



KTH Engineering Sciences



# Active Secondary Suspension in Trains

## A Literature Survey of Concepts and Previous Work

by

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*In memory of Bertil and to a new beginning with Leo.*



## **Preface**

This literature survey is a course part of my doctoral studies concerning active secondary suspension in railway vehicles. The doctoral project is a part of the research and development programme Gröna Tåget (Green Train), financed by the Swedish National Rail Administration (Banverket). This literature survey has been performed at the Royal Institute of Technology (KTH) in Stockholm, Sweden.

Many thanks to my supervisors Sebastian Stichel and Rickard Persson who have contributed with valuable comments and inputs.

Also grateful thanks to Dan Brabie and Dirk Thomas who have provided me with many useful articles in the present subject.

Stockholm, September 2008

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## **Abstract**

With increased railway vehicle speeds, the vehicle's dynamic performance is negatively affected. The suspension of the vehicle has to be modified in order to compensate for the deteriorated dynamic behaviour. However, improvement possibilities by means of passive suspension technology will eventually reach a limit. Therefore, active suspension technology in railway vehicles is considered as an alternative solution for this issue, since it offers better possibilities of improving the vehicle's dynamic performance compared to the conventional passive solution.

Active technology in rail vehicles can be divided into two general categories; one improving running stability and wheelset guidance (mainly controlled through the primary suspension), and another improving passenger ride comfort (controlled through various modifications of the secondary suspension). Also included in the latter suspension concept is tilting technology and lateral carbody centring, i.e. using a so-called Hold-Off-Device. This study concentrates on the secondary suspension concept, concerning ride comfort improvements by means of active suspension.

Most of the studies concerning active secondary suspension concentrate on improving, or at least maintaining, ride comfort despite increased vehicle speed or worse track conditions, which may offer a cost-efficient solution if vehicle speed can be increased or track maintenance costs can be minimized.

An active suspension system consists of actuators, sensors and a specific control law, which generates the force demand for the actuator. The actuator should be able to generate the demanded control force. How well this is done depends on the characteristics of the actuator. There are various types of actuators that can be applied in railway vehicles, such as electro-mechanical, electro-magnetic, hydraulic, servo-pneumatic and rheological (electrical or magnetic). Together with the actuator an appropriate control strategy has to be chosen. One of the most implemented and analyzed during the years is sky-hook damping, but also optimal control, such as LQ/LQG and  $H_\infty$  control have been thoroughly investigated.

Active technology in order to improve ride comfort has been well studied during several decades and shows satisfactory results, but has not yet reached a convincing final breakthrough in service operation. The reason for this is probably related to the relatively high costs for implementing and maintaining active technology, since the work with active suspension always is a balance between, on the one hand, good performance and, on the other hand, acceptable costs.



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# **1 Introduction**

## **1.1 Background**

In the beginning the railway system was a strictly mechanical concept. Over the years, however, it has tended to become more electronically based. Railway vehicles of today are very much dependent on electronics and computer processing. The concept of active technology in rail vehicles has been analyzed theoretically and experimentally since the 1970s, but has not yet made its convincing breakthrough in operational use (except for the tilting train technology, which will be explained in the next chapter), 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 that it is expensive. 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 manages good performance and acceptable costs simultaneously, there is significant potential for future implementation.

In the general competition between different means of transportation the main subject is a fast, safe and comfortable journey. Furthermore, those quantities have to be offered to a reasonable price for the passengers. For railway vehicles active technology enlarges the possibilities of improving the vehicle's dynamic performance compared to what the conventional passive solution has to offer. Improvements of ride comfort, running behaviour and curving ability have, of course, a positive impact on the vehicle's dynamic performance, which, in turn, may enable higher vehicle speeds.

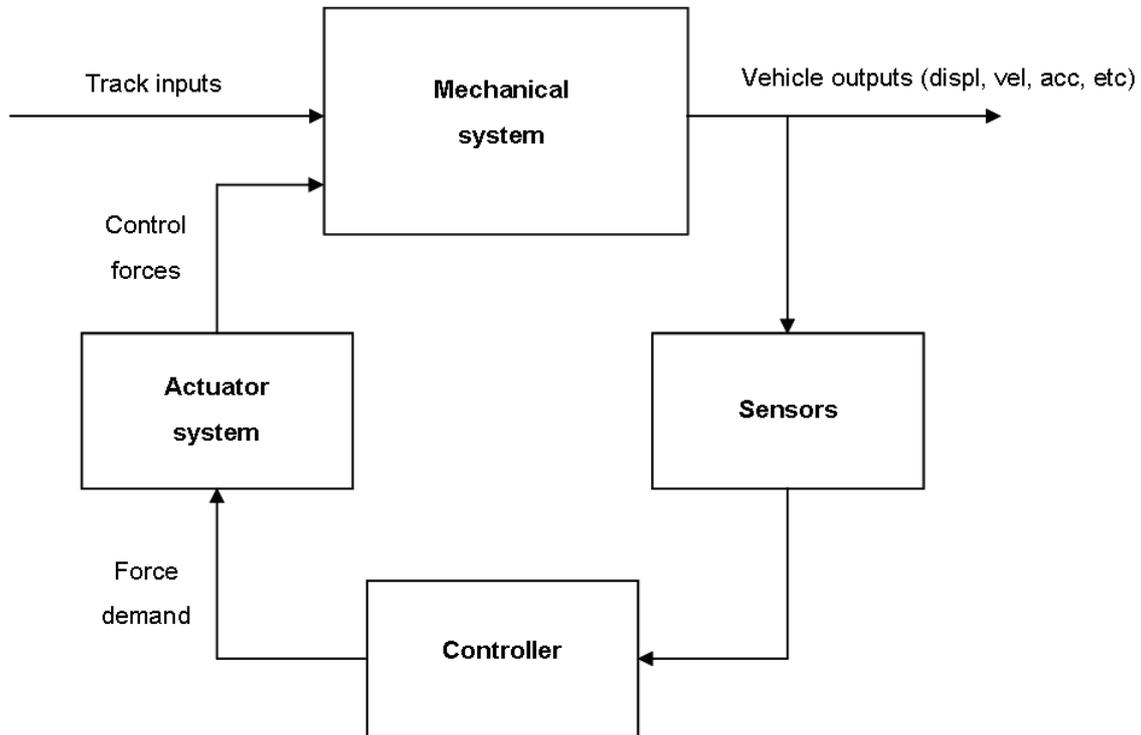
Active technology in rail vehicles can be utilized in order to achieve one or more of the following goals;

- a) improve passenger ride comfort,
- b) maintain good ride comfort although vehicle speed is increased,
- c) maintain good ride comfort although track conditions are worse,
- d) reduce wheel and rail wear by means of improved curve negotiation,
- e) secure running stability at higher vehicle speed.

If the ride comfort is already good, further improvement at unchanged vehicle speed and track conditions is generally not justified due to the high costs of implementing active suspension. However, goals b) and e) allow large possibilities for cost-efficient improvements, since vehicle speed can be increased. Moreover, goals c) and d) have good potential of being worth the investment of active technology, since track maintenance costs can be saved.

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.



*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 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. Different actuator types with their specific characteristics, as well as advantages and disadvantages, will be described in Chapter 3.

## 1.2 Purpose with this work

This study gives an overview of the concepts and theories of active suspension technology in rail vehicles, focusing on the secondary suspension, both in lateral and vertical direction. It will refer to previous studies that have covered the present subject. The areas of tilting technology and active primary suspension will be briefly mentioned but are not further covered by this survey. The suspension concepts are introduced in Chapter 2. Previous extensive surveys that include the different suspension concepts of active technology in rail

vehicles origin from Hedrick and Wormley in 1975 [26], Goodall and Kortüm in 1983 [21], Goodall in 1997 [22] and Bruni, Goodall, Mei and Tsunashima in 2007 [8].

In Chapter 3 the function, advantages and drawbacks of different actuator types will be described. Furthermore, in Chapter 4 different control methods will be presented, that to some extent have been used for active technology in rail vehicles. This will be followed by a chapter that mentions the practical use, i.e. full-scale tests that have been performed or active suspension in commercial service. Finally, conclusions and future trends will be presented in Chapter 6.

## **2 Suspension concepts**

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). The following sections describe the basic ideas behind the different suspension concepts.

### **2.1 Active suspension to improve stability and guidance**

The largest challenge in the area of railway vehicles is to master the contradiction of stability and guidance. A vehicle's running stability at high speed on straight track is strongly correlated to a quite stiff wheelset guidance, especially in the longitudinal direction, often in combination with a stiff yaw damping between carbody and bogie. However, the curving performance is thereby negatively affected, since a high primary suspension stiffness reduces the radial steering ability, which causes larger track shift forces and a higher amount of wheel and rail wear. With actively controlled primary suspension, the stiffness can be adapted to the current situation.

The idea of active primary suspension has been under consideration for many years and has been theoretically and practically developed and improved. However, it has not yet reached a final breakthrough; but still, it offers a potential of great benefits when the technology is fully developed and reliable. One of the reasons why there might be resistance to implement this technology is that active primary suspension is safety critical.

There is a variety of approaches to improve stability and guidance through active technology, which has been categorized and summarized by Goodall, Bruni and Mei [23] and [8]. Five different configurations have been studied and are listed, namely;

- Actuated Solid Wheelset (ASW),
- Actuated Independently Rotating Wheels (AIRW),
- Driven Independently Rotating Wheels (DIRW),
- Directly Steered Wheels (DSW),
- Secondary Yaw Control (SYC).

The concept of ASW was introduced by Shen and Goodall [55] and was first an idea of active traction rods in order to improve curving performance without decreasing the primary suspension stiffness in longitudinal direction, and hence the ability of stable running behaviour at high speed. It developed into the general idea based on control forces applied laterally or longitudinally to a solid wheelset in order to provide steering and/or stabilization.

In contrast to ASW, the concept of AIRW comprises independently rotating wheels, which minimizes the risk of instability at higher speeds. On the other hand, the ability of self-

steering in curves is reduced, since there is no longer a solid connection between the wheels. However, the longitudinal creep forces in curves are near zero, which means that the required actuator forces are much lower than for the ASW concept. Contrariwise, a somewhat more complicated mechanical design is required.

For the third concept, DIRW, active control is achieved by traction motors applied to the independently rotating wheels. The advantage is, as for the AIRW concept, a very good running stability behaviour as well as improved curving performance through active control. On the contrary, this solution is very expensive, since one traction motor is required for each wheel of the vehicle.

The DSW concept is a further simplification of the ASW concept, with two independently rotating wheels instead of a solid axle. The wheels are actively steered in order to achieve radial steering in curves. By means of the independent rotation of the wheels stability is ensured on straight track. The lack of a solid axle enables design of a low-floor carbody, which is particularly favourable for urban transportation, such as trams.

The fifth concept, SYC, differs from the other concepts since it comprises the secondary suspension level. A yaw torque is applied on the bogie by means of actuators between carbody and bogie, basically replacing the traditional passive yaw dampers, in order to maintain stability and improve curving performance. SYC may not be the most effective way to handle the conflict of stability and curving performance compared to the other described concepts, but it is most likely the easiest to implement. In a study by Braghin, Bruni and Resta [6] the SYC concept was implemented with results showing a possibility to raise the critical vehicle speed and to negotiate curves at higher cant deficiency. However, the study requires further investigations to optimize the performance. Further studies in the area of active yaw dampers have been performed by Bruni and Resta [9] and Breuer [7].

In 2001 Streiter, Boller, Riege, Schneider and Himmelstein [57] described the concept of a mechatronic bogie, with active control of both the primary and secondary suspension levels. In the primary suspension the wheelsets were individually controlled by one actuator per axle. The project was a co-operation between Bombardier Transportation and Daimler Chrysler, which led to a prototype of a mechatronic bogie that could be tested on a roller rig. The results were promising, fulfilling all requirements by Deutsche Bahn. Further work with the mechatronic bogie, mainly focusing on actively controlled primary suspension, has been described by Himmelstein in [28]. Tests with a prototype performed on a full-scale roller rig showed good stability properties at high speeds. Moreover, the actively controlled primary suspension of the mechatronic bogie was tested in summer 2007 during on-track tests in Sweden, within the research and development programme Gröna Tåget (Green Train) [32], described further in Section 5.1.

## **2.2 Active suspension to improve ride comfort**

In contrast to control of the primary suspension, which can improve stability and guidance of the wheelsets, active control of the secondary suspension level concerns the passenger comfort. The purpose with active control of the secondary suspension is to provide better isolation of the carbody from excitations transmitted from track irregularities than the passive damping has to offer, hence improve passenger comfort. As mentioned in Section 1.1 the goal with actively controlled secondary suspension is to

- a) improve the passenger comfort under the same vehicle speed and track conditions,
- b) maintain good passenger comfort despite increased vehicle speed or
- c) maintain good passenger comfort despite worse track conditions.

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 and is described separately in the following section.

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 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. 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.

Beside the traditional secondary suspension between bogies and the carbody, there is an alternative approach of damping between the carbodies, so-called inter-vehicle damping. This passive concept has been in use during several years in many countries, e.g. Britain, France and Japan, according to a study performed by Pratt and Goodall [48]. Research investigating the benefits of replacing the passive inter-vehicle dampers with active ones has been ongoing since the mid 1990s. The greatest advantage with this solution is that fewer actuators are needed, which lowers the total vehicle mass and the overall cost. A study from 1994 by Pratt and Goodall [47] comes to the conclusion that active inter-vehicle suspension can reduce either the vertical or the pitch motions of the vehicle, depending on the chosen control strategy. However, compared with traditional active damping between bogies and carbody no significant comfort improvements can be gained. Further studies in 1997 [48] comparing optimized passive and active inter-vehicle suspensions, respectively, show that the active solution does not offer sufficient ride comfort improvements.

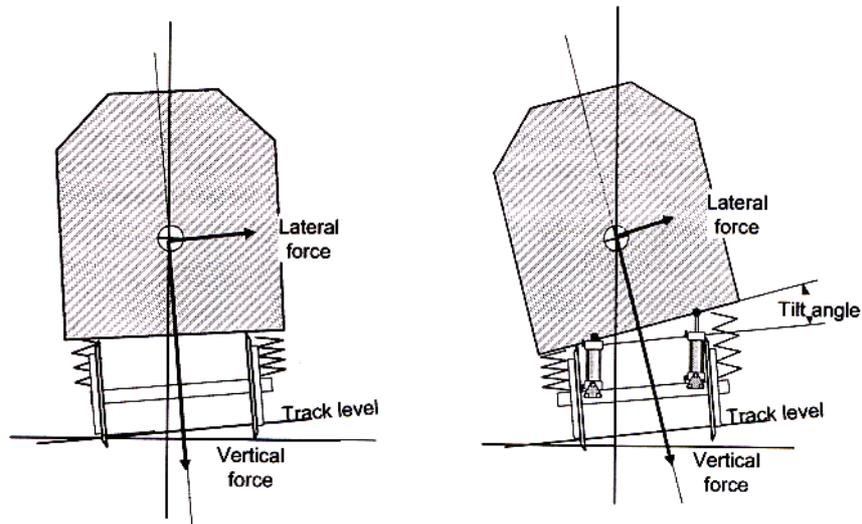
Schandl, Lugner, Benatzky, Kozek and Stribersky [54] have designed and simulated a light-weight vehicle model, typically a tram or metro. Less weight of a vehicle carbody reduces the structural stiffness and hence has negative impact on the ride comfort, which the researchers have tried to compensate for by using an active vibration reduction system, instead of traditional active suspension. Twelve actuators and twelve sensors were mounted on the carbody where structural vibrations could be measured, and a bending moment could be actively applied in order to suppress the first three eigenmodes. The simulations showed a significant reduction of the structural vibration level of the carbody.

## **Tilting**

Tilting is the part of active technology that has been the most successful in the area of railway vehicles. Active tilting technology was introduced as early as 1957 by SNCF (La Société Nationale des Chemins de Fer) in France, according to a study performed by Persson [43]. 1972 the first actively tilted trains were taken into commercial service by DB (Deutsche Bahn) in Germany. The real breakthrough came around 1990 when a series production of tilting trains started in Sweden and Italy. Carbody tilting is now a well established railway technology.

In curves there is a centrifugal force making the passengers feel an outward acceleration, which has a negative impact on the ride comfort. With increased vehicle speed the centrifugal force in curves is increased. However, with tilting technology the vehicle carbody is tilted inwards (a roll motion) and the centrifugal force is decreased, see Figure 2-1. Hence, tilting is used to reduce, or at least maintain, the centrifugal force, or acceleration, felt by the passengers in curves, although the vehicle speed is increased.

Another advantage with tilting compensating for the increased acceleration created by higher speed is that no increase of track cant is needed. Normally, higher vehicle speed is followed by increase of track cant in curves in order to keep the acceleration, and hence ride comfort, at an acceptable level. Increased track cant is not favourable for “mixed” traffic conditions, i.e. high-speed trains, regional trains and heavy freight wagons running on the same track, which is often the case in Sweden.



**Figure 2-1** *The lateral force is decreased due to tilting of the carbody (right) [43].*

There are several possibilities that can be used in order to detect curves and activate the tilt control. Lateral acceleration of the bogie is the most common reference source. In addition, roll and yaw velocity of the bogie can be used. An alternative is to establish the current train position by GPS (Global Positioning System) and then utilize stored track information onboard the train, such as curvature and cant, in order to enable control of carbody tilt by accurate curve data, according to Sasaki [53].

A disadvantage with train tilting is that sensible passengers can be subjected to motion sickness, above all caused by the low-frequency roll motion. Research to survey this issue and to find a possible solution has been and is still being performed by, among others, Förstberg [15] and Persson [44].

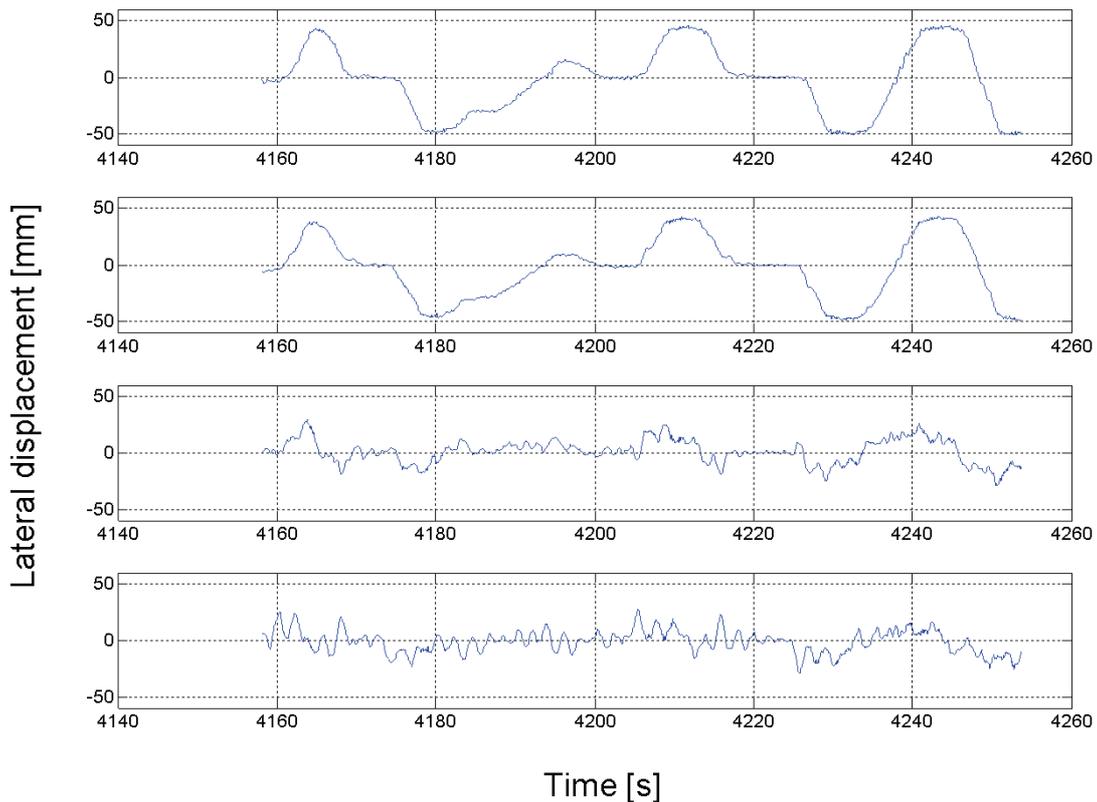
Furthermore, a comprehensive literature survey has been performed 2007 by Persson [43], describing the concept of tilting technology, the development during the years and the present situation.

### **Hold-Off-Device**

When travelling in curves at high speed, causing large lateral accelerations, a quasi-static displacement between carbody and bogies arises. The carbody moves outwards in the curve and there is a risk of hitting the bumpstops, which radically reduces the ride comfort. If the quasi-static displacement can be avoided, not only can good ride comfort be maintained, but a wider carbody profile 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, side wind stability can be improved.

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 denoted *low-bandwidth control*, since it detects the low frequencies of deterministic track inputs, i.e. curves, in order to minimize the lateral displacement of the carbody in relation to the bogies. As well as tilting technology,

the concept of low-bandwidth control is part of the secondary suspension level, but because of its prominent role of improving passenger comfort it deserves to be treated separately. An example of how the lateral displacement between carbody and bogie is minimized by the HOD function can be seen in Figure 2-2. The curves show measurement results from on-track tests performed in summer 2007 in Sweden using a two-car EMU (Electrical Motor Unit). The two uppermost plots exemplify the lateral carbody displacement in relation to two passive bogies, whereas the lowest two plots represent carbody displacement relative to two active bogies including the HOD concept. Hence, contact with the bumpstops, in this case after 46 mm displacement, can be avoided with the HOD function.



**Figure 2-2** *Lateral carbody displacement in relation to passive bogies (two uppermost plots) and active bogies (two lowest plots), respectively. The HOD concept is implemented in the active bogies.*

The HOD concept was introduced in the early 1990s by Allen [2] 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 [45]. Further research has been performed by Stribersky, Steidl, Müller and Rath [58], and Stribersky, Müller and Rath [59] where the benefit of active lateral positioning has been proven through comparisons between simulated and measured results of passive and active solutions, respectively. These studies also included tilting technology.

Furthermore, the mentioned low-bandwidth control can also be applied in the vertical direction, but it then reacts on low-frequency changes in the vertical curvature. This is

mostly handled by air suspensions between bogies and carbody and is actually not included in the HOD concept.

## **2.3 Degree of control**

The conventional, passive suspension has a rather simple design and is cost-effective compared to active suspension. On the other hand, the possibilities of further improvements in, for example, passenger comfort are restricted. 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 [34].

The 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 [37].

Between the passive and the fully-active solutions there is the *semi-active* suspension system. 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 2-3. The actuator force depends on the relative damper velocity, i.e. the velocity difference between the two bodies where the actuator is situated, and 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. This is also explained in Table 2-1, where the actual damping force equals the actuator force when the actuator force and relative damper velocity have the same sign. With opposite signs a zero-force is applied, which is described by Roth and Lizell [49]. Despite this drawback, the semi-active suspension can still work in the passive mode if failure of the control system occurs.

The concepts of semi- and fully-active suspensions are schematically shown in Figure 2-4.

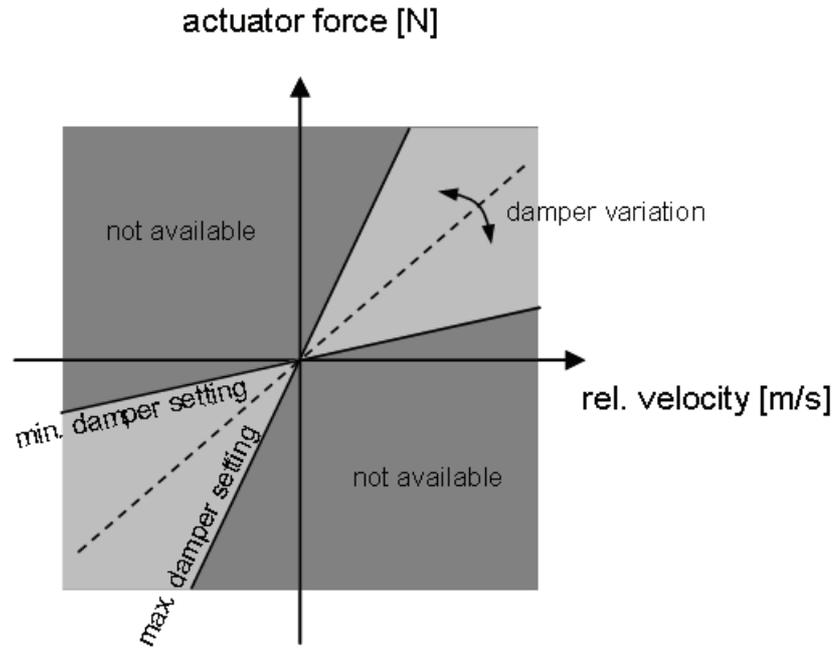
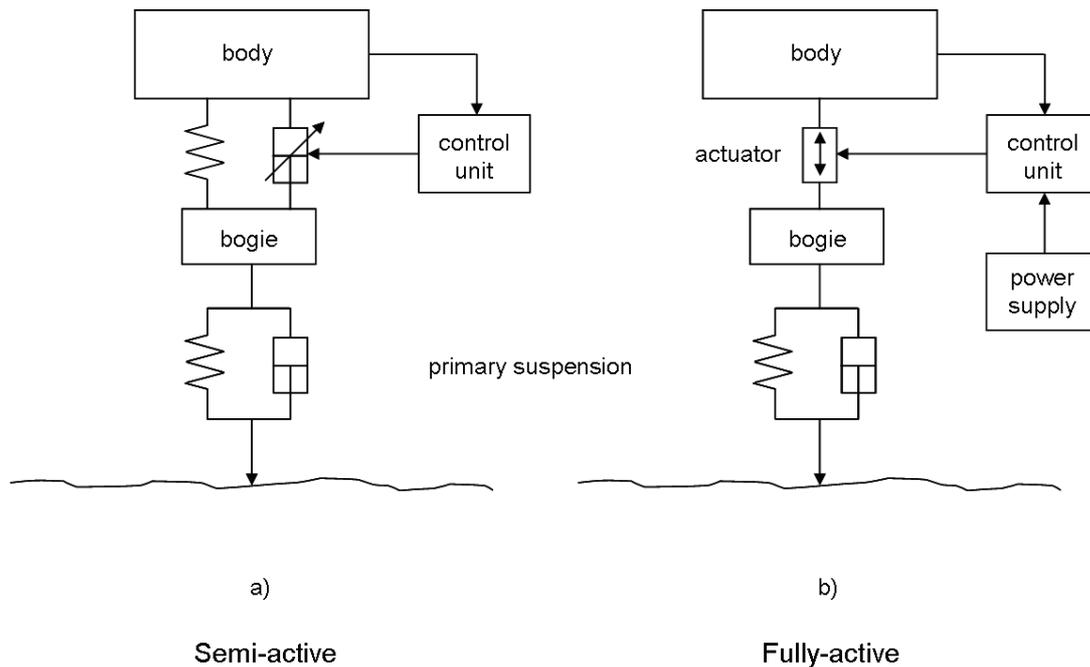


Figure 2-3 Force-velocity diagram for semi-active damping, Goodall and Mei [24].

Table 2-1 The influence of directions of actuator force and relative damper velocity on the actual damping force, in a semi-active system.

Actuator force	Direction of rel. velocity	Actual damping force
$F > 0$	$v > 0$	$F_{actual} = F$
$F < 0$	$v < 0$	$F_{actual} = F$
$F > 0$	$v < 0$	$F_{actual} = 0$
$F < 0$	$v > 0$	$F_{actual} = 0$



**Figure 2-4** Concepts of semi-active and fully-active suspension control, Goodall and Mei [24].

Another suspension system is the *semi-passive* suspension, which is not so often mentioned in the literature. However, it is described by Hedrick [27] as a trade-off between the semi-active and the fully-active concepts, where a discrete control strategy switches between passive and active modes. The active mode is used primarily to minimize transients due to, for example, curve transitions. The damping characteristics are varied according to a variable that is not influenced by the dynamic system being controlled, e.g. vehicle speed, which is shortly mentioned by Goodall and Mei [24].

Semi-active suspension can be realized through different types of devices, which either dissipate or conserve energy. Some of them, such as variable damper, variable force transformer and variable spring stiffness, are described by Valášek and Kortüm [65].

A comparison of fully-active and semi-active suspension concepts has been performed by Ballo [4], however, applied to a quarter-car goods road vehicle. When analyzing the *rms* value of the sprung mass acceleration the semi-active suspension concept offers as good reduction as the fully-active, additionally, to lower power consumption. Contrariwise, the *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 was in need of improved ride comfort when aiming at commercial operation speed of 300 km/h. After optimizing the 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 latter 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 [40], and is further discussed in Section 5.2.

Tanifuji, Koizumi and Shimamune [62] have summarized the Japanese work performed during 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 rail 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 (previously mentioned in Section 2.2 as Hold-Off-Device), in order to avoid bumpstop contact in curves at high track plane acceleration. The applications mentioned in this Japanese study that have led to either full-scale tests or even to service operation are discussed in Chapter 5.

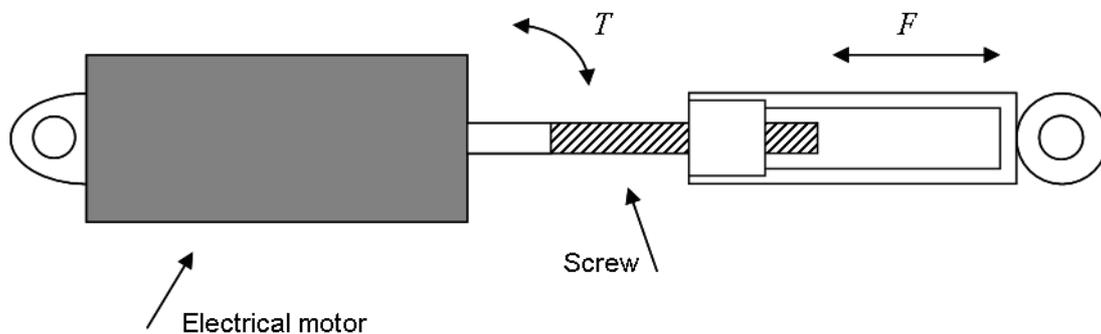
### 3 Actuator types

Several actuator types have been studied in the area of railway technology during 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 [37] and Thomas [63].

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 [37], see Figure 3-1. Electro-mechanical actuators are in general less compact than other actuator types. In [22] it is stated that they can encounter problems with reliability and life of mechanical components.



*Figure 3-1 Principal function of an electro-mechanical actuator [37].*

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 [46]. Furthermore, experimental research in the late 1990s in France using electro-mechanical actuators has been described by Gautier, Quetin and Vincent [18]. 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 analyzed by Kjellqvist [37] and Kjellqvist, Sadarangani and Östlund [36], making a suggestion 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 [13] 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 the 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 [45]. 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 overcome with proper force feedback, according to Goodall, Pearson and Pratt [25], and Pollard and Simons [46].

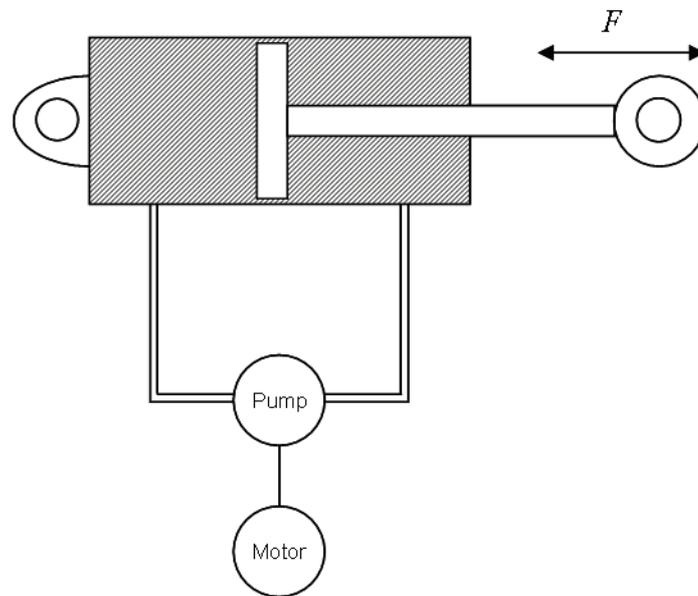
## **3.3 Hydraulic**

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 concepts. Accordingly, in the literature the use of these actuators 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, give 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 [39].

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 [13]. Their cost-effectiveness makes them favourable to be implemented in vehicles for full-scale tests. An experimental analysis was performed by Shimamune and Tanifuji [56], who chose a hydraulic actuator before a pneumatic actuator because of its ability to control up to a 10 Hz frequency range, 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 [25].



**Figure 3-2** *Principal function of an electro-hydraulic actuator.*

A hydraulic actuator was used for the Hold-Off-Device system [2], described in Section 2.2, 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.

### 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 [46]. 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). The elements of the actuator are relatively cheap and there is no liquid that can leak. However, due to the large air compressibility the controllable frequency bandwidth is restricted to 2-3 Hz, and hence the efficiency of the actuator is limited.

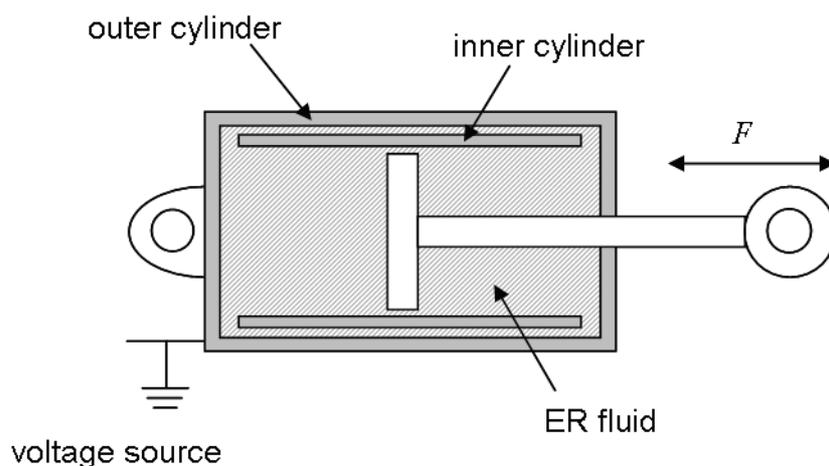
In an investigation performed in the mid 1990s by Sasaki, Kamoshita and Enomoto [52] servo-pneumatic actuators were tested on a roller rig, in order to reduce vibrations in vertical, lateral and roll modes. Up to 50 % reduction of these particular modes of vibration could be shown with the active system. Another Japanese study was performed at approximately the same time by Hirata, Koizumi and Takahashi [30], 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 [40]. Nevertheless, lateral acceleration in the carbody could be reduced although the 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. Depending on the strength of the field applied to the actuator, the viscosity of the fluid is varied (and hence the damping characteristics of the actuator). 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 [66]. 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 [12], and Gao and Yang [17]. The MR actuator has been analyzed in an experimental test rig by means of semi-active control in [17]. It was concluded that the MR actuator could generate damping forces in a very broad range under the influence of a magnetic field.



*Figure 3-3 Principal function of an electro-rheological actuator.*

## 4 Control strategies

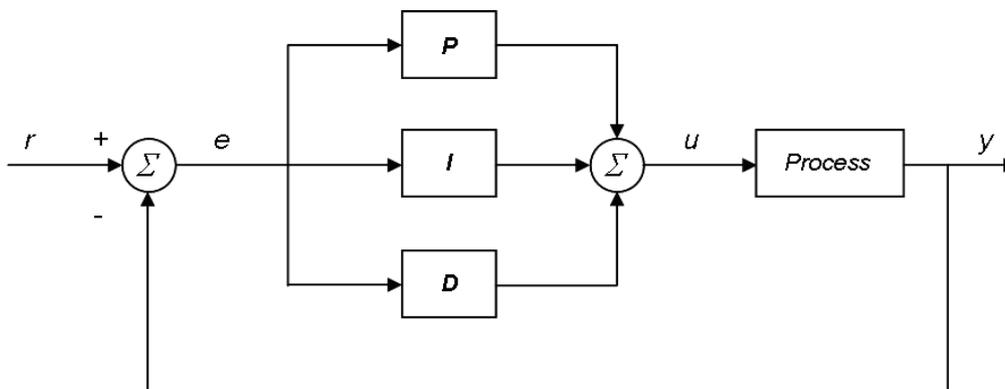
In order to enable steering and control of the actuators in a favourable way an appropriate control algorithm is needed. Several control strategies have been studied and implemented in the area of active technology within rail vehicles. The most common ones are presented in this section.

### 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  $y$ ;  $e(t) = r(t) - y(t)$ , see Figure 4-1. The PID controller is described as

$$u(t) = K_p e(t) + K_I \int_0^t e(s) ds + K_D \frac{d}{dt} e(t) , \quad (4-1)$$

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, which is described by Glad and Ljung [19].



*Figure 4-1 Block diagram for a PID controller.*

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

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, see Figure 4-2. For the conventional passive system, a) in Figure 4-2, Newton's second law can be expressed as [37]

$$m\ddot{z}_m = (\dot{z}_g - \dot{z}_m)c + (z_g - z_m)k \quad , \quad (4-2)$$

which can be rewritten as the transfer function

$$G_{passive} = \frac{cs + k}{ms^2 + cs + k} \quad . \quad (4-3)$$

The corresponding equations for the sky-hook suspension model, b) in Figure 4-2, can be expressed as

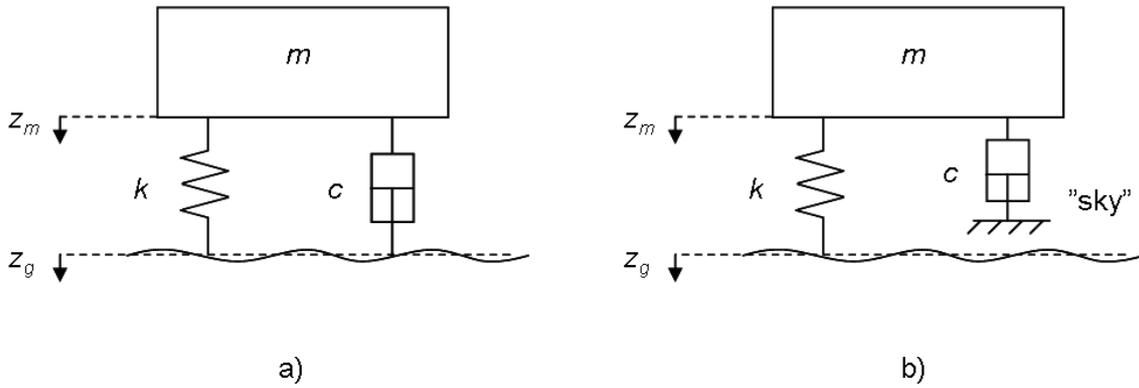
$$m\ddot{z}_m = -\dot{z}_m c_{sky} + (z_g - z_m)k \quad (4-4)$$

and

$$G_{sky-hook} = \frac{k}{ms^2 + c_{sky}s + k} \quad . \quad (4-5)$$

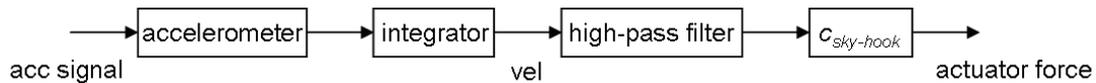
Since the damping term of the ground input (typically high-frequency track irregularities) is not included in the sky-hook model, isolation of these frequencies is better than in the passive suspension model. The concept is also called *absolute velocity damping*, since the desired actuator force is created from the absolute velocity of the body in combination with the current sky-hook damping coefficient. Sky-hook damping gives significant improvement of passenger ride comfort on straight track. However, in curves at high vehicle speed there is a

risk of large displacement between carbody and bogies, leading to deflection of the secondary suspension. This disadvantage can, however, be counteracted by combining sky-hook control with low-bandwidth control, aiming at centring the carbody above the bogies, described in Section 2.2 [22]. The suspension model is here described in the vertical direction, but is naturally also valid for the lateral direction.



**Figure 4-2** a) Conventional suspension model. b) Sky-hook suspension model.

In practice sky-hook damping is usually implemented as shown in Figure 4-3. In fact, the required absolute velocity signal is normally an integrated acceleration signal measured by a sensor/accelerometer on the carbody. Furthermore, the velocity signal is high-pass filtered and multiplied by the sky-hook damping coefficient in order to generate the demanded actuator force, which is described by Goodall and Mei [24].



**Figure 4-3** Practical implementation of sky-hook damping.

The strategy of sky-hook damping was first introduced by Karnopp in the late 1970s and a comprehensive description was published in 1983 [35]. Thereafter, sky-hook damping has been thoroughly investigated and analysed by various researchers throughout the years. Stribersky, Kienberger, Wagner and Müller [60] have performed simulations and showed that sky-hook damping significantly reduces resonance peaks and *rms* (root mean square) acceleration, thus improving ride comfort, in both vertical and lateral direction. 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 [49] 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 optimize the trade-off between enhanced comfort and suspension deflection during curving. Nevertheless, acceptable results can be achieved by optimizing the filtering of the

absolute velocity signal. Li and Goodall [38] have theoretically analyzed 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 the ride quality by nearly 23 %, while keeping the 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 as for the passive case.

However, Hohenbichler and Six [31] analyzed 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 these special track conditions, compared to a passive case, the sky-hook damping offers no more than 10 % comfort improvement.

Baier, Hohenbichler, Six and Abel [3] 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 are 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 compensate only for the track irregularities and not the deterministic track curvature.

In a study by Suda, Nakadai and Nakano [61] an experimental test rig was built up in order to analyze a hybrid suspension system using sky-hook damping. It is a method of active vibration control by means of regenerated vibration energy, which implies that less power is consumed and no external energy is required. Compared to a passive and a semi-active suspension system the hybrid suspension system provided better isolation performance.

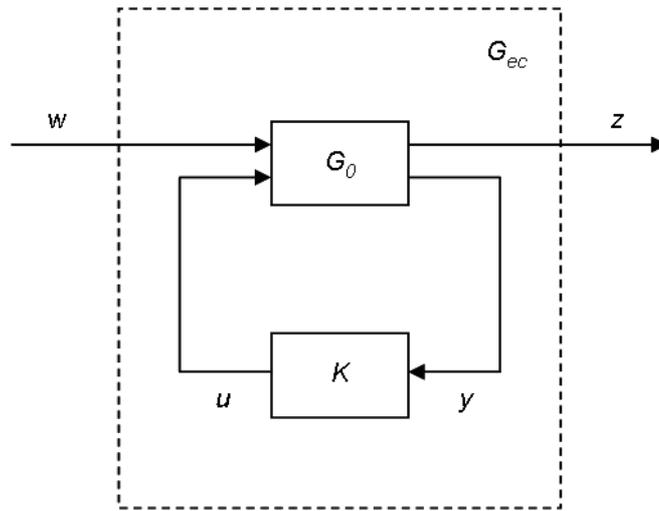
### 4.3 $H_\infty$ control

Advanced control methodology is concerned with finding a controller for the open-loop system, such that the closed-loop system has good performance, stability and robustness. A typical configuration of a control system can be seen in Figure 4-4, where  $K$  is the controller that together with the open-loop system  $G_0$  constitute the closed-loop system  $G_{ec}$  (a so-called “plant” in terms of control theory).  $G_{ec}$  is the transfer matrix from the external disturbance vector  $w$  (including the reference signal) to the error signal vector  $z$ . Moreover, the measurement vector  $y$  is used in  $K$  to calculate the control input vector  $u$ . In order to achieve secured stability and robustness of the system the error signal  $z$  should be minimized. The relations of the signals in the system can be described as

$$\begin{bmatrix} z \\ y \end{bmatrix} = G_0(s) \begin{bmatrix} w \\ u \end{bmatrix} \quad (4-6)$$

$$u = K(s) y \quad (4-7)$$

$$z = G_{ec}(G_0, K) w \quad (4-8)$$



**Figure 4-4** Typical configuration of a control system.

$H_\infty$  control design is an advanced control method that deals with finding a controller  $K$  that solves an optimization control problem. The optimization problem could be expressed as

$$\|G_{ec}\|_\infty = \max_\omega \bar{\sigma}(G_{ec}(i\omega)) < \gamma, \quad (4-9)$$

where  $\gamma$  is a chosen boundary criterion and  $G_{ec}$  is the transfer matrix that describes the closed-loop system. The idea is to find the controller  $K$  that minimizes the so-called  $H_\infty$  norm, i.e. the largest singular value  $\sigma$  of the plant  $G_{ec}$  at a certain frequency  $\omega$ . 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 by Glad and Ljung [20]. The advantage with  $H_\infty$  is its robust stability and offering of good system performance. A drawback is that the  $H_\infty$  control model tends to reach a rather high order number, since the order number of the weight functions is included. Control models of high order number are more complex; therefore, model reduction is preferable.  $H_\infty$  control design has throughout the years been investigated in the area of active vehicle suspensions, but not specifically bounded to rail vehicles.

Simulations and experimental tests were performed in the mid 1990s in Japan by Hirata and Takahashi [29], and Hirata, Koizumi and Takahashi [30], controlling the lateral, roll and yaw motions of a railway vehicle through  $H_\infty$  control theory. It was shown with the experimental test vehicle that the controlled motions could be significantly reduced, leading to damping of the low-frequency vibrations caused by suspension resonance.

In the late 1990s theoretical and experimental tests were performed in China by Zeng, Zhang, Dai, Wu and Shen [67], implementing  $H_\infty$  control on the secondary lateral suspension in order to improve stability and dynamic behaviour of a high-speed railway carbody. Simulations showed more promising results with the active suspension compared to the passive case, and hence initial tests on a roller rig were carried out.

Fukao, Yamawaki and Adachi [14] have investigated active suspension in a simulated railway vehicle model, using a so-called back-stepping technique, which principally combines a known non-adaptive  $H_\infty$  control with an unknown adaptive control. The desired ride quality could be attained with this approach, despite uncertainties of the actuator's parameters.

Abdellahi, Mehdi and M'Saad [1] used a quarter-car vehicle model, trying to minimize a controller with  $H_2$  and  $H_\infty$  control in combination, applying a so-called *LMI* method (Linear Matrix Inequality). The  $H_2$  norm is basically a reduction of all singular values of the plant  $G_{ec}$  for all frequencies, whereas the  $H_\infty$  norm is a reduction of the largest singular value at a certain frequency, as described earlier. The *LMI* method is a powerful tool to approach control problems that appear hard to solve analytically, as described by Gahinet, Nemirovski, Laub and Chilali [16]. The simulations were compared with a pure  $H_\infty$  controller, however, with somewhat ambiguous results.

Studies have been performed by Sammier, Sename and Dugard [50] [51], where the influence of  $H_\infty$  on the vehicle dynamic behaviour has been analyzed, using a half-car and a quarter-car suspension model, respectively. The ride comfort has efficiently been improved, but on the other hand, the control method is complex and because of the rather high amount of design parameters it requires good knowledge from the control design engineer.

## 4.4 LQ/LQG control

Another control theory that is concerned with optimization 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 minimized is called an LQ problem. If normally distributed (Gaussian) disturbances are considered the control theory is extended to LQG [20].

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

$$\begin{cases} \dot{x}(t) = Ax(t) + Bu(t) + Nv_1 \\ y(t) = Cx(t) + v_2 \end{cases}, \quad (4-10)$$

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 signals with the intensity  $\begin{bmatrix} R_1 & R_{12} \\ R_{12}^T & R_2 \end{bmatrix}$ .  $A$  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 minimized is described as the sum of the quadratic norm of the control error  $e$  and the control signal  $u$ , respectively,

$$J = \min (\|e\|_{Q_1}^2 + \|u\|_{Q_2}^2) = \min \int e^T(t) Q_1 e(t) + u^T(t) Q_2 u(t) dt , \quad (4-11)$$

where  $Q_1$  and  $Q_2$  are weighting functions or matrices. The optimal linear feedback control law that minimizes the cost function  $J$  is given by

$$u(t) = -L\hat{x}(t) \quad (4-12a)$$

$$\dot{\hat{x}}(t) = A\hat{x}(t) + Bu(t) + K(y(t) - C\hat{x}(t)) , \quad (4-12b)$$

where Equation (4-12b) 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

$$K = (PC^T + NR_{12})R_2^{-1} , \quad (4-13)$$

where  $P$  is the solution to the Riccati matrix equation

$$AP + PA^T - (PC^T + NR_{12})R_2^{-1}(PC^T + NR_{12})T + NR_1N^T = 0 . \quad (4-14)$$

$L$  in Equation (4-12a) is given by

$$L = Q_2^{-1}B^T S , \quad (4-15)$$

where  $S$  is the solution to the Riccati matrix equation

$$A^T S + SA + Q_1 - SBQ_2^{-1}B^T S = 0 . \quad (4-16)$$

A study that assessed the LQG control law has already been mentioned in Section 2.2 [47]. 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 in the cost function of the LQG theory is designed.

Another already mentioned study by Pratt and Goodall [48] 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 the 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 improvement in ride comfort compared to passive inter-vehicle damping, but however, offers further development possibilities.

Shimamune and Tanifuji [56] 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 [64] 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.

Chantranuwathana and Peng [11] have simulated an electro-hydraulic actuator in a quarter-car vehicle implementing so-called adaptive robust control (ARC). The ARC controller shows good performance compared to a PID controller. However, the authors are not convinced of the advantages of an LQG controller, since it performs poorly in comparison to the analyzed ARC controller.

## **5 Practical implementation**

A large amount of theoretical studies within the field of active secondary suspension have been mentioned in the present work. The following sections summarize the ones that have not only concentrated on simulation work, but have taken a further step into practical implementation. On the one hand, there are full-scale tests, either in laboratory environment, e.g. on roller rigs, or on track. On the other hand, some of the studies have resulted in active secondary suspension in operational service.

### **5.1 Full-scale tests**

One of the earliest practical implementation of active secondary suspension found in literature was made by British Rail in the UK in the late 1970s and the early 1980s [46]. Fully-active suspension in vertical and pitch modes was investigated by means of electro-magnetic and electro-mechanic actuators. Moreover, servo-hydraulic actuators in parallel with the air-springs at each end of the test vehicle, as well as electro-mechanic actuators, were used in order to control lateral and yaw modes. Ride comfort improvements in the range from 30 to 50 % could be achieved. The study summarizes the potential of active suspension, as well as the identified problems that have to be tackled. The main obstacle is the overall cost of the active system, i.e. initial implementation cost and further maintenance cost. The system must offer high reliability in order to be considered cost-efficient.

In the mid 1990s in Japan a study aiming at improving ride comfort by means of fully-active lateral and vertical secondary suspensions was initiated. Experiments on a roller rig were performed using a test train with one carbody equipped with servo-pneumatic actuators and the other with hydraulic actuators. Vertical, lateral and roll modes of carbody vibrations could be reduced by 50-70 % [52]. It is not known whether the experiments led to further development of active suspension.

Another Japanese study performed at approximately the same time used an experimental rail vehicle to develop new concepts for Shinkansen trains [29] [30]. Fully-active pneumatic actuators for secondary lateral suspension were implemented, separating lateral and yaw modes.  $H_\infty$  control theory was applied, resulting in significantly reduced lateral, roll and particularly yaw accelerations.

Furthermore, in the mid 1990s in the UK the potential of semi-active control of the lateral secondary suspension using an electro-hydraulic exciter system in a laboratory rig was studied by GEC Alsthom. The results from these experiments have been summarized by O'Neill and Wale [41], showing an improvement in ride comfort of 25 %.

In the mid and late 1990s experimental studies on active technology were performed by Siemens in Austria [58] [59] [60]. In addition to a tilt system, fully-active control was applied to the lateral secondary suspension of a test vehicle, including yaw mode, using servo-pneumatic actuators. Integrated in the lateral suspension concept was the aim to achieve lateral positioning control in order to avoid bumpstop contact, in this study mentioned as a

Hold-Off-Device function. Furthermore, semi-active control of vertical and pitch modes in the secondary suspension was implemented by hydraulic actuators. The field tests showed overall acceptable vehicle performance, with up to 15 % ride comfort improvements.

In the late 1990s in France a test bench was used to qualify the performance of an active suspension system [18]. An electro-mechanical actuator was chosen and could together with LQ control algorithm show ride comfort improvements in the range from 20 to 50 %, in both lateral and vertical direction. The good results from the test bench were promising for further tests with a prototype vehicle.

Bombardier in Sweden is evaluating a new high-speed train concept suitable for Nordic conditions (particularly regarding track geometry and climate), within the research and development programme Gröna Tåget (Green Train) [32]. The programme includes development of active secondary suspension to improve passenger ride comfort. This specific part of the programme, regarding active suspension, lies within the author's doctoral studies. The aim is to find a solution with good performance, however at a reasonable cost that allows future series production. Fully-active lateral secondary suspension has been implemented, including the HOD function. On-track tests were performed during summer 2007 with a modified Regina train. The test results were promising and the development will continue with on-track tests during summer 2008.

## **5.2 Active secondary suspension in service operation**

In the early 1990s the tilting train X2000 was developed by Adtranz (today Bombardier) and put into traffic 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 [49]. A combination of hydraulic and electro-magnetic actuator was used together with sky-hook damping control. Ride comfort improvements of 30-40 % could be achieved. However, the active lateral secondary suspension stayed on the experimental stage, whereas the tilt control is still in operational use today (2008).

In Italy the Fiat Pendolino started using the lateral centring (HOD) function in combination with tilt technology in service operation in the 1990s, by means of pneumatic actuators. However, during the latest years lateral centring of the carbody has been achieved only by tilting below the secondary suspension, without a particular HOD device. Spanish and British Pendolinos still use the fully-active HOD function in combination with the tilt system, though, as described in [8] and by Casini, Piro and Mancini [10].

In Japan, JR West implemented active suspension in service operation in 1997 on a Shinkansen Series 500 after experimental investigations of fully-active and semi-active secondary suspensions. Satisfactory improvements in ride quality could be shown for both suspension systems. The decision was made that the requirements were sufficiently fulfilled with the semi-active suspension system. The semi-active suspension is achieved by pneumatic actuators using a sky-hook damping control [40]. The ride comfort is equal, or even slightly better, at the speed of 300 km/h compared to a conventional train running at 270 km/h [62].

Moreover, semi-active suspension using sky-hook control theory was tested by JR Central on the Shinkansen Series 300. The semi-active suspension system was later installed in Shinkansen Series 700, which started commercial operation in 1999 [62].

Furthermore, Shinkansen trains Series E2 and E3 have been in service for JR East since 2002. This is the first operational fully-active lateral suspension, using pneumatic actuators in parallel with secondary suspension dampers together with  $H_\infty$  control. Improvements of yaw and roll vibrations of the carbody have been achieved [8]. Fully-active suspension has been applied to the end cars and the green cars (first-class passenger compartment), whereas semi-active suspension has been applied to all other cars of the vehicle [42]. Moreover, this concept has been further developed for the Shinkansen Fastech 360 S and Z by JR East, where higher bandwidth electro-magnetic actuators have been used, described by Hughes [33].

The latest contribution with an actively suspended train-set to the Japanese Shinkansen fleet is the Series N700, which is a further development of the Series 700. After two years of test runs performed by JR Central and JR West the Series N700 has been put into commercial operation in 2007. Semi-active suspension is used in all cars of the vehicle in combination with tilting technology [42].

## **6 Summary and discussion**

Traditional passive suspension has reached a limit when it comes to improving running stability, curve negotiation and ride comfort, along with the requirements on higher vehicle speeds. Therefore, active suspension has been considered during the last decades as an option to take care of these issues. This study has given an overview of different active suspension concepts and research performed in this area during the years, mainly focusing on the secondary suspension to improve passenger ride comfort.

Active technology applied to the secondary suspension level to improve ride comfort has in this work been divided into three parts; a) lateral and/or vertical damping between carbody and bogie to minimize accelerations transferred from track irregularities up to the carbody, b) tilting technology, to reduce accelerations felt by passengers in curves at high speed in combination with high cant deficiency and c) the concept of Hold-Off-Device (HOD). The objective of the HOD concept is to minimize the lateral displacement of the carbody in relation to the bogie in curves with high track plane acceleration in order to avoid bumpstop contact. If achieved, this can lead to possibilities of designing a wider carbody profile and improving side wind stability.

Different types of actuators have been investigated and applied in railway technology during the years in order to improve ride comfort by means of active technology. The most common ones are electro-mechanical, electro-magnetic, hydraulic, servo-pneumatic and rheological (electrical or magnetic) actuators. Furthermore, in combination with the actuator an appropriate control strategy has to be chosen. One of the most implemented and analyzed control strategies is sky-hook damping. Moreover, optimal control, such as LQ/LQG and  $H_\infty$  control have been thoroughly investigated.

Many studies have investigated characteristics and performance of active technology by means of rail vehicle simulation models. Some of the research considered in this study has even led to experimental analyses on roller rigs, test tracks or as far as to commercial operation. But still there seems to remain some work before active secondary suspension to improve ride comfort reaches its final break-through. The technology has been well studied during several decades and shows promising results; but why is it not used in service operation more often? The probable answer to that question is related to the relatively high costs for implementing and maintaining active technology. There is always a balance between, on the one hand, good performance and, on the other hand, acceptable costs.

Furthermore, the use of active suspension in rail vehicles may not be required for all kinds of services. For example, where new high-speed lines are built, with good quality and condition of the tracks, optimized passive rail vehicles are most likely sufficient. However, it is not always possible to build new lines. Instead, the vehicles have to be adapted to already existing tracks, for example as is the case in Sweden. If the track suffers from poor condition and unfavourable geometry, active technology may be a probable option and solution to improve, or at least maintain ride comfort although vehicle speed is increased. But still, it comes down to a matter of trade-off between performance standard and cost.



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## **Appendix A - Notations**

### **Latin symbols**

$A$	system matrix
$B$	input matrix
$c$	damping (Ns/m)
$C$	output matrix
$e(t)$	error signal
$F$	force (N)
$J$	cost function
$k$	stiffness (N/m)
$K$	Kalman filter gain matrix
$K_P$	coefficient for proportional part
$K_I$	coefficient for integral part
$K_D$	coefficient for derivative part
$m$	mass (kg)
$N$	disturbance input matrix
$Q$	weighting function or matrix
$r(t)$	reference signal
$R$	intensity (W/m <sup>2</sup> )
$T$	torque (Nm)
$u(t)$	input signal
$v$	white noise signal
$w$	disturbance vector
$x$	system state vector
$\hat{x}$	estimated state space vector
$y(t)$	output signal
$z$	output vector
$z_g$	vertical displacement (m)
$z_m$	vertical displacement (m)

### **Greek symbols**

$\gamma$	boundary criterion
$\omega$	angular frequency (1/s)
$\sigma$	singular value

### **Abbreviations**

ARC	Adaptive Robust Control
ASW	Actuated Solid Wheelset
AIRW	Actuated Independently Rotating Wheels
DB	Deutsche Bahn
DIRW	Driven Independently Rotating Wheels
DSW	Directly Steered Wheels
EMU	Electrical Motor Unit
HOD	Hold-Off-Device
KTH	Kungliga Tekniska Högskolan (Royal Institute of Technology)
LMI	Linear Matrix Inequality
LQ	Linear Quadratic
LQG	Linear Quadratic Gaussian
PID	Proportional-Integral-Derivative
rms	root mean square
SNCF	La Société Nationale des Chemins de Fer
SYC	Secondary Yaw Control