall and Mei [16]
A comparison between fully-active and semi-active suspension concepts has been performed by Ballo [4], however, applied to a quarter-car goods road vehicle. When analysing the \textit{rms} value of the sprung mass acceleration the semi-active suspension concept offers as good reduction as the fully-active, additionally, to much lower power consumption. In contrast, the \textit{rms} value of the force transmitted to the roadway is significantly reduced with the fully-active system compared to the semi-active. It was also shown that the fully-active suspension concept offers possibilities of further increase of the effectiveness (however, at the expense of increased power consumption), whereas the semi-active concept is rather limited.

JR West in Japan needed to improve ride comfort when aiming at commercial operation speed of 300 km/h. After having optimised passive suspension parameters, a need of further comfort improvement still remained. Both fully-active and semi-active secondary suspension systems were implemented on a Shinkansen train Series 500 to perform experimental investigations. The fully-active suspension was applied to the end cars, however, only operating on the rear car in the direction of travel. The actuators were placed in parallel with existing passive dampers. The passive components were kept in case of actuator failure. The semi-active dampers were implemented on three first-class cars and two pantograph-equipped cars, replacing the existing passive dampers. The results showed that both suspension systems offered satisfactory ride quality improvements. However, due to mass production cost of the active system it was considered that the requirements were sufficiently fulfilled with the semi-active suspension system. Therefore, the fully-active suspension was replaced by semi-active suspension before the train was taken into service operation. This study has been described by Norinao in [31].

Tanifuji, Koizumi and Shimamune [46] have summarised the Japanese work performed over the years regarding active applications in rail vehicles. The study focuses on active suspension to improve ride comfort, but shortly also deals with active technology in driving and braking, carbody tilting, steering and pantographs. In many observed studies fully- and semi-active suspensions are being compared, with the result that fully-active actually offers better performance and ride comfort improvements. However, when it comes to implementing active suspension for operational use the semi-active concept is unexceptionally chosen. The explanation is that fully-active suspension is more costly and that railway companies not yet fully believe in the system’s reliability. The study also mentions the rise of interest in research combining active vibration control with carbody centring control, in order to avoid bumpstop contact in curves at high track plane acceleration (previously mentioned in Section 2.1 as Hold-Off-Device).
Active secondary suspension
3 Actuator types

Several actuator types have been studied in the area of railway technology over the years. The following sections give an overview of the different types, the concept of how they work and their advantages and disadvantages. Other studies that have collected general information about different actuator types have been performed by Brabie [5], Kjellqvist [28] and Thomas [47].

The choice of actuator is dependent on the trade-off between, on the one hand, actuator performance and, on the other hand, cost considerations. The ideal actuator design is most likely not possible to physically implement, since it would not be economically justified.

3.1 Electro-mechanical

An electro-mechanical actuator is powered by an electrical motor (AC or DC), which is able to rotate a screw mechanism (e.g. a roller or a ball screw). The rotational motion, or torque, of the screw is transferred to a translational motion, or force, which acts on the body that the actuator is mounted on [28]; see Figure 3-1. Electro-mechanical actuators are in general less compact than other actuator types. In [15] it is stated that they can encounter problems with reliability and life of mechanical components.

The development of electro-mechanical actuators has been in progress during at least three decades. The performance of an electro-mechanical actuator in a rail vehicle was studied already in 1984 in England by Pollard and Simons [35]. Furthermore, experimental research in the late 1990s in France using electro-mechanical actuators has been described by Gautier, Quetin and Vincent [11]. The electro-mechanical actuator was chosen due to its low noise levels and compact design (what stands in contrast to what has just been mentioned). Lately, an electro-mechanical actuator with a roller screw has been analysed by Kjellqvist [28] and Kjellqvist, Sadarangani and Östlund [27], making a suggestion on how to manage the design conflict between actuator size, temperature and dynamic properties.
3.2 Electro-magnetic

The electro-magnetic actuator consists of two pairs of electro-magnets mounted back to back operating in attraction mode. The magnets produce a force in both directions between two masses connected through the actuator, e.g. carbody and bogie. In a study by Foo and Goodall [9] an electro-magnetic actuator was added between the centre of the carbody and an auxiliary mass of one ton in order to suppress the first symmetrical flexible mode, which, if unsuppressed, has a negative impact on ride comfort.

The electro-magnetic actuator is often preferred because of its property of large frequency bandwidth. It is considered to show good frequency response up to 50 Hz. Since it does not contain any moving parts it is a robust and reliable device, described by Pollard [34]. However, it suffers from a relatively high unit size and weight, and can be difficult to fit in narrow places between two bodies of the vehicle. The effect of air gap variations between the magnets causes an unstable system, which, certainly, can be mitigated with proper force feedback, according to Goodall, Pearson and Pratt [17], and Pollard and Simons [35].

3.3 Hydraulic

There exist several variations of hydraulic actuators used in active railway technology, sometimes mentioned as servo-hydraulic and sometimes as electro-hydraulic, with no clear distinction between the two concepts. Accordingly, the use of these actuators in the literature is mostly referred to as just hydraulic.

The general concept of hydraulic actuators is based on the idea that a control signal activates valves or a pump controlling the flow of the hydraulic fluid into and out from the actuator. Hereby, a pressure difference appears between the two chambers of the actuator cylinder, which, in turn, gives rise to the actuator force. Figure 3-2 shows the basic principle of an electro-hydraulic actuator with a hydraulic-filled cylinder consisting of two chambers divided by a movable piston.
Generally, hydraulic actuators have a fast response time and they are able to maintain a demanded loading capacity indefinitely without excessive heat generation. However, hydraulic systems are highly non-linear and subject to parameter uncertainty, described by Niksefat and Sepehri [30].

Hydraulic actuators are well studied and often used in railway applications. They are compact and can easily be fitted in narrow spaces between carbody and bogie [9]. Their cost-effectiveness makes them favourable to be implemented in vehicles for full-scale tests. An experimental analysis was performed by Shimamune and Tanifuji [40], who chose a hydraulic actuator before a pneumatic actuator because of its ability to control in a frequency range up to about 10 Hz, compared to 2–3 Hz (as is described in the following section). The major disadvantage with hydraulic actuators is the risk of oil leakage. Furthermore, questions regarding maintainability and maintenance costs of hydraulic actuators can be raised [17].

![Figure 3-2 Principal function of an electro-hydraulic actuator](image)

A hydraulic actuator was used for the Hold-Off-Device system [1], described in Section 2.1, where the aim was to maintain a centred position of the carbody above the bogies in curves at high speed. The hydraulic actuator was preferred to a pneumatic or electrical system, since the other two systems were either too bulky or more expensive.

The present study uses an electro-hydraulic actuator, since it is considered to be a cost-effective option for future series production, while still yielding satisfactory performance. It is described in more detail in Paper B.
3.4 Servo-pneumatic

In an active servo-pneumatic system the air pressure is controlled, which gives rise to desired suspension characteristics. In vertical direction the air pressure in an already existing air spring system with fixed reservoir volume can be actively controlled by a reservoir with variable volume, as described by Pollard and Simons [35]. In lateral direction the pneumatic actuator can be of the same principle as a hydraulic actuator, but instead varying the air pressure by controlling the air flow into and out from the actuator cylinder.

The advantage with servo-pneumatic actuators is that they can be linked to already existing pneumatic systems of the vehicle (e.g. to air springs and the braking system). Components of the actuator are relatively cheap and there is no liquid that can leak. However, due to large air compressibility the controllable frequency bandwidth is restricted to 2–3 Hz, and hence the efficiency of the actuator is limited.

In an investigation performed in the mid 1990s by Sasaki, Kamoshita and Enomoto [39] servo-pneumatic actuators were tested on a roller rig in order to reduce vibrations in vertical, lateral and roll modes. Up to 50% reduction of these particular modes of vibration could be shown with the active system. Another Japanese study was performed at approximately the same time by Hirata, Koizumi and Takahashi [19], where an experimental rail vehicle was used. One of two passive lateral hydraulic dampers was replaced by a pneumatic actuator. With active suspension the lateral, yaw and roll motions caused by track irregularities could be reduced. Furthermore, in the late 1990s, still in Japan, each bogie of a test vehicle was equipped with one pneumatic actuator in parallel with the existing passive lateral dampers. However, these pneumatic actuators were rather weak compared to, for example hydraulic actuators, since they could only produce a force up to 7 kN [31]. Nevertheless, lateral accelerations in the carbody could be reduced although vehicle speed was higher than in a passive suspension case.

3.5 Rheological

There are two types of rheological actuators, namely electro- and magneto-rheological actuators (ER or MR). They are cylindrical dampers divided into two chambers by a movable piston. The chambers are entirely filled with a low-viscosity fluid containing fine electric or magnetic particles. The actuator is exposed to an electric or a magnetic field, with help of electrodes or electro-magnets, respectively. The viscosity of the fluid is varied according to the strength of the field applied to the actuator. Hence, the damping characteristics of the actuator are varied. The stronger the field, the larger the actuator force. The increase in viscosity can be as much as 10 times higher with the MR fluid, according to Yao, Yap, Chen, Li and Yeo [50]. In Figure 3-3 a schematic picture of the electro-rheological concept is shown.
The ER and MR actuators are relatively cheap to manufacture and have low energy consumption. The response to the electric or magnetic field is fast, which enables a wide control bandwidth, which has been stated by Choi, Choi, Chang, Han and Kim [8], and Gao and Yang [10]. The MR actuator has been analysed in an experimental test rig by means of semi-active control in [10]. It was concluded that the MR actuator could generate damping forces in a very broad range under the influence of a magnetic field.

![Diagram of an electro-rheological actuator](image)

**Figure 3-3 Principal function of an electro-rheological actuator**
4 Control strategies

In order to enable steering and control of the actuators in a favourable way an appropriate control algorithm is needed. Several control strategies have been studied and implemented in the area of active technology within rail vehicles.

4.1 PID control

Classical loop-shaping with a proportional-integral-derivative controller ($PID$ controller) is widely used in industrial control systems. The $PID$ controller creates an input signal $u$ to the system process by attempting to correct the error between a demanded reference signal $r$ and the actual output signal $y$: $e(t) = r(t) - y(t)$ (Figure 4-1). The $PID$ controller is described as

$$u(t) = K_P e(t) + K_I \int_0^t e(s)ds + K_D \frac{d}{dt} e(t),$$

(4-1)

where $K_P$, $K_I$ and $K_D$ are controller coefficients for the proportional, integral and derivative parts, respectively. Appropriate design of the coefficients makes it possible to achieve a control system with desired performance characteristics. The $PID$ algorithm is relatively simple and offers a robust performance. The largest challenge is to find the appropriate design of the control parameters. To facilitate the fine-tuning of the control parameters, different methods have been developed for this purpose, e.g. the Ziegler-Nichols method, as described by Glad and Ljung [12].

Increased $K_P$, i.e. tuning of the proportional part, enables a faster controller, whereas an increased integral part, $K_I$, eliminates errors in the output signal. However, both $K_P$ and $K_I$ decrease the margins of stability, so by increasing the derivative part, $K_D$, possible instability can be suppressed.
4.2 Sky-hook damping

One of the most implemented control algorithms in the area of active technology in trains is the so-called sky-hook damping. The name is based on the idea that the system is damped relative to a fictive sky reference point, instead of the ground (Figure 4-2). The concept of sky-hook damping is described in Paper A.

The strategy of sky-hook damping was first introduced by Karnopp in the late 1970s and a comprehensive description was published in 1983 [26]. Thereafter, sky-hook damping has been thoroughly investigated and analysed by various researchers throughout the years. Stribersky, Kienberger, Wagner and Müller [45] have through simulations showed that sky-hook damping significantly reduces resonance peaks and \( rms \) acceleration, 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 [38] in the late 1990s could 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 large challenge with active sky-hook damping is to be able to optimise the trade-off between enhanced comfort and suspension deflection during curving. Nevertheless, acceptable results can be achieved by optimising the filtering of the absolute velocity signal. Li and Goodall [29] 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 improved ride quality 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, showed over 50 % ride quality improvement, however, with larger suspension deflection than as for the passive case.

Hohenbichler and Six [20] 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, Hohenbichler, Six and Abel [3] have performed simulations using preview data (accelerations) in combination with sky-hook damping in order to optimise 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 deterministic track curvature.

4.3 $H_\infty$ control

$H_\infty$ control is a more advanced control methodology, which is concerned with finding a controller $K_c$ for the open-loop system $G_0$, such that the closed-loop system $G_{ec}$ has good performance, stability and robustness. Figure 4-3 illustrates a typical configuration of a general control system. The closed-loop system $G_{ec}$ is a transfer matrix from the external disturbance signal $w$ to the error signal $z$. Moreover, the measurement signal $y$ is used in $K_c$ to calculate the control signal $u$. In order to achieve secured stability and robustness of the system the signal $z$ should be minimised. The $H_\infty$ concept is described in Paper C together with studies concerning this control method.
Another control theory that is concerned with optimisation is the so-called LQ (Linear Quadratic) control, or extended to LQG (Linear Quadratic Gaussian) control. A dynamic system that is described through linear differential equations and a quadratic cost function that should be minimised is called an LQ problem. If normally distributed (Gaussian) disturbances are considered the control theory is extended to LQG, as described by Glad and Ljung [13].

A linear system can be described on state-space form,

\[
\begin{align*}
\dot{x}(t) &= Ax(t) + Bu(t) + Nv_1, \\
y(t) &= Cx(t) + v_2
\end{align*}
\]

where \(x\) is the system state vector, \(u\) is the control signal, \(y\) is the output signal and \(v_1\) and \(v_2\) are white noise intensity signals. \(A\) is the system matrix, \(B\) the input matrix, \(N\) the disturbance input matrix and \(C\) the output matrix. The quadratic cost function \(J\) that should be minimised is described as the sum of the quadratic norm of the control error \(e\) and the control signal \(u\), respectively,

\[
J = \min \left( \|e\|^2_{Q_1} + \|u\|^2_{Q_2} \right) = \min \left\{ \int e^T(t)Q_1 e(t) + u^T(t)Q_2 u(t) dt \right\},
\]

where \(Q_1\) and \(Q_2\) are weighting functions. The optimal linear feedback control law that minimises the cost function \(J\) is given by
where Equation (4-4b) is the Kalman filter for the system, which estimates the system states that are not directly measurable. The matrix \( K \) (Kalman filter gain) is determined by

\[
K = (PC^T + NR_{12})R_2^{-1},
\]

(4-5)

where \( P \) is the solution to the Riccati matrix equation

\[
AP + PA^T - (PC^T + NR_{12})R_2^{-1}(PC^T + NR_{12})^T + NR_1N^T = 0
\]

(4-6)

and \( R_i \ (i = 1, 2, 12) \) are elements of the intensity matrix. \( L \) in Equation (4-4a) is given by

\[
L = Q_2^{-1}B^T S,
\]

(4-7)

where \( S \) is the solution to the Riccati matrix equation

\[
A^T S + SA + Q_1 - SBQ_2^{-1}B^T S = 0.
\]

(4-8)

A study that has assessed the LQG control law was performed by Pratt and Goodall [36]. Its aim was to compare traditional active secondary suspension between each bogie and the carbody with the alternative approach of active suspension between carbodies, so-called inter-vehicle suspension. The conclusion was drawn that active inter-vehicle suspension could achieve reduction of motions in either bounce or pitch direction, but not at the same time. Thus, it is a matter of trade-off and it depends on how the weighting of the cost function of the LQG theory is designed.

Another study by Pratt and Goodall [37] deals with the same subject of active inter-vehicle damping by means of LQG control law. The trade-off between ride quality and suspension deflection is dependent on the weighting of the cost function of the LQG controller. Simulations with an active three-car inter-vehicle model were performed and the results showed that ride comfort in the centre carbody could be considerably improved compared to a corresponding passive inter-vehicle model. On the other hand, ride comfort in the two outer carbodies was deteriorated. Conclusions drawn from the simulations were that active inter-vehicle damping does not show significant
Control strategies

improvement in ride comfort compared to passive inter-vehicle damping, but however, offers further development possibilities.

Shimamune and Tanifuji [40] have performed experimental analyses of an oil-hydraulic actuator applied to a half-car rail vehicle model. An $LQG$ controller was used with estimated state variables through a Kalman filter, which slightly deteriorated the $LQG$ control performance compared to an $LQ$ controller. It was also recommended to use the weight of an empty carbody when designing the controller, since problems with performance and stability may occur if the actual carbody weight becomes lower than the weight used for the controller design.

Tibaldi and Zattoni [48] have performed a study investigating $LQ$ and $LQG$ control law applied to active suspension design. A quarter-car vehicle model was used where a non-linear hydraulic actuator was linearized in order to design the $LQ$ and $LQG$ controllers. Despite the fact that the $LQG$ control law uses Kalman filtering to estimate immeasurable states it does not cause significant performance loss compared to the $LQ$ controller.
5 Simulation model

The simulation model used in the present study has been built-up in the multi-body systems (MBS) simulation software SIMPACK [23] and was originally developed by Bombardier Transportation. It constitutes a one-car vehicle with two bogies, and models a Swedish Regina train (Figure 5-1). However, the original Regina bogies are replaced with new high-speed bogies that are developed within the Gröna Tåget research programme (Figure 5-2).

![Figure 5-1 Regina test train within the Gröna Tåget research programme. Photo courtesy of Bombardier Transportation.](image1)

![Figure 5-2 The new Regina bogie. Photo courtesy of Bombardier Transportation.](image2)

In the simulation model, the carbody is rigid and connected to two motor bogies through the secondary suspension, modelled by a non-linear airspring. Additionally, the secondary suspension consists of two non-linear vertical dampers per bogie, one longitudinal traction rod in the middle of each bogie, one non-linear yaw damper at each side of the bogie and one anti-roll bar per bogie with linear stiffness. The two conventional lateral dampers in each bogie are replaced by one actuator, placed diagonally in relation to the actuator in the other bogie (Figure 5-3).
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. Rail inclination is 1:30. Creep forces are calculated with the FASTSIM algorithm [25]. Measured track irregularities are applied as excitation.

The applied control algorithm is modelled in the MATLAB tool Simulink, which also includes a model of the actuator, provided by Liebherr [22]. The co-simulation interface SIMAT enables communication between SIMPACK and MATLAB/Simulink; see Figure 5-4. The sample time period of the simulation is divided into equidistant time steps. Between each time step, both processes, SIMPACK and MATLAB, are solving their part of the equation system independently and then exchange data [23].

A schematic picture of the control loop is shown in Figure 5-5. Vehicle output parameters, such as displacements, velocities and accelerations of the carbody and bogies are sent to the control model in Simulink. The signals are divided into lateral and yaw motions, filtered and the required force demand is fed to the actuator model; see Figure 5-6 (originating from Paper A). The actuator delivers the control force back to the vehicle model in SIMPACK. Hence, the control loop is closed.
Figure 5-5  Control loop of the simulation model

Figure 5-6  Filtering of control signals creates a force demand to the actuator (from Paper A).
Simulation model
6 Ride comfort evaluation

Ride comfort is measured by evaluating accelerations in the carbody. Firstly, this section describes the accelerations experienced by the passengers. Secondly, the methods used for comfort evaluation in the present study, \( W_z \) and ISO 2631, are presented.

6.1 Accelerations

When a rail vehicle enters a horizontal curve it tends to move laterally outwards in the curve and the vehicle is exposed to accelerations. On the one hand, there is a centrifugal acceleration, \( v^2/R \), which is parallel to the horizontal plane and, on the other hand, a gravitational acceleration, \( g \), parallel to the vertical plane. The resulting acceleration vector of these two can be seen in Figure 6-1 (a). This vector can alternatively be divided into acceleration vectors according to a track-following coordinate system, which are parallel and perpendicular to the track plane; see Figure 6-1 (b). The acceleration parallel to the track plane, \( a_z \), is often called track plane acceleration. With a high track plane acceleration the ride comfort is negatively affected. Further, the resulting acceleration vector can also be divided into acceleration vectors according to a carbody-following coordinate system, parallel and perpendicular to the carbody plane; see Figure 6-1 (c). The acceleration parallel to the carbody plane, \( a_{yc} \), is often called carbody plane acceleration. This acceleration is used for the ride comfort evaluation described in Sections 6.2 and 6.3.

\[ \text{Figure 6-1} \quad \text{Definition of track plane acceleration } a_z \text{ and carbody plane acceleration } a_{yc}. \quad \text{The three subfigures are equivalent to each other [2].} \]

(a) Accelerations in earth-following coordinate system
(b) Accelerations in track-following coordinate system
(c) Accelerations in carbody-following coordinate system
The quasi-static accelerations $a_y$ and $a_z$ are given by

$$a_y = \frac{v^2}{R} \cos \varphi_l - g \sin \varphi_l = \frac{v^2}{R} \cos \varphi_l - g \frac{h_t}{2b_0} \approx \frac{v^2}{R} - g \frac{h_t}{2b_0} \quad (6-1)$$

$$a_z = \frac{v^2}{R} \sin \varphi_l + g \cos \varphi_l \approx g \quad (6-2)$$

where $h_t$ is the track cant, $\varphi_l$ the track cant angle and $2b_0$ is the track base. The approximations can be made when the track cant angle $\varphi_l$ is small (often less than 7 degrees), which is almost always the case. The side force angle $\Phi$ in Figure 6-1 (b), is related to the accelerations $a_y$ and $a_z$.

The accelerations in the carbody reference system are given by

$$a_{yc} = \frac{v^2}{R} \cos (\varphi_l + \varphi_c) - g \sin (\varphi_l + \varphi_c) \approx \frac{v^2}{R} - g \sin (\varphi_l + \varphi_c) \quad (6-3)$$

$$a_{zc} = \frac{v^2}{R} \sin (\varphi_l + \varphi_c) + g \cos (\varphi_l + \varphi_c) \approx g \quad (6-4)$$

where $\varphi_c$ is the carbody roll angle in relation to the track. In the case of cant deficiency and lack of carbody tilt, this angle is negative relative to the track plane. The approximations can be made when $v^2/R$ is less than approximately $0.3g$ and $0 \leq \varphi_l + \varphi_c < 0.2 \text{ rad} \ (\approx 12^\circ) \ [2]$. 
6.2 Wertungszahl ($Wz$)

$Wz$ is a ride comfort number that originates from German research in the 1940s and 1950s by Sperling and Betzhold [41][42]. 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

$$Wz = 4.42(a_{rms})^{0.3}$$  \hspace{1cm} (6-5)

where $a_{rms}$ 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 illustrated in Figure 6-2 and are described by the following equations [32]:

$$B_l(f) = 0.737 \sqrt{\frac{1.911 f^2 + (0.25 f^2)^2}{(1 - 0.277 f^2)^2 + (1.563 f - 0.0368 f^2)^2}}$$  \hspace{1cm} (6-6)

$$B_v(f) = \frac{0.588}{0.737} B_l(f)$$  \hspace{1cm} (6-7)

With $f = \omega/(2\pi)$ and $s = i\omega$, the corresponding transfer function for lateral $Wz$ is written as

$$B_l(s) = 0.737 \frac{0.25}{(2\pi)^2} s^2 + \frac{\sqrt{1.911}}{2\pi} s \frac{0.0368}{(2\pi)^3} s^3 + \frac{0.277}{(2\pi)^2} s^2 + \frac{1.563}{2\pi} s + 1$$  \hspace{1cm} (6-8)

According to $Wz$ evaluation, human subjects are considered to be most sensitive to frequencies in the 4–7 Hz range, laterally as well as vertically, cf. Figure 6-2. Table 6-1 describes the ride comfort levels evaluated by $Wz$. The levels are the same for the lateral as well as the vertical direction.
Figure 6-2 Frequency weighting of accelerations according to Wz ride comfort evaluation (lateral and vertical)

Table 6-1 Ride comfort levels (lateral and vertical) evaluated by Wz

<table>
<thead>
<tr>
<th>Comfort Level</th>
<th>Magnitude (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Very comfortable</td>
<td>1.0–2.0</td>
</tr>
<tr>
<td>Comfortable</td>
<td>2.0–2.5</td>
</tr>
<tr>
<td>Less comfortable</td>
<td>2.5–3.0</td>
</tr>
<tr>
<td>Unpleasant</td>
<td>&gt; 3.0</td>
</tr>
</tbody>
</table>
6.3 ISO 2631

Ride comfort evaluation according to ISO 2631 is well described in the European standard prEN 12299 [7]. The \( \text{rms} \) values of frequency weighted accelerations on the carbody floor level are evaluated as

\[
a_{\text{rms}}^{w} = \left[ \frac{1}{T} \int_{0}^{T} \left[ a_{w}(t) \right]^{2} \, dt \right]^{0.5}
\]  

(6-9)

where \( a_{w}(t) \) is the frequency weighted acceleration as a function of time \( t \). \( T = 5 \text{ s} \) is the duration of the measurement. The filter functions for the horizontal and vertical directions are illustrated in Figure 6-3.

The weighting filter for horizontal comfort evaluation is a product of band-limiting and transition filters: \( H(s) = H_{u}(s)H_{o}(s)H_{w}(s) \), where \( H_{u}(s) \) is a high-pass second order filter

\[
H_{u}(s) = \frac{s^{2}}{s^{2} + \frac{2\pi \cdot 0.4}{0.71}s + (2\pi \cdot 0.4)^{2}}
\]  

(6-10)

with cut-off frequency at 0.4 Hz, \( H_{o}(s) \) is a low-pass second order filter

\[
H_{o}(s) = \frac{(2\pi \cdot 100)^{2}}{s^{2} + \frac{2\pi \cdot 100}{0.71}s + (2\pi \cdot 100)^{2}}
\]  

(6-11)

with cut-off frequency at 100 Hz and \( H_{w}(s) \) is a transition filter where weighting is proportional to acceleration at lower frequencies and to velocity at higher frequencies

\[
H_{w}(s) = \frac{(s + 2\pi \cdot 2)4\pi}{s^{2} + \frac{2\pi \cdot 2}{0.63}s + (2\pi \cdot 2)^{2}}.
\]

(6-12)

According to ISO 2631 evaluation, human subjects are considered to be most sensitive to frequencies in the 0.5–2 Hz range in horizontal direction and in the 4–10 Hz range in vertical direction, cf. Figure 6-3. The ride comfort levels for the individual lateral and vertical directions according to ISO 2631 are described in Table 6-2. The levels are the same for the lateral as well as the vertical direction.
When comparing the filter functions for $W_z$ and ISO 2631 for lateral comfort evaluation it is clear that the results will differ. Therefore, it is preferable to use both evaluation methods, in order to be able to make a fair judgement of the passenger comfort. In Paper A, only $W_z$ is used as ride comfort evaluation method, whereas both methods are used in Paper B.

![Figure 6-3](image)

*Figure 6-3  Frequency weighting of accelerations according to ISO 2631 (horizontal and vertical)*

<table>
<thead>
<tr>
<th>Ride comfort levels (lateral and vertical) evaluated by ISO 2631 [7]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Very comfortable</strong></td>
</tr>
<tr>
<td><strong>Comfortable</strong></td>
</tr>
<tr>
<td><strong>Medium</strong></td>
</tr>
<tr>
<td><strong>Less comfortable</strong></td>
</tr>
</tbody>
</table>
7 Summary of present work

**Paper A** presents the initial development of an active lateral secondary suspension concept based on sky-hook damping in order to improve vehicle dynamic performance, particularly on straight tracks. Furthermore, a Hold-Off-Device (HOD) function has been included in the suspension concept in order to centre the carbody above the bogies in curves and hence to avoid bumpstop contact. By avoiding bumpstop contact in curves not only is ride comfort improved, but also a wider carbody is possible since the bumpstops can be moved to new tighter positions.

Moreover, experience from previous on-track tests with conventional passive secondary suspension has shown some problems with low-frequency periodic motions when travelling in large-radius curves at high speed (high track plane acceleration), which negatively affect ride comfort. The on-track tests with active lateral secondary suspension showed that the carbody centring HOD function is able to suppress these unfavourable motions.

Preparatory simulations as well as the subsequent on-track tests in the summer of 2007 showed that the active suspension manages to improve passenger ride comfort. Hence, the presented active suspension concept is considered to have significant potential to be a cost-effective solution for future implementation. Furthermore, validation of the simulation model showed good agreement up to approximately 10 Hz, which is considered to be sufficient for the present purpose.

Since results from the first paper showed very good potential of ride comfort improvements by means of the carbody centring HOD function, 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. In transition curves, the dynamic control will keep the carbody in the tangent direction and hence increase the displacement relative to the bogie. The gyroscope detects transition curves by means of the bogie frame roll velocity; information that is used to switch off the dynamic damping in these sections. The relative lateral displacement between carbody and bogies in curve sections was compared to corresponding measurement results from the previous year. It could be shown that the relative displacement in transition curves was lower when the active damping was not in use.

Ride comfort in the actively suspended carbody was significantly improved compared to that in the passively suspended car. The satisfactory results led to implementation of the active suspension 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 try a more advanced algorithm to examine whether even better results could be achieved. The choice fell on so-called $H\infty$ control and a quarter-car model was built in MATLAB, which is described in **Paper C**.
**Summary of present work**

$H_\infty$ control offers the possibility to control several signals simultaneously because of the many parameters that can be changed. In particular, this is done by applying weight functions to selected signals in the system. The two control strategies are compared by evaluating how lateral carbody acceleration and relative lateral displacement between carbody and bogie in particular are affected by the applied control force. Simulation results show that $H_\infty$ control generates similar carbody accelerations at the same control force as sky-hook; however, carbody displacement is somewhat lower with $H_\infty$ control.

**Track force evaluation**

When working with active secondary suspension the question of the influence on track forces is usually raised. There is an apprehension that track forces may increase when active secondary suspension is being used. The measured track forces from the on-track tests in 2008 have been evaluated by Interfleet Technology, but have not been mentioned in any of the appended papers. The results are summarised in Table 7-1, comparing the passive suspension with the active lateral secondary suspension (ALS). Overall, the evaluation shows that the ALS bogie does not risk exceeding the derailment criterion $Y/Q$. According to the running safety criterion $\Sigma Y_{2m}$, both bogies generate approximately the same track forces, well under the limit value. Evaluation of the running stability, $\Sigma Y_{100rms}$, shows comforting values below the limit. However, a few minor effects need to be explained.

1) The HOD function of the ALS minimises quasi-static lateral displacements, which will influence the balance of the vertical track forces (decreased force on outer wheel). Therefore, the $Y/Q$ values in narrow curves tend to be somewhat higher with the ALS bogie.

2) Bumpstop contact can be avoided at high cant deficiency curving when the quasi-static lateral displacements are minimised, which, particularly in large-radius curves decreases high peaks of the lateral track forces. This results in lower $Y/Q$ values.

3) The ALS partly decouples the bogie motions and the carbody motions, which may have an effect on bogie stability. Therefore, the stability criterion $\Sigma Y_{100rms}$ shows a higher value in the case with ALS.

However, it should be noted that the chosen evaluation method implies a statistical uncertainty, which may have contributed to the results. Furthermore, track forces for the passive suspension and the ALS concepts are evaluated from different test periods. This implies that different track sections may be evaluated along with variations in track quality and vehicle speed.

The ALS bogie fulfills all requirements in the standard UIC 518 [49] for a maximum operating speed of 250 km/h and a maximum operational cant deficiency of 183 mm. In addition, in September 2008, a new Swedish high-speed record of 303 km/h was set with the ALS bogie integrated in the vehicle. Running stability was provided and the track forces were kept under the limit values.
Table 7-1  Comparison of track forces of the passive and active bogie, respectively

<table>
<thead>
<tr>
<th>Test zone</th>
<th>$Y/Q$ [-]</th>
<th>$\Sigma Y_{2m}$ [kN]</th>
<th>$\Sigma Y_{100rms}$ [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Passive</td>
<td>ALS</td>
<td>Passive</td>
</tr>
<tr>
<td>250 &lt; $R$ &lt; 400 m</td>
<td>0.62</td>
<td>0.68</td>
<td>46</td>
</tr>
<tr>
<td>400 &lt; $R$ &lt; 600 m</td>
<td>0.54</td>
<td>0.65</td>
<td>40</td>
</tr>
<tr>
<td>900 &lt; $R$ &lt; 1 500 m</td>
<td>0.48</td>
<td>0.46</td>
<td>42</td>
</tr>
<tr>
<td>3 000 &lt; $R$ &lt; 5 000 m</td>
<td>0.40</td>
<td>0.28</td>
<td>42</td>
</tr>
<tr>
<td>Straight track</td>
<td>-</td>
<td>-</td>
<td>27</td>
</tr>
<tr>
<td>Limit values</td>
<td>0.8</td>
<td></td>
<td>61</td>
</tr>
</tbody>
</table>
Summary of present work
8 Conclusions and further work

The purpose of this study has been to develop an active lateral secondary suspension concept that offers good ride comfort improvements and enables centring of the carbody above the bogies when negotiating curves at unbalanced speed. Simultaneously, it should be a cost-effective solution for future series production.

By means of multi-body simulations and subsequent on-track tests, the active suspension concept presented in this thesis has proved to significantly improve passenger ride comfort. Moreover, the chosen solution is considered to be cheap enough to be interesting for customers. Particularly the HOD function, which minimises the lateral displacement of the carbody in relation to the bogies in curves with high track plane acceleration, has shown great potential of comfort improvements. Its ability to centre the carbody above the bogies and hence to avoid bumpstop contact enables a wider carbody and improves cross wind stability.

Previous research in this area has shown similar results with promising ride comfort improvements. Nevertheless, those active secondary suspension concepts have rarely reached implementation in service operation. The likely reason for this is related to the relatively high costs for implementing and maintaining an active system. There is always a balance between, on the one hand, satisfactory performance and, on the other hand, acceptable costs.

The active suspension concept presented in this thesis is, however, believed to fulfil these requirements and be a desirable solution for the market. The choice of the relatively cheap electro-hydraulic actuator for this study enables future series production since it, in combination with the chosen control algorithm, offers good performance.

Further work in this research field should focus on the following points:

- The initial study on $H_\infty$ control should be extended to a full-scale vehicle model in order to prepare and tune the concept for future on-track tests.
- Preview data of track geometry can be used in order to receive an even more precise control of the carbody movements in relation to the bogies.
- Up to now the active secondary suspension has been implemented in the lateral direction. Control of the carbody’s vertical vibrations is intended to be studied in the next phase of the present PhD project.
Conclusions and further work
Notations

Latin symbols

\( a^w \) frequency weighted acceleration (m/s^2)
\( a_{\text{rms}} \) \( rms \) of frequency weighted acceleration (m/s^2)
\( a_y \) track plane acceleration (m/s^2)
\( a_{yc} \) carbody plane acceleration (m/s^2)
\( a_z \) vertical acceleration in track-following system (m/s^2)
\( a_{zc} \) vertical acceleration in carbody-following system (m/s^2)
\( A \) system matrix
\( b_0 \) half track base (m)
\( B \) input matrix
\( c \) damping (Ns/m)
\( c_{\text{sky}} \) sky-hook damping (Ns/m)
\( C \) output matrix
\( e \) error signal
\( f \) frequency (Hz)
\( F \) force (N)
\( g \) acceleration of gravity (m/s^2)
\( h_t \) track cant (m)
\( J \) cost function
\( k \) stiffness (N/m)
\( K \) Kalman filter gain matrix
\( K_c \) controller
\( K_p \) coefficient for proportional part
\( K_I \) coefficient for integral part
\( K_D \) coefficient for derivative part
\( m \) mass (kg)
\( N \) disturbance input matrix
\( Q \) vertical track force (N)
**Notations**

- $Q_i$: weighting function or matrix ($i = 1, 2$)
- $r$: reference signal
- $R$: curve radius (m)
- $R_i$: elements of intensity matrix ($i = 1, 2, 12$)
- $t$: time (s)
- $T$: integration time (s)
- $T_e$: torque (Nm)
- $u$: input/control signal
- $v$: velocity (m/s)
- $v_i$: white noise intensity signal ($i = 1, 2$)
- $w$: disturbance signal
- $x$: system state
- $y$: output/measurement signal
- $Y$: lateral track force (N)
- $\Sigma Y$: track shift force (N)
- $z$: output/error signal
- $z_g$: vertical displacement of ground (m)
- $z_m$: vertical displacement of body (m)

**Greek symbols**

- $\varphi_c$: carbody roll angle in track plane (rad, °)
- $\varphi_t$: track cant angle (rad, °)
- $\omega$: angular velocity (rad/s)
- $\Phi$: side force angle (rad, °)

**Abbreviations**

- HOD: Hold-Off-Device
- KTH: Kungliga Tekniska Högskolan (Royal Institute of Technology)
- $LQ$: Linear Quadratic
- $LQG$: Linear Quadratic Gaussian
- $PID$: Proportional-Integral-Derivative
- $rms$: root mean square
Active Lateral Secondary Suspension in a High-Speed Train to Improve Ride Comfort

References


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Active Lateral Secondary Suspension in a High-Speed Train to Improve Ride Comfort


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