



KTH Engineering Sciences

Methods for Reducing Vertical Carbody Vibrations of a Rail Vehicle

A Literature Survey

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Preface

This literature survey is a course part of my doctoral studies concerning active secondary suspension in rail vehicles. So far, the project has focused on lateral ride comfort improvements by means of active suspension. This study focuses on methods for reducing vertical carbody vibrations and should be regarded as an introduction to future work in the project.

The doctoral project is a part of the research and development programme Gröna Tåget (Green Train), financed by the Swedish Transport Administration (Trafikverket). This literature survey has been performed at the Royal Institute of Technology (KTH) in Stockholm, Sweden.

I would like to thank my supervisors Sebastian Stichel and Rickard Persson for valuable comments and inputs regarding this survey.

Stockholm, April 2010

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Abstract

The trend towards higher rail vehicle speeds generally results in increased vibrations in the carbody, which has a negative impact on ride comfort. Carbody vibrations can be reduced either by focusing on the structural stiffness of the system or by optimizing the damping components. When a conventional passive damping system cannot be further optimized, active components can be a solution to achieve improvements.

Previous research concerning active control in rail vehicles to improve ride comfort show that significant benefits may be gained compared to a passive system. The overall goals are normally to improve, or at least maintain, ride comfort at increased vehicle speed or when running on tracks of inferior quality. Therefore, active suspension can be regarded as a cost-efficient solution if vehicle speed can be increased or track maintenance costs can be minimised. However, despite satisfactory results throughout the years, active suspension in rail vehicles has not yet made a convincing breakthrough in operational use. The main reason for the lack of success is most likely that the solutions offered so far have been too expensive in relation to the benefits gained.

The purpose of this literature survey is to give an overview of previous studies regarding methods to passively or actively achieve vertical vibration reduction in a rail vehicle. The main focus is on active components.

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1 Introduction

In rail operation today, there is a general trend towards increased vehicle speeds. However, higher speeds usually generate increased forces and accelerations on the vehicle, which has a negative impact on ride comfort. Furthermore, the demands for higher speeds in turn increase the requirements for lighter carbodies, which reduce the impact between wheels and rails, energy consumption and mostly also manufacturing costs. However, lowering the carbody weight also means a reduction of the stiffness of the structure, which results in lower natural frequencies. This increases the risk of resonance vibrations, which negatively affects ride comfort.

There are a number of approaches to maintain good vertical ride comfort. Carbody vibrations can be reduced either by focusing on the structural stiffness of the system or by optimizing the damping components. When a conventional passive damping system cannot be further optimized, active components can be a solution to achieve improvements. With active technology, actuators are installed to add forces that counteract and hence suppress vibrations [42].

Active control in rail vehicles has been investigated for several decades, showing significant benefits compared to passive systems. It is a measure to improve, or at least maintain, ride comfort at increased vehicle speeds or when track conditions are unfavourable. Therefore, active suspension should be regarded as a cost-efficient solution if vehicle speed can be increased or track maintenance costs can be minimised. However, despite satisfactory results, active rail vehicle suspension has not yet made a convincing breakthrough in operational use (except for the tilting train technology). The main reason for the lack of success is most likely that the offered solutions so far have been too expensive in relation to the benefits gained.

Research that has been done during the years concerning improved ride comfort in rail vehicles by means of active suspension has been compiled in a number of literature surveys. The first survey on active control in ground transportation, known to the author, was performed in 1983 by Goodall and Kortüm [16]. In 1997, a literature survey on active suspension in rail vehicles was performed by Goodall [17], summarising the research in the areas of tilting, active primary and secondary suspensions. Ten years later, another survey was performed by Bruni, Goodall, Mei and Tsunashima [4], updating the research performed in the area of active control for railway vehicles.

1.1 Purpose of this work

This study is part of the author's doctoral thesis, focusing on different methods of passively or actively reducing vertical carbody vibrations of a rail vehicle. The study gives an overview of previous research in the area. Chapter 2 describes the general carbody dynamics, including the rigid and flexible modes that have the largest influence on vertical ride comfort. In Chapter 3, different methods of ride comfort assessment are presented. Chapter 4 describes previously investigated methods to reduce vertical vibrations of the carbody, and hence to improve vertical ride comfort. Finally, this study is summarised in Chapter 5.

The sky-hook damping and *LQG* (Linear Quadratic Gaussian) control strategies mentioned in this study are not further explained, since a detailed description has been given in a previous study performed by the author [28]. The same applies for the different actuator types mentioned.

2 Carbody dynamics

In order to improve ride comfort it is important to have knowledge about the different vibration modes of the carbody. The vibrations of a carbody are mainly caused by track irregularities that are transmitted up to the carbody via the bogies. In order to achieve good ride comfort, these vibrations have to be suppressed, since ride comfort is negatively affected by high accelerations in the carbody.

The carbody vibrations can be divided in two types of modes: *rigid* and *flexible*. The rigid body modes that affect vertical ride comfort are bounce, pitch and roll. These modes normally lie in a relatively low frequency range, around 1 Hz [51]. The flexible modes are twisting and bending deformations of the carbody due to forces acting on it. Typical flexible modes that are interesting when focusing on vertical ride comfort are the first vertical bending mode, first torsion mode (torsional motion along the longitudinal axis) and the first so-called breathing mode, resulting from cross-section shear. Normally, the frequencies for the first flexible modes lie in the range of 8–15 Hz, however, depending on the vehicle configuration and settings. The higher the eigenfrequency, the more complex the mode shapes [2]. To avoid resonance between carbody and bogie, the lowest possible bending frequency of the carbody is determined by the eigenfrequencies of bounce and first bending mode of the bogie. The human body is most sensitive to vertical vibrations in the frequency range 4–10 Hz [10, 40, 41]. Therefore, to secure good ride comfort, it is of importance not to let the eigenfrequencies of the carbody coincide with this sensitivity range.

Carlbon has performed a comprehensive study on the carbody structural dynamics of a rail vehicle [5, 6, 7, 8, 9]. A Swedish commuter coach is used as case study, where both simulations and measurements are performed. The study examines the structural flexibility of the carbody and what role it plays on ride comfort. Vibrations up to 20 Hz are considered. This is the frequency range where structural flexibility of the carbody has the largest influence. In some cases, the carbody structural flexibility accounts for nearly half of the comfort weighted *rms* values of the vertical vibrations. The mode shapes of the eight lowest flexible modes can be seen in Figure 2.1. The dark shading corresponds to deformations with small amplitudes. The corresponding undamped eigenfrequencies are listed in Table 2.1.

Zhou, Goodall, Ren and Zhang [51] have studied how the vertical carbody flexibility influences vertical ride comfort. As mentioned before, the first vertical bending mode has the largest influence on vertical ride comfort. A theoretical vehicle model with rigid and flexible carbody modes is used. When the structural stiffness of

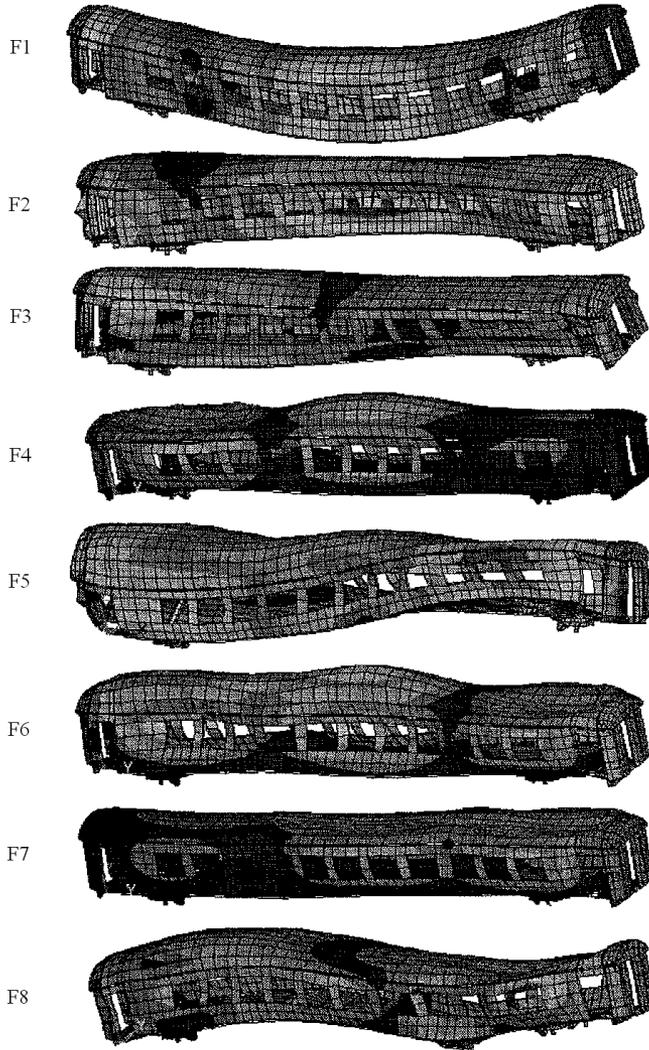


Figure 2.1: The eight lowest flexible eigenmodes of the studied carbody in [9].

the carbody is decreased, so are the bending frequencies, and there is a risk of significant vibrations in the carbody. Simulation results show that when the first vertical bending frequency is less than 7 Hz the carbody vibrates violently; see Figure 2.2. However, increased damping ratio of the first bending mode of the carbody, ζ , may attenuate the resonant peak values to some extent. Further, it is shown that when the first bending frequency is greater than 10 Hz ride comfort of

Table 2.1: The eight lowest mode shapes and undamped frequencies of the studied carbody in [9].

No.	Mode shape	Frequency (Hz)
F1	First vertical bending	9.1
F2	First lateral bending	12.2
F3	Torsion 1	12.8
F4	Breathing 1	13.4
F5	Torsion 2	13.9
F6	Breathing 2	14.3
F7	Breathing 3	15.0
F8	Second vertical bending	16.2

the flexible carbody is approaching the value for the rigid carbody, used as reference in the study.

However, the resonance peak around 6.2 Hz seen in Figure 2.2 depends on vehicle running speed. The bending frequency where ride comfort of the flexible carbody is approaching the one for the rigid carbody is increased with increased speed. Hence, the higher the speed, the higher stiffness of the carbody is required.

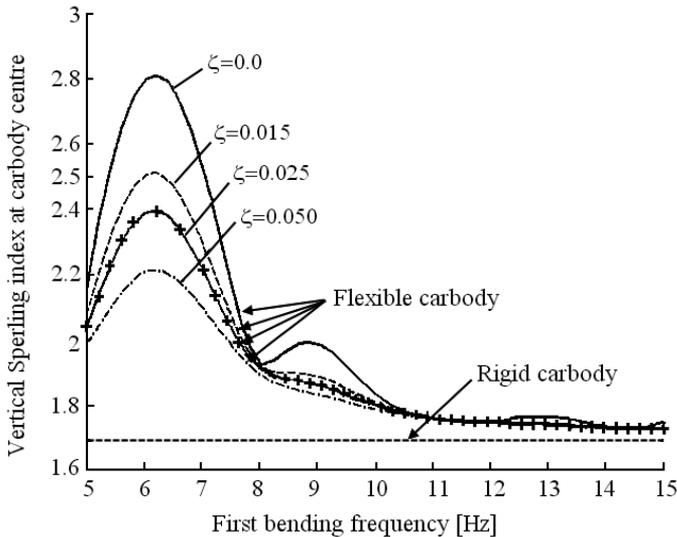


Figure 2.2: Vertical ride comfort in the centre of the carbody as a function of the first carbody bending frequency, for different damping ratios of the bending mode [51].

3 Ride comfort evaluation

Good ride comfort is one important issue to aim for within rail vehicle development. Different conditions require various evaluation methods. Therefore, it is difficult to establish a universally applicable international standard on ride comfort of rail vehicles.

A study performed by Suzuki [46] gives a survey of different ride comfort evaluation approaches applied in Japan. The *rms* value of the carbody acceleration is the most common way to evaluate ride comfort. However, there are a number of other possible quantities, such as peak or peak-to-peak values of accelerations, stationary lateral acceleration in curves, crest factor (i.e. the degree of vibrational shock, derived from the peak value divided by the *rms* value), acceleration and deceleration in longitudinal direction, frequency of roll, yaw and pitch. For tilting trains, the problem with motion sickness is an important issue. Here, carbody roll angular velocity and acceleration in transition curves are common indices to measure ride comfort.

In addition to measuring physical parameters, the response of the human being must also be taken into consideration when evaluating ride comfort. This can be done by verbal reporting (answering questions about the comfort), physiological response (e.g. measuring the heart rate) and activity interference (e.g. the ability to read and write) [46].

Ride comfort evaluation according to ISO 2631 is well described in the European standard EN 12299 [10]. The *rms* values of frequency-weighted accelerations on the carbody floor are evaluated as

$$a^{wrms} = \left[\frac{1}{T} \int_0^T [a^w(t)]^2 dt \right]^{0.5}, \quad (3.1)$$

where $a^w(t)$ is the frequency-weighted acceleration as a function of time t . $T = 5$ s is the duration of the measurement. The filter functions for the horizontal and vertical directions are illustrated in Figure 3.1. According to ISO 2631 evaluation, human bodies 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 3.1. The ride comfort levels for the individual lateral and vertical directions according to ISO 2631 are described in Table 3.1. The levels are the same for the lateral as well as the vertical direction.

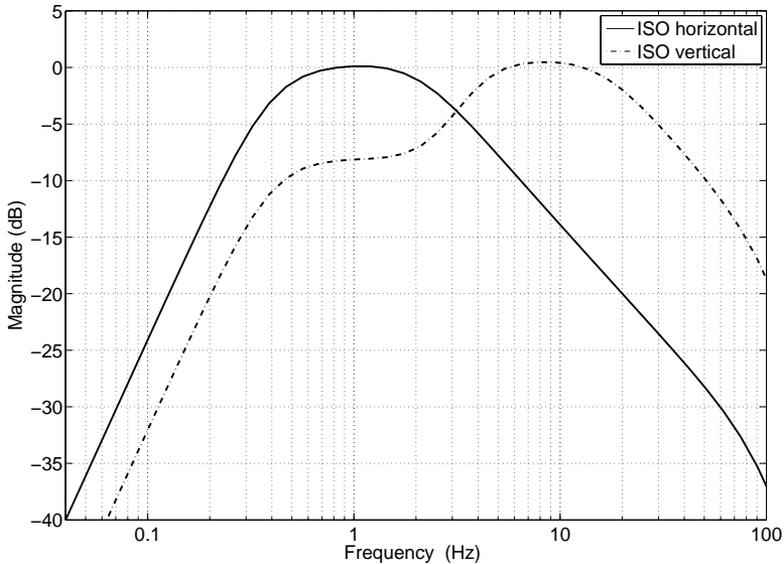


Figure 3.1: Frequency weighting of accelerations according to ISO 2631 (horizontal and vertical).

Table 3.1: Ride comfort levels (lateral and vertical) according to ISO 2631 [10].

a^{wrms} (m/s ²)	Ride comfort
$a^{wrms} < 0.2$	Very comfortable
$0.2 \leq a^{wrms} < 0.3$	Comfortable
$0.3 \leq a^{wrms} < 0.4$	Medium
$0.4 \leq a^{wrms}$	Less comfortable

Another comfort index described in [10] is the mean comfort standard method, N_{MV} . It is a comfort number according to accelerations in the three translational directions, measured on the carbody floor

$$N_{MV} = 6 \cdot \sqrt{(a_{XP95}^{w_d})^2 + (a_{YP95}^{w_d})^2 + (a_{ZP95}^{w_b})^2}, \quad (3.2)$$

where the three terms under the root are the 95th percentiles of the *rms* values of the frequency-weighted accelerations. The applied filter curves are approximately the

same as for ISO 2631 evaluation. Table 3.2 describes ride comfort levels evaluated by N_{MV} . The levels are the same for the lateral as well as the vertical direction.

Table 3.2: Ride comfort levels (lateral and vertical) according to N_{MV} [10].

N_{MV} (-)	<i>Ride comfort</i>
$N_{MV} < 1.5$	Very comfortable
$1.5 \leq N_{MV} < 2.5$	Comfortable
$2.5 \leq N_{MV} < 3.5$	Medium
$3.5 \leq N_{MV} < 4.5$	Uncomfortable
$4.5 \leq N_{MV}$	Very uncomfortable

A common comfort evaluation method applied, for example, in Sweden is the comfort number Wz . It originates from German research in the 1940's and 1950's by Sperling and Betzhold [40, 41] and is a frequency-weighted *rms* value of the lateral or vertical accelerations on the carbody floor, normally evaluated over a one kilometre distance. Wz is defined as

$$Wz = 4.42(a^{wrms})^{0.3}, \tag{3.3}$$

where a^{wrms} is the *rms* value of the frequency-weighted acceleration. The filter functions for the lateral and vertical directions are illustrated in Figure 3.2, originating from equations defined in [27]. The filter curves have the same shape, however, with a slightly higher sensitivity for lateral accelerations. According to Wz evaluation, human bodies are considered to be most sensitive to frequencies in the 4–7 Hz range, laterally as well as vertically, cf. Figure 3.2. Table 3.3 describes vibration levels according to Wz . The levels are the same for the lateral as well as the vertical direction.

Table 3.3: Vibration levels (lateral and vertical) according to Wz [27].

Wz (-)	<i>Vibration level</i>
1.0	Just noticeable
2.0	Clearly noticeable
2.5	Pronounced, but not unpleasant
3.0	Strong, but tolerable
3.5	Very strong and unpleasant
4.0	Extremely strong and unpleasant

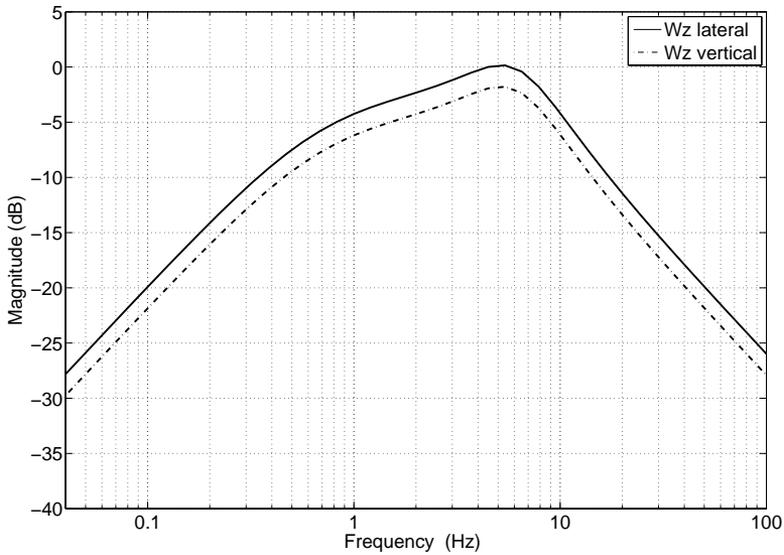


Figure 3.2: Frequency weighting of accelerations according to W_z ride comfort evaluation (lateral and vertical).

In Japan, a commonly applied comfort evaluation method is the ride quality level L_T (dB) [43]. It is designed based on comfort evaluation according to ISO 2631 [24] and is defined as

$$L_T = 10 \log_{10} \frac{1}{\alpha_{ref}} \int_{f_1}^{f_2} W_{LT}^2(f) P_\alpha df, \quad (3.4)$$

where $\alpha_{ref} = 1.0 \times 10^{-5} \text{ m/s}^2$, $f_1 = 0.5 \text{ Hz}$ and $f_2 = 80 \text{ Hz}$. W_{LT} is the comfort filter representing human sensitivity to vertical vibrations with a peak around 4–8 Hz. P_α is the power spectral density of the carbody acceleration.

4 Methods for carbody vibration reduction

This chapter describes previously investigated methods to reduce vertical vibrations of the carbody, and hence to improve vertical ride comfort. If the carbody is exposed to the same frequencies (or close to) as its natural frequencies, the corresponding eigenmodes will be excited. Vertical carbody vibrations can be reduced either by focusing on the structural stiffness of the system or by optimizing the spring and damping components. When an optimum has been reached for a conventional passive damping system, active components can be a solution for further improvements.

Tilt control mentioned in this section is concerned with the roll mode of the carbody, mainly to reduce lateral accelerations experienced by passengers. However, the vertical mode is to some extent connected to rolling and therefore it is treated in this study.

4.1 Primary suspension

The primary suspension consists of spring and damper components between the bogie and the wheelset. The purpose with this suspension level is to secure a stable running behaviour, but simultaneously to ensure low track forces, low wear and good curving behaviour. Depending on how it is designed, it can also be a measure to influence ride comfort in the carbody. The component in the primary suspension that—if properly designed—can have an influence on ride comfort is the vertical *axle-box damper*. There are normally four axle-box dampers in each bogie.

Vibrations transmitted up to the carbody can be suppressed by focusing on reduction of bogie vibrations. Zhou, Goodall, Ren and Zhang [51] have suggested a passive approach to improve ride comfort by increasing the primary damping coefficient. A higher primary damping coefficient reduces the vibrations of the bogie so that a smaller amount of vibrations is transmitted to the carbody; see Figure 4.1. It should be noted that ride comfort here is evaluated by the Sperling index W_z , whose sensitivity range coincides with the resonance frequency. However, this approach may increase the dynamic wheel-rail forces and might transfer short wavelength track irregularities to the carbody.

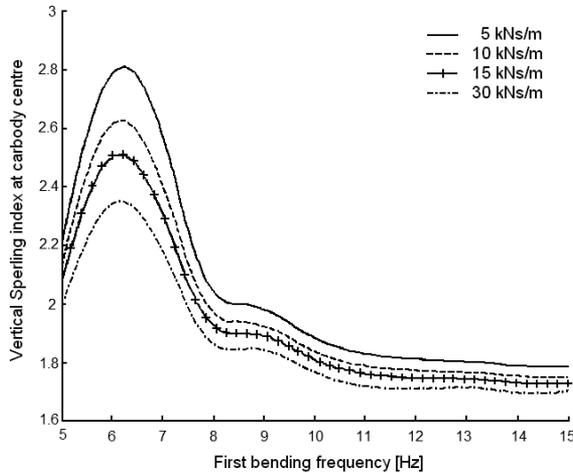


Figure 4.1: Vertical ride comfort in the centre of the carbody as a function of the first carbody bending frequency, for different primary damping coefficients [51].

For further improvements, when the passive suspension has been optimized, active suspension can be considered. In a Japanese study performed by Sugahara, Takigami and Koganei [43], semi-active vertical actuators are placed in parallel with the primary axle spring to reduce vertical vibrations of the bogie in order to suppress the first vertical bending mode of the carbody. Firstly, simulations with a seven degrees of freedom vehicle model are performed. Secondly, running tests are carried out with two types of Shinkansen vehicles on two different test sections. The two control strategies sky-hook and LQG are compared. Test results show that both vehicle types suppress the vibration amplitude of the first bending mode to 15–20 % of that obtained with a passive system. Both considered control strategies provide good results; the ride comfort level L_T is reduced by at least 3 dB, which is an improvement noticeable by passengers.

Active primary suspension in a two-axle rail vehicle is investigated by Pacchioni, Goodall and Bruni [29]. The bogie frame and hence the secondary suspension are removed to reduce the vehicle weight. The actuator is placed directly between the carbody and the axle-box. This active primary suspension is supposed to compensate for the non-existing secondary suspension. Only bounce and pitch modes are considered. The control strategies sky-hook and LQG are investigated to improve ride comfort and to keep suspension deflection at an acceptable level. Simulation results show that both control strategies offer significant ride comfort improvements. However, sky-hook damping is slightly better than LQG control when the modelled actuator is ideal. When actuator dynamics are considered in the model, LQG control performs better than sky-hook damping.

In a study performed by Sugahara, Kazato, Takigami and Koganei [42], both the primary and secondary suspensions are considered when using active damping to suppress the modes that significantly affect ride comfort. Damping control in the primary suspension is applied to the vertical axle-box dampers to suppress the vertical vibrations of the bogies, and hence to reduce the first vertical bending mode of the carbody. Furthermore, active damping in the secondary suspension is applied to the air spring to suppress the rigid vibration modes bounce and pitch. Measurements are performed on a rolling stock test rig with characteristics of a Shinkansen vehicle. The natural frequency of the first vertical bending mode is 8.5 Hz, whereas the carbody bounce and pitch modes are 1.1 and 1.6 Hz, respectively. The control strategies sky-hook and LQG are considered. Measurement results show that axle-box damper control reduces the first vertical bending mode around 8.5 Hz, whereas air spring control is most efficient in reducing the bounce and pitch modes around 1–2 Hz. With combined control, the ride comfort value L_T is reduced by 4.7 dB at the centre of the carbody. Active damping of the vertical axle-box dampers reduces the vertical acceleration around 8.5 Hz in the centre of the carbody down to 15 % of the passive system when LQG control is used. The corresponding reduction for sky-hook damping is 22 % of the passive system.

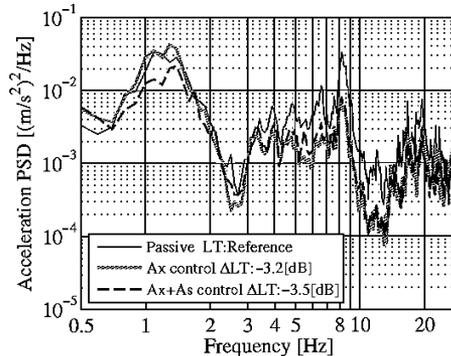


Figure 4.2: Measured vertical accelerations (psd) in the centre of the carbody. Running tests with a Shinkansen vehicle [44].

This study was further extended and described in [44], where running tests were performed with a Shinkansen vehicle. In each car, eight active dampers are mounted parallel to the existing axle springs in the primary suspension, replacing the passive vertical dampers. Further, the four air springs are made active by controlling the flow rate of the air passing through the orifice. In this study, sky-hook damping is the only considered control method. When only the primary vertical dampers are controlled, measurement results from the running tests show that the first bending mode vibrations around 8.2 Hz in the centre of the carbody are reduced to approximately 20 % of the passive system. In addition, when the air springs are controlled, the rigid body mode vibrations of the carbody around 1 Hz are

reduced to approximately half of the passive system. The improvements around 8.2 Hz are maintained although control of the air springs is included. The results are illustrated in Figure 4.2, where also the ride comfort improvement of 3.5 dB for the combined control is seen.

4.2 Secondary suspension

The secondary suspension interconnects the carbody and bogie, with the purpose of isolating the carbody from excitations transmitted from track irregularities via the wheelsets and bogie frames.

The *air spring* is part of the secondary suspension of most modern passenger rail vehicles, placed between the carbody and bogie. Its main task is to reduce carbody accelerations in the lower frequency range, around 1 Hz. Figure 4.3 illustrates a typical air spring system with its different components. The amount of air in the air bag (no. 3) decides the stiffness of the air spring. By means of the level control (no. 8), the air volume of the air bag can be regulated and the stiffness can be adapted to the preload (amount of passengers). The air is provided from the air container (no. 9), which is regulated by the compressor (no. 10). The main benefits with an air spring compared to a conventional coil spring are

- increased stiffness for increased preload,
- height independent of preload due to level control,
- significant horizontal stiffness,
- low height,
- good sound and vibration isolation.

Cost and complexity are two disadvantages mentioned together with air springs [2].

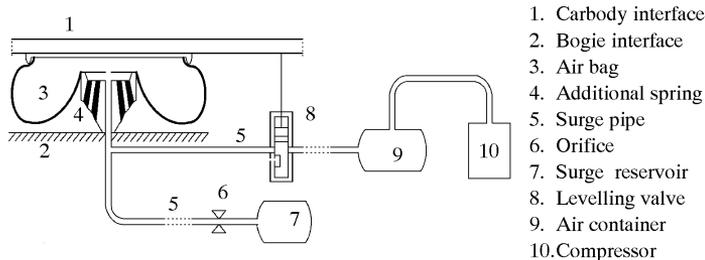


Figure 4.3: A typical air spring system in a rail vehicle [3].

Berg [3] has proposed a non-linear air spring model for dynamic analysis, representing air spring behaviour in the frequency range 0–20 Hz. The quasi-static displacement for the model is in the range 0–50 mm, whereas dynamic displacements between 0 and 25 mm are of main interest. The proposed model is validated against measurements. Generally, good agreement is shown between simulation and measurement results.

Further, the secondary suspension of a rail vehicle normally consists of *vertical dampers*. These are often hydraulic dampers, with the purpose of reducing low-frequency vehicle vibrations. There are normally two vertical dampers connected to each bogie.

The carbody and bogie can be interconnected by an *anti-roll bar*, sometimes known as stabiliser, with the purpose of reducing the carbody roll. The anti-roll bar is normally transversely mounted on the bogie with vertical links connected to the carbody, and can be regarded as a torsional spring.

Moreover, the secondary suspension consists of *emergency springs* (bumpstops) that limit the relative vertical displacements between carbody and bogies. There is an air gap, ensuring that the bumpstop is activated first when the displacement has become large enough. The bumpstop is a rubber component with significant stiffness. When bumpstop contact occurs it has a negative impact on ride comfort.

The most common way to improve ride comfort is to focus on the secondary suspension. For several decades, active control applied to the secondary suspension has been studied. Pratt gives a comprehensive introduction to active suspensions in a rail vehicle [35], focusing on the application in high-speed rail vehicles in order to maintain or improve ride comfort at higher speeds. Both vertical and lateral directions of a full-scale vehicle model are considered. The study concerns the investigation of how actuator dynamics influence the overall operation of an active suspension system. Simulation results show that active secondary suspension applied to a single rail vehicle can improve ride comfort by 30 % in the vertical direction, and 45 % in the lateral direction. However, with actuator dynamics included, ride comfort may be degraded by up to 15 % with certain actuator types. Further, the potential for preview control in a multiple-car vehicle with active suspension is studied, showing that ride comfort may be improved by 10 %.

Li and Goodall [26] have investigated different approaches of sky-hook damping applied to the secondary suspension of a quarter-car model. Three linear and two non-linear approaches with different filtering solutions are theoretically analysed to optimize the trade-off between good ride comfort and minimised suspension deflection. The linear method with a so-called complementary filter improves ride comfort 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, show over 50 % ride comfort improvement, however, with larger suspension deflection compared to the passive system.

Simulations with a 31 degrees of freedom vehicle model have been performed by Wu, Zeng and Dai [49], where the interesting modes are the first vertical bending mode, first lateral bending mode and first torsion mode. A flexible carbody is used when comparing semi-active secondary suspension with a passive system. Results show that ride comfort improvements can be achieved with lateral semi-active damping compared to conventional passive damping. However, with vertical semi-active damping no significant improvements could be observed. Simulation results were validated against measurements from a roller test rig.

In an Italian study performed by Pugi, Rindi, Bartolini and Auciello [37], semi-active magneto-rheological (*MR*) actuators are modelled in Simulink. The four vertical passive dampers in the secondary suspension are replaced by four *MR* actuators. The *MR* actuators are independently controlled to reduce vertical, pitch and roll accelerations of the carbody. Compared to a passive system, vertical ride comfort—measured by the ride index N_{MV} —is reduced by 10–20 %.

In an Italian study performed by Alfi, Bruni and Facchinetti [1], active control of the air spring is used to reduce the low-frequency vertical and pitch vibrations of the carbody (below 2 Hz). Generally, an air spring contains a levelling valve connected to the reservoir volume. In this study, the "passive" levelling valve is replaced by a flow regulation servo-valve. The proposed mathematical air spring model is validated against experimental data, showing good agreement. The vehicle model has six degrees of freedom and sky-hook damping is applied as control law. Simulations show that the actively controlled air spring significantly reduces vertical carbody accelerations around 1 Hz compared to a passive system. Figure 4.4 shows the vertical carbody acceleration with passive and active suspensions at different speeds. However, the disadvantage with this approach is the high air consumption that is required.

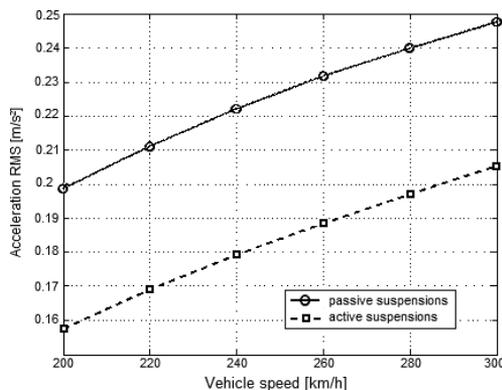


Figure 4.4: Vertical carbody acceleration (*rms*) over the rear bogie for different running speeds [1].

Another study concerning vertical ride comfort improvements by modifying the air spring was performed by Freiholtz [14]. Simulations are performed on a Swedish Regina train, showing that the air reservoir volume has a large influence on ride comfort. Therefore, a second reservoir is added to the air spring, resulting in increased volume and hence lower stiffness. By this measure, the stiffness peak of the air spring is reduced in order not to coincide with the one for the carbody.

Vertical actuators in the secondary suspension have been studied by Fagerlund [12]. Simulations are performed using a one-car rail vehicle with rigid bodies. One considered scenario is to remove the anti-roll bar and the passive vertical dampers in the secondary suspension. Instead, two ideal vertical actuators (i.e. no dynamics are considered) in each bogie are placed in parallel with the passive springs. The actuator performance, such as required maximum forces, is studied. Simulation results show that the actuator forces are higher than the actuator proposed by Kjellqvist [25] is able to handle. Further, LQ control is studied, but is shown not to be appropriate for the vehicle model.

For good vertical comfort it is desirable to have high damping to reduce roll motions, but low damping to reduce vertical accelerations of the carbody. A damping arrangement is presented in [15], which is designed to optimize ride comfort according to vertical and roll modes of the carbody. The proposed damping arrangement also comprises lateral dampers that provide damping in the lateral direction as well as of yaw motions between carbody and bogie. The vertical damping arrangement is illustrated in Figure 4.5. Each damper contains two chambers separated by a movable piston. The dampers are connected with each other by pipes, through which the medium (e.g. oil) can flow. One pipe interconnects the upper respective the lower chamber of the two dampers. A second pipe conversely interconnects the other two chambers. In case of a roll motion, the connected chambers experience the same pressure (underpressure or overpressure), which yields that no flow arises in the pipes. Hence, the roll motion is damped internally by the respective dampers. On the other hand, in case of a vertical motion, underpressure in a chamber connected to a chamber with overpressure gives rise to flow in the pipe. The flow between the dampers results in weaker damping. Hence, comfort related to vertical and roll motions can be optimized.

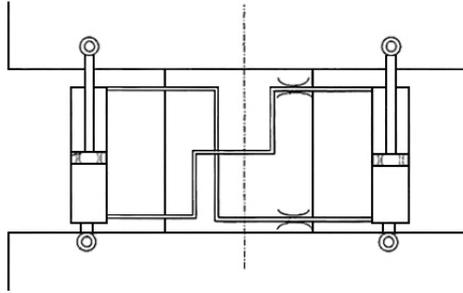


Figure 4.5: Damping arrangement for vertical and roll motions [15].

4.3 Damping of carbody eigenmodes

Stiffness of the carbody

From a ride comfort point of view it is preferable to increase the structural stiffness of the carbody in order to reduce the risk of carbody oscillation resonance, cf. Chapter 3. The following measures can be taken to increase the carbody stiffness, and hence to increase the natural frequencies of the carbody [2]:

- Make the carbody as short as possible. This conflicts, however, with the desire to maximise the number of passengers in relation to e.g. the number of bogies.
- Make the carbody cross-section as large as possible. However, the cross-section must not interfere with the prescribed carbody gauge profile. Furthermore, crosswind stability problems may arise.
- Use stiffer carbody materials or a design resulting in high carbody stiffness. This is, however, normally contradictory with the requirements on lightweight carbodies for higher speeds.

In a study performed by Suzuki and Akutsu [45], it is investigated how vertical vibrations affect ride comfort (assessed by comfort level L_T (dB)). If the structural stiffness is decreased, the natural frequency of the first bending mode is decreased as well, which gives poorer ride comfort, especially at higher speeds. To decrease flexural vibrations, higher structural stiffness is desirable.

Separated mass

By separating a part mass dynamically from the carbody the natural frequency of the carbody can be increased [32]. By this separation the part mass is no longer included in the carbody weight and does not contribute to the dynamics of the carbody. Hence, the natural frequency of the carbody is increased and the risk of carbody oscillation resonance is reduced. The separation is achieved by suspending the part mass elastically from the carbody; see Figure 4.6. The weight of the separated mass is of importance. If an improvement of the carbody oscillations is to be noticed, the separated mass has to be at least 10 % of the remaining carbody mass. A common application of the approach with a separated mass is to implement dynamic damping between the carbody and transformer to increase the natural frequency of the carbody. In addition, traction motors can be treated as separate masses.

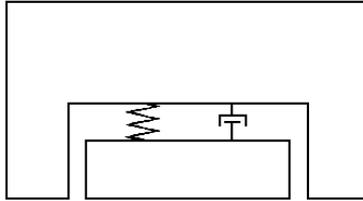


Figure 4.6: The principle of mass separation to increase the natural frequency of the carbody [32].

The suspension between the separated mass and the carbody can be actively controlled. This approach is investigated by means of simulations in a study performed by Holts [23]. The aim is to oscillate the transformer in anti-phase with the carbody and hence to suppress the carbody vibrations. The study shows that active control significantly suppresses both lateral and vertical carbody vibrations. However, best results are achieved in the lateral direction.

In a study performed by Foo and Goodall [13], the first vertical bending mode of the carbody is suppressed by adding an electro-magnetic actuator between the centre of the carbody and an auxiliary mass of one tonne. Additionally, hydraulic actuators across the secondary suspension in the front and rear of the vehicle are included. Three different configurations of sky-hook modal control—to separately handle the bounce and pitch modes of the carbody—are investigated. The central actuator effectively reduces the flexibility effect on the ride comfort. Compared to a passive system one of the considered control configurations with actuator dynamics included reduces vertical accelerations by approximately 60 % in the rear part of the vehicle.

Piezoelectric actuators

Piezoelectric elements can be used to suppress vibration modes of the carbody and hence to improve ride comfort. These elements are able to convert mechanical energy, i.e. vibration energy from the structure, to electrical energy and vice versa [20]. If a piezoelectric element is attached to the carbody and subjected to strain generated from vibrations of the carbody, a voltage is produced in the element. This electrical energy can then be dissipated in a shunt circuit connected to the element. The energy loss has a damping effect of the carbody, since the electrical energy originates from vibrations of the carbody [47].

A study performed by Schandl, Lugner, Benatzky, Kozek and Stribersky [39] describes the use of piezo-stack actuators attached to a metro vehicle in order to create a bending moment that reduces the vertical vibrations of the carbody. Twelve actuators are attached to the carbody to increase the damping of the first three eigenmodes (first vertical bending mode, first diagonal distortion mode and first torsion mode); see Figure 4.7. The considered eigenmodes have a large impact on ride comfort since their frequencies for this model coincide with the ones where the human body is most sensitive to vibrations (up to 10 Hz).

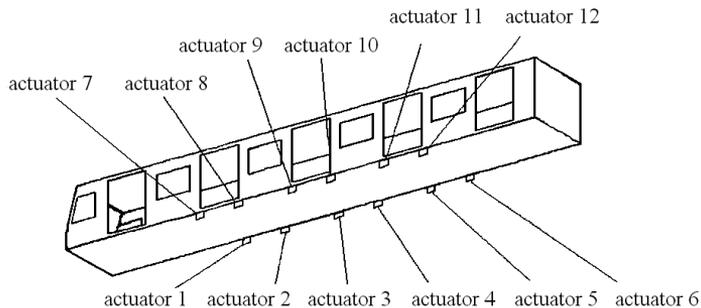


Figure 4.7: Twelve actuators are mounted on the longitudinal beams of the underside of the carbody [39].

Piezoelectric elements and their ability to simultaneously suppress two structural modes have been studied and described by Hansson [21] and Hansson, Takano, Takigami, Tomioka and Suzuki [22]. Single-mode damping is applied to suppress the first bending mode of the carbody. In addition, two-mode damping also includes the third bending mode of the carbody. Simulations with a simplified model are compared with experiments performed with a 1:5 scale model of a Shinkansen vehicle. Further, simulations with a full-scale vehicle model are performed. It is shown that simultaneous damping of two structural modes of the carbody is possible. Simulations show that two-mode damping can reduce the first bending mode by 37 % (Figure 4.8a) and the third bending mode by 43 % (Figure 4.8b).

However, the effectiveness of the mode suppression depends among other things on the placement, size and number of piezoelectric elements. Furthermore, the simulation results are better than the experiment results, most likely due to simplifications in the simulation model.

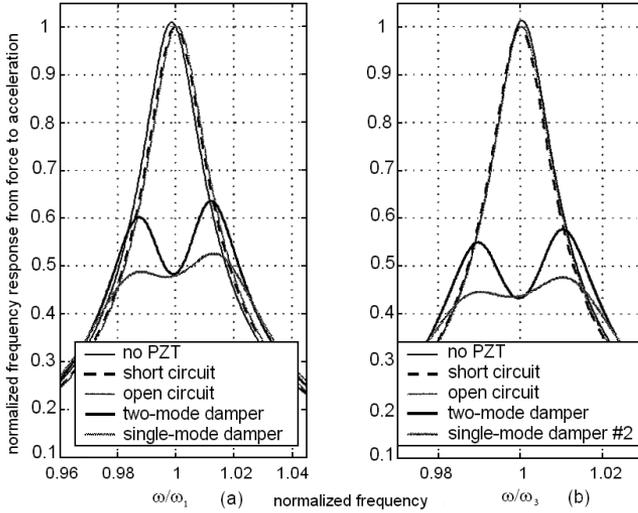


Figure 4.8: Simulated peak reduction of the frequency response near the carbody centre at a) first bending mode and b) third bending mode [21].

In a Japanese study performed by Takigami and Tomioka [47, 48], the influence of piezoelectric actuators on the vertical bending vibrations of a rail vehicle is evaluated. In a first stage, tests are performed on a 1:5 scale of a Shinkansen vehicle with eight piezoelectric actuators mounted in two rows under the carbody floor. The actuators cause a distributed carbody bending moment due to the applied voltage. As a second step in the research, stationary excitation tests are performed on a 19.5 m commuter coach. The number of piezoelectric actuators is now increased to 264, arranged in so-called "Piezoelectric Damper Units" (*PDU*s). Despite the large number of actuators, the total weight is no more than approximately 60 kg. A 30 % reduction of the frequency response function (*FRF*) in the vertical direction is observed.

4.4 Active inter-vehicle damping

Instead of considering actuators conventionally placed across the secondary suspension between carbody and bogies to improve ride comfort, the actuators can be mounted between two cars. This is referred to as active inter-vehicle damping; see Figure 4.9 for the principle of inter-vehicle damping. The greatest benefit of this approach is the reduction of number of actuators required, which in turn leads to lower weight, power and cost. Another benefit of active inter-vehicle damping is the reduced actuator force bandwidth requirements. Conventional actuators placed across the secondary suspension are exposed to higher frequencies due to the bogie accelerations. Inter-vehicle actuators, in contrast, deal with the carbody accelerations, which to some extent already have been filtered by the primary and secondary suspensions, and hence contain lower frequencies [36].



Figure 4.9: Principle of inter-vehicle damping.

The first study known to the author concerning active inter-vehicle damping was presented in 1994 by Pratt and Goodall [34]. A three-vehicle model is used in order to compare inter-vehicle actuators with conventionally placed actuators across the secondary suspension. Bounce and pitch are the considered modes to evaluate ride comfort. *LQG* control is used in all cases. The active inter-vehicle damping manages to reduce either bounce or pitch, depending on the weighting in the cost function. However, the two modes cannot be reduced simultaneously.

Three years later, the same authors presented simulation results from an extended study, again based on a three-vehicle model [36]. Optimal active inter-vehicle suspension is compared with the corresponding optimal passive suspension. Results imply that active inter-vehicle damping does not perform much better than the concept of optimized passive inter-vehicle damping. A strategy concentrating on the middle car shows significant ride comfort improvements in that car, however, at the expense of degraded ride comfort in the outer cars. One way to overcome this problem is to install actuators conventionally placed across the secondary suspension in the outer cars.

Goodall and Speedie [19] describe simulations that are performed with a three-vehicle model, where vertical displacement and pitch are the considered modes. Both an ideal actuator and an actuator with dynamics included are modelled. The results with inter-vehicle actuators connecting the middle car with the two outer are compared with the corresponding results when using conventionally placed

actuators across the secondary suspension. The results show that vertical accelerations in the middle carbody are decreased with inter-vehicle actuators, however, at the expense of increased accelerations in the outer carbodies, as described in the previously mentioned study.

4.5 Tilting

When a rail vehicle travels through a curve the passengers experience an outward acceleration. Increased vehicle speed means increased acceleration, which has a negative impact on ride comfort. The objective of tilt control is to lean the vehicle inwards in the curve to reduce the acceleration felt by the passenger; see Figure 4.10. By this measure, higher speed in curves can be allowed without the need of increasing the track cant.

As already mentioned, tilt control is concerned with the roll mode of the vehicle. However, the vertical mode is to some extent connected to rolling and therefore it is briefly treated in this study.

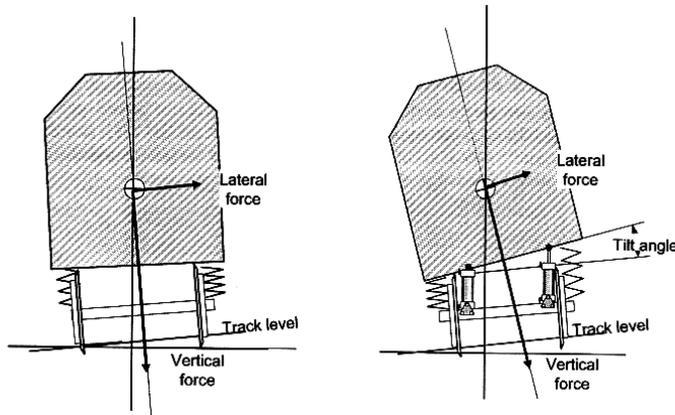


Figure 4.10: The lateral force is decreased due to tilting of the carbody (right) [33].

Tilt control is the most successful part of active technology in rail vehicles. In 1972, the first actively tilted trains were taken into commercial service by DB (Deutsche Bahn) in Germany [33]. The real breakthrough came around 1990, when a series production of tilting trains started in Sweden and Italy. Carbody tilting is now well established within railway technology.

Zolotas and Goodall [52] have given an overview of control applied to a rail vehicle. Firstly, the fundamentals of a rail vehicle and its dynamics are described together with an overview of different types of control concepts. Secondly, a case study of control of the secondary suspension is presented by describing different approaches

to the concept of tilt control. One common way to implement tilt control is to use *precedence control*. This means that information from the leading bogie is used for the rest of the vehicle in order to get a more precise control. The control design must take the vehicle speed and length into account as well as delay introduced by the filters. The objective of tilt control is to reduce the lateral acceleration felt by the passengers to zero, which is referred to as nulling tilt. However, one problem with tilting trains is the not ignorable number of passengers that experience motion sickness. Therefore, it is normal not to fully compensate for the acceleration to minimise this motion sickness phenomenon. This is referred to as partial tilt compensation with a typical compensation around 60–70 %.

Normally, the secondary suspension of a passenger rail vehicle consists of an anti-roll bar—also known as stabiliser. One way to control the roll mode of the carbody is to add an active element to the anti-roll bar; see Figure 4.11. This measure is investigated in a study performed by Pearson, Goodall and Pratt [30], where an active anti-roll bar is included in series with the roll stiffness to provide active tilt. The advantages are the small weight increase, low cost and that they are easily fitted. However, it may be difficult to provide the same degree of tilt as achieved with other tilting concepts. Three types of control strategies are designed and analysed (two classical and one optimal). The results show that all three control strategies provide good tilting performance. However, the optimal control approach is slightly better. For further research, the authors propose addition of a lateral actuator in the secondary suspension to deal with carbody displacements in curves and hence to allow higher speeds.

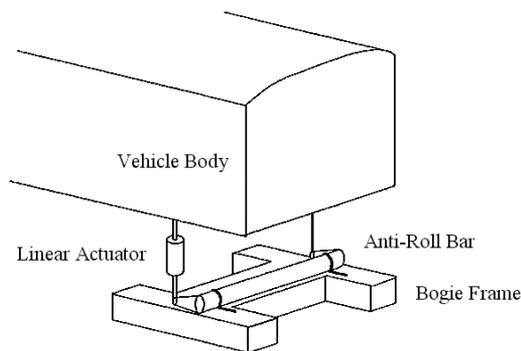


Figure 4.11: Principle of an active anti-roll bar [30].

The concept with integrated tilt and active lateral secondary suspension control was studied by Zhou, Zolotas and Goodall [50]. Simulations are performed with a four degrees of freedom end-view model, where only lateral and roll motions of the bogie and carbody are considered. The vertical modes are ignored. The roll actuator is placed in series with the roll stiffness and the lateral actuator in parallel with the lateral damper between carbody and bogie. The actuators are assumed to be ideal, i.e. no dynamics are considered in the actuator model. The lateral carbody acceleration and the effective cant deficiency are the feedback signals within the control-loop. When investigating the performance of an approach called New Dual-Actuator Control (*NDAC*), it shows improved ride comfort compared to a commercial precedence tilting control approach. Furthermore, the lateral suspension deflection is not increased, which usually is a problem when using active lateral suspensions in curving conditions.

4.6 Practical implementations

Although many control systems for reducing vertical carbody vibrations of a rail vehicle have been studied, very few systems have been applied to actual rail vehicles and tested on commercial lines. In Japan, running tests with a Shinkansen vehicle equipped with active primary and secondary vertical suspensions have been performed (described in Section 4.1) [43, 44]. The system with active primary vertical suspension has now reached an advanced development stage and is close to practical implementation in Japanese service operation [31].

As mentioned before, tilting technology is the most successful application within the area of active suspensions in trains when it comes to service implementation. In the early 1990's the tilting train *X2000* was developed by Adtranz (today Bombardier) and taken into traffic in Sweden. A few years later active secondary suspension working in lateral and yaw modes was tested in combination with the tilting technology [38]. However, the active lateral secondary suspension stayed on the experimental stage, whereas the tilt control is still in operational use today (2010).

Another application concerned with control of the roll motion of a carbody is applied to Bombardier's regional Talent train. An active anti-roll bar (stabiliser) is used to achieve carbody tilt. It is a transversely-mounted torsion rod on the bogie with vertical links to the carbody. One of the links is replaced by a hydraulic actuator, which applies carbody tilt via the torsion rod [11, 18].

5 Summary and discussion

There are different methods to reduce vertical vibrations of the carbody and hence to improve vertical ride comfort. Carbody vibrations can be suppressed either by focusing on the structural stiffness of the system or by optimizing the spring and damping components. When an optimum has been reached for a conventional passive damping system, active suspension can be a solution for further improvements. This study has focused mainly on active components to achieve vibration reduction in the carbody.

Vibrations transmitted up to the carbody can be reduced by suppressing the bogie vibrations. This suppression is achieved preferably by applying active control to the primary suspension. The research regarding this approach looks promising since it seems as if it soon will be implemented in service operation in Japan [31].

Furthermore, secondary suspension can be considered when aiming for ride comfort improvements. To control carbody dynamics, active components can be added to the secondary suspension. Vertical dampers and air springs are components in the secondary suspension that to some extent can be made active. Another measure to suppress vibrations of the carbody is to attach piezoelectric actuators to the carbody. These elements convert vibration energy to electrical energy, which has a damping effect on the carbody.

Moreover, actuators can be mounted between two cars in a train set, referred to as active inter-vehicle damping. To the author's understanding it may be difficult to design a concept of pure active inter-vehicle damping that yields overall ride comfort improvements in a rail vehicle. However, if the concept is supplemented by actuators conventionally placed across the secondary suspension in the leading and trailing bogies, acceptable comfort levels may be attained also in the outer ends of the vehicle. The question is if the levels are good enough. Further research in the area is required in order to find a beneficial solution.

Carbody tilt control is the most successful part of active technology in rail vehicles. The concept has been in commercial use for more than 20 years and is a well-established technology to maintain ride comfort in curves at high speed. However, the question regarding motion sickness is still an area that requires further research. Another way to control the roll mode of the carbody is to add an active element to the anti-roll bar in the secondary suspension.

Many approaches to reduce vertical carbody vibrations have been investigated over the years by means of rail vehicle simulation models. Some of the research considered in this study has also described experimental analyses on roller rigs

or on test tracks. However, despite promising results, the methods involving active technology have not yet been completely successful when it comes to implementation in service operation. The main reason for the lack of success is most likely that the solutions offered so far have been too expensive in relation to the benefits gained. Therefore, the challenge for all researchers working with active technology to improve ride comfort is to develop solutions with high performance and reliability, but at the same time at reasonable costs. Other areas within active technology in rail vehicles—not considered in this study—are at approximately the same development level and on the edge of reaching practical implementation. These are, for example, active wheelset steering, active lateral secondary suspension and lateral carbody centring in curves. Although some work remains before active technology in rail vehicles reaches its final breakthrough it will surely successively be more used in the coming years.

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Appendix A – Notations

Latin symbols

a^w	frequency-weighted acceleration (m/s^2)
a^{wrms}	<i>rms</i> of frequency-weighted acceleration (m/s^2)
f	frequency (Hz)
P_α	power spectral density of carbody acceleration ($(\text{m/s}^2)^2/\text{Hz}$)
t	time (s)
T	integration time (s)
w_b	vertical weighting curve
w_d	lateral/longitudinal weighting curve
W_{LT}	comfort filter for L_T

Greek Symbols

α_{ref}	acceleration constant (m/s^2)
ω	angular frequency (rad/s)
ζ	damping (Ns/m)

Abbreviations

DB	Deutsche Bahn
<i>FRF</i>	Frequency Response Function
ISO	International Organization for Standardization
KTH	Kungliga Tekniska Högskolan (Royal Institute of Technology)
<i>LQ</i>	Linear Quadratic
<i>LQG</i>	Linear Quadratic Gaussian
<i>MR</i>	Magneto-Rheological
<i>NDAC</i>	New Dual-Actuator Control
<i>PDU</i>	Piezoelectric Damper Unit
<i>psd</i>	power spectral density
<i>rms</i>	root mean square
<i>Wz</i>	Wertungszahl