Passive Components in Active Suspension System

By

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Master of Science Thesis
Preface and Acknowledgements

The present study is the result of the master thesis work carried out at Bombardier Transportation Västerås, Sweden, in cooperation with KTH Royal Institute of Technology, Stockholm, Sweden. The aim of this project was to set requirements on the passive suspension components when Active Lateral System (ALS) is put in the secondary suspension.

The thesis has been conducted under the supervision of Dr. Rickard Persson from Bombardier Transportation and Prof. Sebastian Stichel from KTH.

The financial support from Bombardier Transportation is greatly appreciated.

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My very special thanks to my wife Deepika Arora, for sacrificing everything to make my dream, come true.

Last, but not the least, thanks to my family for the kind support.

Västerås, December 2012

Arun Arora
Abstract

The concept of active technology in rail vehicles has been studied theoretically and experimentally for several decades and has now reached the stage of implementation. Active Lateral Suspension (ALS) is the active technology that leads the development in active secondary suspensions if carbody tilting is disregarded.

Active suspension systems may have an influence on running safety of the vehicle. The requirements to fulfil are related to forces between wheel and rail. The safety must be assured by the manufacturer by a safety assessment, which must be sent to the authorities before entering service. An important part of the assessment is to show that the active system, under all conditions, is part of a vehicle that runs safely on the track. The passive components in the vicinity of the active system have an important role in assuring that even a defective active system fulfils the required safety.

This master thesis aims to set requirements on the passive suspension components for the failed ALS.

The study has been conducted by varying various parameters of the vehicle with different running conditions and failure cases. The study highlights that the secondary lateral bumpstop is the most important parameter for the vehicle safety.

With soft bumpstop (low stiffness) the vehicle runs within safe limits for all studied conditions, and the effect of varying other parameters, running conditions and failure cases, is marginal. For somewhat higher stiffness (medium bumpstop), the effect of other parameters plays a critical role in ensuring safe run. For hard bumpstop, the track shift forces are above the limit values, independently of the passive component settings.

High vertical forces have been observed for certain cases with medium bumpstop, due to primary vertical bumpstop contact, which can be prevented by increasing the primary vertical damping or by increasing the vertical play. Reduction in track shift forces has been observed with the increase of primary vertical damping. The reason for the effect is not known and is proposed for further study.
### Notations

<table>
<thead>
<tr>
<th>Notation</th>
<th>Explanation</th>
<th>Unit/value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Y</td>
<td>Lateral wheel force</td>
<td>kN</td>
</tr>
<tr>
<td>Q</td>
<td>Wheel load</td>
<td>kN</td>
</tr>
<tr>
<td>( Q_0 )</td>
<td>Static wheel load</td>
<td>kN</td>
</tr>
<tr>
<td><strong>Soft Bumpstop</strong></td>
<td>Very low stiffness</td>
<td>29%</td>
</tr>
<tr>
<td><strong>Medium Bumpstop</strong></td>
<td>Medium stiffness</td>
<td>53%</td>
</tr>
<tr>
<td><strong>Hard Bumpstop</strong></td>
<td>Original stiffness</td>
<td>100%</td>
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</tbody>
</table>
## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Explanation</th>
</tr>
</thead>
<tbody>
<tr>
<td>AIRW</td>
<td>Actuated Independently Rotating Wheels</td>
</tr>
<tr>
<td>ALS</td>
<td>Active lateral suspension</td>
</tr>
<tr>
<td>ASW</td>
<td>Actuated Solid Wheelset</td>
</tr>
<tr>
<td>DIRW</td>
<td>Driven Independently Rotating Wheels</td>
</tr>
<tr>
<td>DSW</td>
<td>Directly Steered Wheels</td>
</tr>
<tr>
<td>HOD</td>
<td>Hold off device</td>
</tr>
<tr>
<td>IRW</td>
<td>Independently Rotating Wheel</td>
</tr>
<tr>
<td>LQ</td>
<td>Liner Quadratic</td>
</tr>
<tr>
<td>LQG</td>
<td>Liner Quadratic Gaussian</td>
</tr>
<tr>
<td>PD</td>
<td>Vertical primary damper</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional Integral derivative</td>
</tr>
<tr>
<td>PSD</td>
<td>Power Spectral Density</td>
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<tr>
<td>PX</td>
<td>Longitudinal primary suspension stiffness</td>
</tr>
<tr>
<td>PY</td>
<td>Lateral primary suspension stiffness</td>
</tr>
<tr>
<td>qst</td>
<td>Quasi-static</td>
</tr>
<tr>
<td>rms</td>
<td>Root mean square</td>
</tr>
<tr>
<td>SX</td>
<td>Longitudinal secondary suspension stiffness</td>
</tr>
<tr>
<td>SYC</td>
<td>Secondary Yaw Control</td>
</tr>
<tr>
<td>SSZ</td>
<td>Vertical secondary suspension stiffness</td>
</tr>
<tr>
<td>SSA</td>
<td>Simpack Simulation Automation</td>
</tr>
<tr>
<td>VHS</td>
<td>Very high speed</td>
</tr>
</tbody>
</table>
# Table of contents

1 Introduction ......................................................................................................................... 8

1.1 Background ...................................................................................................................... 8

1.2 Thesis Objective .............................................................................................................. 8

2 Literature review ................................................................................................................. 9

2.1 Suspension ....................................................................................................................... 9

2.2 Active suspension system .............................................................................................. 9

2.3 Active primary suspension ............................................................................................. 10

2.4 Tilting ............................................................................................................................... 11

2.5 Active secondary suspension ......................................................................................... 12

2.5.1 Configuration .............................................................................................................. 13

2.5.2 Control ......................................................................................................................... 13

2.5.3 Trends ......................................................................................................................... 14

2.6 Failure of active suspension system .............................................................................. 15

3 Wheel rail forces ............................................................................................................... 16

3.1 Running safety ............................................................................................................... 16

3.2 Track loading .................................................................................................................. 17

4 Vehicle and model description .......................................................................................... 18

4.1 Active Lateral Suspension ............................................................................................ 18

4.2 Actuator description ...................................................................................................... 18

4.3 Controller ....................................................................................................................... 19

4.3.1 Dynamic damping ...................................................................................................... 19

4.3.2 Hold Off Device ......................................................................................................... 20

4.4 Vehicle Description ....................................................................................................... 20

4.5 Operating Conditions .................................................................................................... 21

4.6 Track description ........................................................................................................... 23

4.7 Vehicle Parameters ....................................................................................................... 24

4.8 Failure scenarios ........................................................................................................... 25

5 Solution Methodology ...................................................................................................... 28
1 Introduction

1.1 Background

The concept of active technology in rail vehicles has been studied theoretically and experimentally for several decades and has now reached the stage of implementation. Active Lateral Suspension (ALS) is the active technology that leads the development in active secondary suspensions if carbody tilting is disregarded.

Active suspension systems may have an influence on running safety of the vehicle. The requirements to fulfil are related to forces between wheel and rail. The safety must be assured by the manufacturer by a safety assessment, which must be sent to the authorities before entering the service. An important part of the assessment is to show that the active system, under all conditions, is part of a vehicle that runs safely on the track. The passive components in the vicinity of the active system have an important role in assuring that even a defective active system fulfils the required safety.

1.2 Thesis Objective

The study is part of a high speed train project at Bombardier Transportation. The aim has been to set requirements on the passive suspension components in case of failure of the ALS, in the secondary suspension system. The overall focus has been to vary the properties of passive components in the suspension system, in relation to the relevant safety requirements. An acceptance space has been created to define what combinations of the passive components lead to safe running conditions, in all modes of the active system.
2 Literature review

The purpose of this chapter is to give the reader the necessary background information to understand the suspension system. This chapter introduces some of the fundamental ideas involved in the design and development of a rail vehicle suspension. It also includes the current knowledge related to active suspensions, both in terms of the necessary hardware and the method of control.

The chapter describes only the basic concepts understood from the literature study from the references. Reader is advised to further study the referred documents for a complete understanding.

2.1 Suspension

In general, the suspension is the set of elastic elements (springs), dampers and associated components which connect the wheelsets to the carbody [1]. The suspension can be divided into primary and secondary suspension, (Figure 1). The primary suspension is the suspension between the wheelset and the bogie frame. The secondary suspension is the suspension between the bogie frame and the car body. The springs are used to equalize the vertical loads between the wheels, stabilize the motion of the vehicle on the track, and to reduce the dynamic forces and accelerations due to track irregularities. Dampers are used to damp the oscillations in the suspension.

![Figure 1: Schematic picture of where the secondary suspension is located (It is within the dotted box) [1].](image)

2.2 Active suspension system

In the field of rail technology, there are continuously rising requirements concerning riding comfort, running safety, and speed from the side of the railway operators. These requirements are opposed by the fact that the condition of the tracks is getting worse and maintenance is becoming expensive. In view of this conflict, conventional suspension concepts are quickly at their limits. To meet the very conflicting requirements for introducing active suspension systems, in an economical way, is paramount [2].

An active suspension system mainly comprises of actuators, sensors, and electronic controllers [1]. As a comparison, the conventional, passive, suspension is purely mechanical. There are three major categories of active suspension that
are studied for railway vehicles: active tilting, active secondary suspensions, and active primary suspensions.

Tilting of the car body is used to reduce the quasi-static lateral acceleration experienced by the passengers in curves, and by this improving passenger ride comfort. Active tilting is a standard technology for railway vehicles [1].

Secondary active suspension is intended to improve the vehicle dynamic response and provide a better isolation of the vehicle body to the track irregularities, compared to a fully passive suspension. The improved performance could for instance be used to improve the ride comfort for the passengers [1].

Active primary suspension is intended to improve running stability and curving performance. There is a trade-off between those issues, and it is difficult to further improve both simultaneously with passive techniques. When using active primary suspension, the wheelsets can be either independently rotating, or connected by a solid axle [1].

2.3 Active primary suspension

In 2001, Streiter, Boller, Riege, Schneider and Himmelstein [2] 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 [3]. The results were promising, fulfilling all requirements by Deutsche Bahn, and launched the potential for real mechatronic bogie structures with economical character, especially for high speed applications.

In 2007, Bruni, Goodall, Mei and Hitoshi [4] studied the control and monitoring for rail vehicle dynamics, and have categorized and summarized variety of approaches to improve stability and guidance through active technology. Five different configurations for primary suspension are listed below:

- 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 [5], who investigated the use of controlled traction rods to improve steering performances without compromising on stability. It later 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. The concept was introduced by Mei and Goodall [6]. In this concept the actuation is introduced in the form of a steering torque applied on the axle, and it is demonstrated that the
control torque required to run the Independently Rotating Wheel (IRW) vehicle through a curve is much lower than the one required for a solid wheelset.

The DIRW concept has been investigated by Gretzschel and Bose [7] & [8] with reference to high speed/long-distance applications. In this concept an active control is achieved by traction motors applied to the independently rotating wheels. The advantage of this concept with respect to the IRW concept is that it has a very good running stability behaviour as well as improved curving performance through active control.

The DSW concept was investigated by Aknin et al. and Wickens [9] & [10]. It is a further simplification of the ASW concept, with two independently rotating wheels instead of a solid axle. The pairs of wheels are actively steered in order to achieve radial steering in curves which ensures stability on straight track. The concept has been recently studied by Michitsuji and Suda [11] as well, where active wheel steering control is used to complement and improve the performance of the vehicle with independently rotating steerable wheels.

The SYC fifth concept has been treated by Diana et al [12]. In this concept, 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.

2.4 Tilting

Tilting is the part of active technology that has been the most successful in the area of railway vehicles. In 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 [3].

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. 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 [3].
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 [3].

2.5 Active secondary suspension

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 ride comfort [3]. 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 ride comfort. The goal with actively controlled secondary suspension is to

1) Improve the passenger ride comfort under the same vehicle speed and track conditions,

2) Maintain good passenger ride comfort despite increased vehicle speed or

3) Maintain good passenger ride comfort despite worse track conditions.

To improve ride comfort in railway vehicles, an investigation has been conducted by Y Sugahara, A Kazato, R Koganei, M Sampei, and S Nakaura [13]. The investigation aims to reduce bending and rigid-body-mode vibration in the car body simultaneously, by introducing damping control devices in the primary and secondary suspensions. The proposed technique involves a control system of primary vertical dampers and air springs; the former are used to suppress the first bending mode vibration; the latter, to suppress the rigid-body-mode vibration. Experiments have been conducted with computer simulations and on the vehicle, on the Sanyo–Shinkansen line. The results demonstrated reduction in vertical vibrations in the car body as well as the bogies, leading to better ride comfort.
2.5.1 Configuration

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. A number of different configurations for active secondary suspension are possible. Actuators can either be used to replace the passive suspensions completely or can be used in conjunction with passive components. In the former the suspension behaviour will be completely controlled via active means whereas, in the latter, the size of an actuator can be significantly reduced as the passive component will be largely responsible for providing a constant force to support the body mass of a vehicle in the vertical direction or quasi-static curving forces in the lateral direction [3] & [4].

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. Whereas, on the other hand 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 [3]. In practice, a combination of a parallel spring for load-carrying and a series spring to help with the high frequency response is the most appropriate arrangement [3] & [4].

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 [3].

2.5.2 Control

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 namely

- PID control
- Sky-hook control
- $H_\infty$ control
- LQ/LQG control

PID controller is a classical loop-shaping with a proportional-integral-derivative controller. It is widely used in industrial control systems. The PID controller creates an input signal to the system process by attempting to correct the error between a demanded reference signal and the actual output signal [3].

Sky-hook damping is one of the most implemented control algorithms in the area of active technology in trains. The name is based on the idea that the system is damped relative to a fictive sky reference point, instead of the ground. It is a high bandwidth system, which can be used to give other improvements in suspension performance, largely through the provision of damping to an absolute datum. High levels of modal damping can be achieved without increasing the suspension’s transmissibility at high frequencies [3] & [4].
H∞ control is an advanced control methodology concerned with finding a controller for the open-loop system, such that the closed-loop system has good performance. In simpler terms it deals with finding a controller such that control signal ensures internal stability of the closed loop system and counteracts the influence of the disturbances. The advantage of this control model is its robust stability and offering of good system performance. A drawback is that the 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 [3].

LQ/LQG control theory is another control theory that is concerned with optimization is the so-called Linear Quadratic (LQ) control, or extended to Linear Quadratic Gaussian (LQG) 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 [3].

2.5.3 Trends

Active secondary suspension concepts have now been well developed and the technology proven to a level where widespread introduction would be straightforward; but the survey conducted by Bruni, Goodall, Mei and Hitoshi [4], suggests that there is very limited use of the concept in service. The reality is that, as long as a vehicle’s ride quality is good enough, further improvements do not bring clear system benefits to justify the business case. Whether the option for exploiting the improved performance of an active secondary suspension (i.e. enabling the use of lower quality track) has been properly explored is not clear, but the general trend for high speed railway infrastructure is towards very high quality track, which therefore is a contra-indication for their more widespread use.

More advanced concepts involving control of multiple vehicle dynamic modes are possible. One such concept has been demonstrated experimentally for an innovative ‘RailCab’ system being developed in Germany [4]. This involves a technically complex, full-active suspension/tilt module that yields a comfortable suspension and a reduced sensitivity to track irregularities.

Recently, the research and development engineering team at Bombardier Transportation has produced a completely new and innovative type of bogie called as FLEXX bogie [14]. It is said to overcome conventional bogie limitations such as passive steering and suspension elements. Mechatronics technology in the bogie, arguably a train’s most crucial component, is said to provide unique new functionalities, including:

- Active Radial Steering (ARS) and bogie stabilization
- Active comfort improvement
- Tilting
- Intelligent condition monitoring (FLEXX Guide)

The technology is said to enable higher speeds (up to 15% increase) with increased passenger ride comfort on curved tracks. Integrated into the existing secondary suspension, this innovative system is said to compensate for the
carbody’s natural roll movement. Hence, reducing both travel times and infrastructure investment.

The state-of-the-art technology will be incorporate on TWINDEXX double-deck trains, delivered to the Swiss Federal Railways (SBB).

2.6 Failure of active suspension system

Failure of an active suspension system could have an adverse effect on the running stability of the rail vehicle as well as passenger ride comfort. A recent study conducted by Richard Schneider and Mike Baert [15] suggests that a failure of an active suspension system could lead to excessive lateral movements of the carbody with respect to the running gear, leading to a violation of the kinematic envelope defined for the specific track the vehicle is negotiating. In order to avoid such violations of the kinematic envelope under any circumstances, typically, the outer contour of the wagon body and the suspension system of a rail vehicle are specifically adapted to the track system the vehicle is to be operated on. On one hand this approach has the disadvantage of restricting the outer counter of the carbody, which reduces the transport capacity of the vehicle whereas, on the other hand, a rather rigid suspension of the carbody leads to a poor ride comfort for the passengers.

A malfunction of, for example, the tilting control in an active suspension system, could lead to the introduction of opposite lateral excursions of the carbody with respect to the leading running gear and the trailing running gear. Such a situation, due to the specific kinematics of such a tilting system, could lead to a torsional loading of the carbody, leading to undesired unloading of some of the wheels of the running gears and, consequently, leading to a considerable increase in the risk of derailment [15].

Studies have been conducted by Bombardier Transportation on different failure modes of the active lateral suspension system [16] & [17]. The study has been conducted as part of the safety assessment for both normal working state and failure of the ALS to be equipped on the ETR1000 – V300ZEFIRO and Regina train respectively. The failure modes discussed in the study [16], (section 4.5), have been extensively considered as references for the study of this thesis.
3  Wheel rail forces

3.1 Running safety

Table 1 shows the assessment criteria for the running safety of the vehicle. The criteria are considered as safety-critical assessment values according to EN 14363 [18] & [19]. The value $\sum Y$ is used for assessing compliance with regard to safety against track shifting. The quotient $Y/Q$ is the criterion for safety against derailment resulting from the climbing of the wheel flange on to the rail.

Table 1: Assessment criteria

<table>
<thead>
<tr>
<th>Assessment</th>
<th>Description</th>
<th>Parameter</th>
<th>Applied Evaluation</th>
<th>Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Derailment</td>
<td>99.85 percentile of dynamic wheel climbing coefficient</td>
<td>$(Y/Q)_{99.85%}$</td>
<td>2 m sliding mean</td>
<td>0.8</td>
</tr>
<tr>
<td>Track shifting</td>
<td>99.85 percentile of lateral wheelset force</td>
<td>$(\Sigma Y)_{99.85%}$</td>
<td>2 m sliding mean</td>
<td>Equation 1</td>
</tr>
<tr>
<td>Stability</td>
<td>RMS of lateral wheelset force</td>
<td>$(\Sigma Y)_{rms}$</td>
<td>100 m sliding mean</td>
<td>Half of track shifting forces</td>
</tr>
</tbody>
</table>

The safety-critical limit for track shift force, $\sum Y$, has been calculated using Equation 1

$$\sum Y = K \left(10+2Q_0/3\right) \text{[kN]}$$ (1)

Where $K=1$, for locomotives, power cars, multiple units, and passenger coaches $K = 0.85$, for freight wagons.

$Q_0$ is the static vertical wheel load in kN.
3.2 Track loading

Table 2 shows the track loading quantities and limit values chosen as per EN 14363 [18] & [19].

Table 2: Track loading quantities and limit values

<table>
<thead>
<tr>
<th>Assessment</th>
<th>Description</th>
<th>Parameter</th>
<th>Applied evaluation</th>
<th>Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Track loading</td>
<td>Quasi-static lateral wheel force</td>
<td>$Y_{qst}$</td>
<td>Mean value in circular curve</td>
<td>60 kN</td>
</tr>
<tr>
<td></td>
<td>Quasi-static wheel load</td>
<td>$Q_{qst}$</td>
<td>Mean value in circular curve</td>
<td>145 kN</td>
</tr>
<tr>
<td></td>
<td>99.85 percentile of wheel load</td>
<td>$Q_{99.85%}$</td>
<td>2 m sliding mean</td>
<td>Equation 2</td>
</tr>
</tbody>
</table>

The maximum permissible vertical load, $Q_{99.85\%}$, has been calculated using the minimum of Equation 2 and 160 kN.

$$Q_{99.85\%} = 90 + Q_0 \text{ [kN]}$$

(2)
4 Vehicle and model description

4.1 Active Lateral Suspension

The ALS as discussed above is part of an active suspension system. It has mainly two tasks:

1) To create a lateral force in curves corresponding to lateral quasi-static acceleration times the pivot load of the bogie. The aim with this function is to centre the carbody above the bogie, allowing increased carbody width and improved cross-wind stability.

2) To reduce the vibration level in the carbody, due to imperfections in the track, by creating a force counteracting the vibration transfer from the bogie to the carbody. The aim of this function is to improve the average ride comfort.

Figure 3 shows the block diagram of an ALS [20]. In general, the principal of an active suspension system is based on the idea of controlling certain signals (e.g. carbody accelerations) with the signals themselves, i.e. by means of a closed loop.

![Figure 3: Block diagram ALS [20]](image)

In order to achieve this control loop the following must be included in the vehicle system:

1) An actuator placed between carbody and bogie. It could replace the conventional passive dampers between carbody and bogie.

2) A controller.

3) Sensors placed in the carbody and the bogie.

4.2 Actuator description

The actuator used in the chosen train is an electro hydraulic actuator developed by Liebherr in Germany [21] & [22]. It is a cylindrical damper with two chambers separated by a moveable piston, (Figure 4 & Figure 5). The chambers are provided with hydraulic fluid by means of two pumps which are driven by an asynchronous motor fed by power from the train. A pressure difference between the two chambers is generated by controlling the outflow of the hydraulic fluid by means of pressure control valves. The pressure difference enables the actuator to create
forces in both push and pull directions. In turn, the valves are controlled by varying the coil current, determined by force demand fed to the actuator. The actuator is able to generate forces up to 35 kN in both directions at frequencies up to a few Hertz \([21] \& [22]\).

![Electro hydraulic actuator](image1)

**Figure 4: Electro hydraulic actuator (developed by Liebherr)**

![Schematic figure of electro hydraulic actuator](image2)

**Figure 5: Schematic figure of electro hydraulic actuator**

### 4.3 Controller

The controller’s function in the chosen train can be divided into the following two parts

#### 4.3.1 Dynamic damping

Dynamic damping, applying the Sky-hook principle, is a straightforward and widely used methodology in rail vehicles. It controls the dynamic motion of the carbody. In this system, unlike the conventional passive system, the damping force is generated independently of the bogie velocity. This allows the sky-hook damping to be set larger than the conventional damping, thus avoiding vibration transfer from the bogie to the carbody.
4.3.2 Hold Off Device

When travelling in a curve at high speed (high track plane acceleration), the carbody tends to move laterally outwards in relation to tracks and bogies. The lateral carbody displacement is, however, limited by bumpstops, (Figure 6) [22].

![Diagram of carbody, gap, and air spring with bumpstop](image)

**Figure 6: Bumpstop connected to the carbody with some gap to the bogie frame [22]**

In order to centre the carbody in curves with the uncompensated lateral track plane acceleration, the active secondary suspension is combined with a Hold-Off-Device (HOD) [23]. The HOD concept is based on low-pass filtered lateral bogie acceleration (deterministic track inputs i.e. curve detection) as a reference signal. Therefore, it is also called low-bandwidth control, since the reference signal is used for detecting the low frequency content of deterministic track inputs, i.e. track geometry (curves). After low-pass filtering, the signal is multiplied with half the carbody mass in order to create an appropriate actuator force that counteracts the lateral movement of the carbody in the circular part of the curve. Hence, the carbody is centred above the bogies and bumpstop contact can be avoided.

4.4 Vehicle Description

A single car of a V300 Zefiro train has been selected for the study. The Zefiro is the latest class of very high speed (VHS) trains from Bombardier. It is claimed to be the most economical and environmentally friendly train in the world [24].

In the standard configuration, the train consists of eight connected cars in a fixed formation including a driver’s cab at each end, Figure 7.
Figure 7: Standard V300 Zefiro Train

The train is composed of the following car types:

- DM1  Driving Motor Car, 1st class, with traction converter and brake resistor
- TT2  Trailer Car, 1st class, with two DC pantographs and main transformer
- M3   Motor Car, 2nd class with bistro, with traction converter and brake resistor
- T4   Trailer Car, 2nd class, with two AC pantographs
- T5   Trailer Car, 2nd class, with two AC pantographs
- M6   Motor Car, 2nd class, with traction converter and brake resistor
- TT7  Trailer Car, 2nd class, with two DC pantographs and main transformer
- DM8  Driving Motor Car, 2nd class, with traction converter and brake resistor

4.5 Operating Conditions

Table 3 shows the various curves, speeds and track irregularities used for the analysis. The combinations have been chosen to obtain a track plane lateral acceleration of 1 m/s² at all speeds (customer requirement).

<table>
<thead>
<tr>
<th>Curve*</th>
<th>Length of straight track (m)</th>
<th>Length of transition track (m)</th>
<th>Curve Radius (m)</th>
<th>Cant (mm)</th>
<th>Vehicle speed (km/h)</th>
<th>Track Irregularity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve 1</td>
<td>100</td>
<td>96</td>
<td>250</td>
<td>120</td>
<td>76</td>
<td>ERRI high</td>
</tr>
<tr>
<td>Curve 2</td>
<td>100</td>
<td>96</td>
<td>500</td>
<td>120</td>
<td>107</td>
<td>ERRI high</td>
</tr>
<tr>
<td>Curve 3</td>
<td>100</td>
<td>120</td>
<td>1007</td>
<td>150</td>
<td>160</td>
<td>ERRI high</td>
</tr>
<tr>
<td>Curve 4</td>
<td>100</td>
<td>150</td>
<td>1903</td>
<td>150</td>
<td>220</td>
<td>ERRI low</td>
</tr>
<tr>
<td>Curve 5</td>
<td>100</td>
<td>225</td>
<td>3540</td>
<td>150</td>
<td>300</td>
<td>ERRI low</td>
</tr>
<tr>
<td>Curve 6</td>
<td>100</td>
<td>275</td>
<td>5097</td>
<td>150</td>
<td>360</td>
<td>ERRI low</td>
</tr>
</tbody>
</table>
Right hand curve
Table 4 shows the other operating conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel Profile</td>
<td>1/40th [25]</td>
</tr>
<tr>
<td>Track gauge [mm]</td>
<td>1435</td>
</tr>
<tr>
<td>Rail Inclination</td>
<td>1/20</td>
</tr>
<tr>
<td>Coefficient of friction</td>
<td>0.40</td>
</tr>
<tr>
<td>Rail Type</td>
<td>UIC 60</td>
</tr>
</tbody>
</table>

Table 4: Other operating conditions

Figure 8 shows the wheel contact geometry. The rail-wheel interface gives an equivalent conicity of 0.025 for most relative lateral displacements.

![Wheelset Contact Geometry](image)

Figure 8: Wheel contact geometry
4.6 Track description

For simulations Chinese measured track data has been used. The measured track section is between Changsha and Xianning. (On the line between Guangzou and Wuhan) and is about 2.5 km long. The track irregularities have been modified to comply with Power Spectral Densities (PSDs) according to ERRI [26]. Table 5 shows the track stiffness and damping.

Table 5: Track stiffness and damping

<table>
<thead>
<tr>
<th>Track stiffness</th>
<th>Track damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral, $c_y$ [N/m]</td>
<td>Vertical, $c_y$ [N/m]</td>
</tr>
<tr>
<td>$5.5 \times 10^7$</td>
<td>$7.5 \times 10^7$</td>
</tr>
</tbody>
</table>

Figure 9 and Figure 10 shows track excitation signal.

![Figure 9: Track excitation](image-url)
Figure 10: Track excitation PSD change to log-scale

4.7 Vehicle Parameters

Table 6 shows the identified vehicle parameters and their variation in percentage, for the simulations.

Table 6: Vehicle parameters

<table>
<thead>
<tr>
<th>Vehicle Parameter</th>
<th>Minimum (%)</th>
<th>Actual (%)</th>
<th>Maximum (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal primary Suspension stiffness (PX)</td>
<td>9%</td>
<td>100%</td>
<td>200%</td>
</tr>
<tr>
<td>Lateral primary Suspension stiffness (PY)**</td>
<td>13%</td>
<td>100%</td>
<td>200%</td>
</tr>
<tr>
<td>Longitudinal Secondary Suspension stiffness (SX)</td>
<td>16%</td>
<td>100%</td>
<td>200%</td>
</tr>
<tr>
<td>Vertical secondary Suspension stiffness (SSZ)</td>
<td>50%</td>
<td>100%</td>
<td>200%</td>
</tr>
<tr>
<td>Yaw Damper</td>
<td>20%</td>
<td>100%</td>
<td>200%</td>
</tr>
<tr>
<td>Lateral secondary bumpstop stiffness</td>
<td>29%</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>------------------------------------</td>
<td>-----</td>
<td>------</td>
<td>------</td>
</tr>
<tr>
<td>Anti-Roll Bar Stiffness</td>
<td>50%</td>
<td>100%</td>
<td>200%</td>
</tr>
</tbody>
</table>

** Includes rotational stiffness of the axle guide rod bush.

### 4.8 Failure scenarios

During the development of the ALS system different failure scenarios have been identified [16]. Table 7 shows twelve of these scenarios. The failure scenarios represent failures of the ALS actuator(s), the ALS controller, the sensors and external effects like power loss.

**Table 7: Failure scenarios**

<table>
<thead>
<tr>
<th>Serial Number</th>
<th>Status of the system</th>
<th>System action</th>
<th>Mitigation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>No AC power, one actuator. The motor is not running, but the pressure valves are still controlled. Semi-active mode for one actuator, the other is working normally.</td>
<td>Transfer to 3</td>
<td>Less critical than 3</td>
</tr>
<tr>
<td>2</td>
<td>No AC power, both actuators. The motors are not running, but the pressure valves are still controlled. Semi-active mode for both actuators.</td>
<td>Transfer to 3</td>
<td>Less critical than 3</td>
</tr>
<tr>
<td>3</td>
<td>ALS system down. The motors are not running and the pressure valves are uncontrolled. Both actuators act as passive dampers.</td>
<td>None.</td>
<td>Failure scenario 1</td>
</tr>
<tr>
<td>4</td>
<td>No force, one actuator. The motors are running, but the pressure valves are uncontrolled. Optionally the oil has leaked out.</td>
<td>Transfer to 3</td>
<td>Less critical than 6</td>
</tr>
<tr>
<td>Scenario</td>
<td>Description</td>
<td>Failure Scenario</td>
<td></td>
</tr>
<tr>
<td>----------</td>
<td>-------------</td>
<td>-----------------</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>One actuator produces very low passive damping, the other is working normally.</td>
<td>None</td>
<td>Less critical than 6</td>
</tr>
<tr>
<td>6</td>
<td>No quasi-static reference. The bogie sensor does not give any information about curving. The system does not provide any centring force, but the dynamic control is working.</td>
<td>Transfer to 3</td>
<td>Failure scenario 2</td>
</tr>
<tr>
<td>7</td>
<td>No force, both actuators. The motors are running, but the pressure valves are uncontrolled. Optionally the oil has leaked out. Both actuator produces very low passive damping.</td>
<td>Transfer to 3</td>
<td>Less critical than 8</td>
</tr>
<tr>
<td>8</td>
<td>Full force, both actuators. Could be due to actuator failure. One actuator is producing full force, the other is working normally.</td>
<td>Transfer to 5</td>
<td>Failure scenario 3</td>
</tr>
<tr>
<td>9</td>
<td>Full force, one actuator. Could be due to actuator failure. One actuator is producing full force, the other is working normally.</td>
<td>Transfer to 3</td>
<td>Less critical than 8</td>
</tr>
<tr>
<td>10</td>
<td>Full force, both actuators. Could be due bogie sensor failure. Both actuators are producing full force.</td>
<td>Transfer to 5</td>
<td>Failure scenario 3</td>
</tr>
<tr>
<td>9</td>
<td>No carbody feedback from one sensor. The system runs without carbody vibration feedback from one of two sensors.</td>
<td>None</td>
<td>Less critical than 10</td>
</tr>
<tr>
<td>10</td>
<td>Corrupt carbody feedback from one sensor. The system runs with corrupt carbody vibration feedback from one of two sensors, risk making the control unstable.</td>
<td>Transfer to 9</td>
<td>Failure scenario 4</td>
</tr>
</tbody>
</table>
|   | Corrupt actuator.  
|   | One actuator produces random  
|   | forces. The controller will try to  
|   | counteract also influencing the  
|   | other actuator. | Transfer to 3 | Failure scenario 5 |
| 11 | One actuator stuck.  
|   | One actuator cannot move, the  
|   | other actuator is working normally | Transfer to 3, but the  
|   | actuator will be stuck. | None. Failure very  
|   | unlikely (comparative to  
|   | risk of stuck  
|   | passive dampers). |

Among these twelve scenarios, five have been identified as necessary to investigate. Figure 11 shows how these five failure scenarios have been derived from the initially eleven (number 12 excluded). Note that the five simulated failures often are less likely representing the double failures, battery power loss, inverted feedback etc.

![Figure 11: Reduced failure scenarios](image)

**Figure 11: Reduced failure scenarios**
5 Solution Methodology

5.1 Phase split-up

Failure scenario 3, (Table 7), has been chosen as the most severe case for the simulations. The solution methodology has been divided into four phases.

5.1.1 Phase 01

In phase 01, passive components which could have influenced the vehicle safety, in case of ALS failure, have been identified. Simulations have been conducted using Simpack Simulation Automation (S.S.A.) for various vehicle speeds, curve radii, and variation of stiffness and/or damping of the identified components, (Table 3 & Table 6). Each component’s property has been varied one at a time.

The typical chosen variations have been 50%, 100% (original), and 200%. The purpose of this phase was to identify parameters with influence on the safety related properties, and discount the others.

5.1.2 Phase 02

In phase 02, based on the results obtained from phase 01, passive components showing variation equal and above of 10% in the output, have been identified.

The MBS-models in SIMPACK and the control models in SIMULINK have been simulated using co-simulation via the interface SIMAT, which allows exchange of simulation results between the two simulation environments every time step. The property of the identified parameters has been varied one at a time.

The purpose of this phase was to identify parameters with influence on the safety related properties and obtain an understanding about their nature.

5.1.3 Phase 03

In phase 03, based on the results obtained from phase 02, a further refinement of the selection of passive components, has been done. Simulations have been conducted by multiple parameter variation of the identified components. The simulations have been conducted using co-simulation via the interface SIMAT.

The purpose of this phase was to study in detail, the influence of the identified parameters for the vehicle safety.

5.1.4 Phase 04

The results obtained from phase 03 have been categorized into three scenarios namely, best scenario, worst scenario, and average scenario. A sensitivity check has been conducted by varying the identified parameters to the specified values, (Table 8). Each component’s property has been varied one at a time, cf. Table 8.
Table 8: Parameters for sensitivity analysis

<table>
<thead>
<tr>
<th>Serial No.</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Anti-Roll Bar Stiffness</td>
</tr>
<tr>
<td>2</td>
<td>Primary damper</td>
</tr>
<tr>
<td>3</td>
<td>Secondary suspension longitudinal stiffness</td>
</tr>
<tr>
<td>4</td>
<td>Secondary suspension vertical stiffness</td>
</tr>
<tr>
<td>5</td>
<td>Secondary vertical damping</td>
</tr>
<tr>
<td>6</td>
<td>Equivalent conicity</td>
</tr>
<tr>
<td>7</td>
<td>Coefficient of friction</td>
</tr>
<tr>
<td>8</td>
<td>Track stiffness</td>
</tr>
<tr>
<td>9</td>
<td>Centre of gravity</td>
</tr>
<tr>
<td>10</td>
<td>Air spring (inflated/deflated)</td>
</tr>
<tr>
<td>11</td>
<td>Unloaded trailer car (T4)</td>
</tr>
<tr>
<td>12</td>
<td>Failure Scenario 4, and 5</td>
</tr>
<tr>
<td>13</td>
<td>Run cases 1 to 5</td>
</tr>
</tbody>
</table>

5.2 Model description

A single motor car (DM8) with crush weight [18] and inflated air springs has been chosen. This is the heaviest vehicle and has been considered as a suitable model for the simulations. The vehicle has been modelled in Simpack, Multi Body Simulation (MBS) software. It includes relevant components such as carbody, bogie frames, axle-box, wheelsets, motor, gearbox, springs, dampers, anti-roll bar, bumpstops etc. Figure 12 shows the model of the DM8 carbody in Simpack.
5.3 Assumptions

The following assumptions have been made in order to carry out the simulations:

1. The carbody is considered as a rigid body.
2. Nonlinear behaviour of the air springs, in the secondary suspension system, has been linearized by converting them into linear coil springs.
3. Single point contact between wheel and rail has been assumed.
4. Results highlight the vehicle behaviour only in the curves. Therefore, the initial straight track and the transition section has been ignored.
5. Band-pass filter with 0.4 to 10 Hz cut-off frequency has been used to obtain resulting values for vehicle stability (Root Mean Square (RMS) of lateral wheelset force). This is unlike the procedure mentioned in EN 14363 [18]. It has been assumed that the vehicle instability lies in the frequency band.
6. Coefficient of friction is assumed to be 0.4 and has been assumed to be constant throughout the vehicle run.
6 Results

The following sections present the results for curve 6 (cant = 150mm, vehicle speed = 360 km/h, curve radius = 5097 m), (Table 3 & Figure 25), for failure case 3 (Full force on both actuators), (Figure 11), as this combination has been identified to be the worst case scenario. The main focus has been on the track shift forces ((∑Y)_{99.85%}), as they have been found to be close or even above the limit; with some cases showing exceedance in the vertical forces as well.

6.1 Longitudinal Secondary stiffness variation

Figure 13 shows the track shift forces variation with respect to longitudinal secondary stiffness. It can be observed that there is hardly any change in track shift forces with respect to variation in longitudinal secondary stiffness.

![Figure 13: Maximum track shift force in percent of permissible value as function of secondary suspension longitudinal stiffness, Curve 6, Failure case 3](image)

6.2 Bumpstop stiffness

Figure 14 shows the variation of track shift forces with respect to lateral bumpstop stiffness. It can be observed that softening the bumpstop decreases the track shift forces. This is attributed to the fact that a softer bumpstop is able to absorb more energy and transfer less energy to the wheelset than the stiffer one, for a certain displacement. A significant variation of track shift forces can be observed, highlighting that the lateral bumpstop stiffness variation could be an important parameter in the analysis.
Figure 14: Maximum track shift force in percent of permissible value as function of secondary lateral stiffness Curve 6, Failure case 3

Figure 15 shows the track shift forces with respect to time, for second axle, and soft, and hard bumpstop. Each peak represents a sudden impact between the wheel flange and the rail. It can be observed that the number of peaks as well their magnitude is much less for the soft bumpstop than for the hard bumpstop.

Figure 15: Track shift forces for second axle

6.3 Yaw damper

Figure 16 shows the track shift forces with respect to yaw damper stiffness for the three bumpstop stiffnesses namely soft, hard and medium. It can be observed that the track shift forces show insignificant variation with the increase of yaw damping, especially with soft and hard bumpstop stiffness. However, there is a marginal drop in track shift forces for medium bumpstop stiffness.

It can also be observed that with soft bumpstop stiffness the track shift forces are always below the limit of 100%, which represents safe operation. However, with
hard bumpstop the track shift forces are always above 100%. With medium bumpstop the track shift forces remain approximately constant between softest and original (100%) damping value. Afterwards it drops down to safer limits, and there after remains constant throughout.

Figure 16: Maximum track shift force in percent of permissible value as function of yaw damping - Curve 6, Failure case 3

6.4 Primary longitudinal stiffness variation

Figure 17 shows the track shift force variation with respect to the primary longitudinal stiffness. For soft bumpstop stiffness, it can be observed that there is very minor variation in track shift forces with respect to variation of primary longitudinal stiffness. However, it can be observed that the track shift forces always remain below the limiting value (100%) for soft bumpstop and always above the limiting value for the hard bumpstop.
6.5 Primary lateral stiffness variation

Figure 18 shows the track shift forces variation with respect to primary lateral stiffness. It can be observed that the track shift forces show insignificant variation with the variation of primary lateral stiffness. This could be attributed to the fact that the lateral bumpstop being softer than the primary lateral stiffness would dominate any variation in the primary lateral stiffness. However, the track shift forces slightly drop towards the softest value of primary lateral stiffness because at these values, the primary lateral stiffness becomes softer than the lateral bumpstop stiffness, and hence starts dominating.
6.6 Primary longitudinal and lateral suspension variations

Figure 19 shows the track shift forces variation with respect to primary longitudinal as well as lateral stiffness, for medium bumpstop stiffness. It can be observed that track shift forces are least for stiffer primary longitudinal stiffness and softer primary lateral stiffness.

![Figure 19: Maximum track shift force in percent of permissible value as function of primary longitudinal and lateral suspension, with medium bumpstop](image)

A softer primary lateral suspension is able to absorb more energy and hence the track shift forces are reduced. A stiffer primary longitudinal suspension arrests the yaw motion of the wheelset, (Figure 20), which reduces the oscillation amplitudes of the wheelset and hence the track shift forces.

![Figure 20: Yaw angle, second axle for stiff (top) and soft (bottom) longitudinal primary suspension](image)

Figure 21 presents the view behind the pinnacle in the Figure 19. It shows that at very soft longitudinal primary stiffness the track shift forces again start reducing.
This is due to the fact that with much softer primary longitudinal suspension; the radial steering of the vehicle is improved and hence the flange contact of wheelset is avoided to the large extent of the run.

Figure 21: Rotated view of Figure 19, Maximum track shift force in percent of permissible value as function of with primary longitudinal and lateral suspension, with medium bumpstop

Upon comparing the upper and the lower signals in Figure 22, it can be observed that the by reducing the primary longitudinal stiffness the number of peaks have been greatly reduced, which represents a better radial steering and lesser flange contact. Each peak represents a sudden impact between the wheel flange and the rail. Upon comparing the middle and the lower signals, it can be observed that with softening the primary lateral stiffness, the magnitude of peaks has been greatly reduced. This is again attributed to the fact that a softer lateral suspension is able to absorb more energy than the stiffer ones.
Figure 22: Track shift forces, second axle for soft longitudinal and lateral primary stiffness (top), stiff longitudinal and lateral primary stiffness (middle) and stiff longitudinal and soft lateral primary stiffness (bottom).

Figure 23 highlights the variation of vertical forces ($Q_{99.85\%}$) with respect to primary stiffness. It can be observed that with stiffer primary lateral stiffness and softer primary longitudinal stiffness the vertical forces increase and for some cases cross the 100% limit.

Figure 23: Vertical forces, medium bumpstop with primary longitudinal and lateral suspension
Figure 24 shows the vertical forces on the primary bumpstop for all four axles. A very sharp peak can be observed for the third and fourth axle, which represents a contact with the vertical primary bumpstop. A bumpstop contact leads to the direct transfer of the forces, leading to a sharp peak in the signal. These peaks are attributed to a high vertical track irregularity. The effect can only be seen for third and fourth axle because of the following reasons:

1. The centre of gravity is shifted slightly backwards.
2. Compared to the wheelsets in the first bogie, the wheelsets of the second bogie are less tightly clung to the rail, and hence are more likely to react to the vertical disturbances, than the leading ones.

![Figure 24: Vertical forces on the left primary bumpstop for the four axles](image)

6.7 Sensitivity analysis

A sensitivity analysis has been carried out for various conditions, (section 5.1.4). The following sections highlight the observation from the results obtained.

6.7.1 Run cases

Figure 25 shows the track shift forces for different curve options, (Table 3). It can be observed that track shift forces increases with the increase of curve number, (Table 3), which is mainly attributed to the vehicle speed. It can also be observed that track shift forces are above the limits for curve 6 except, for soft bumpstop, as a soft bumpstop is able to absorb the more energy.
6.7.2 Equivalent Conicity variation

A higher equivalent conicity enables a rail vehicle to steer better on curved tracks but, it also increases the risk for instability (hunting motion) on straight tracks.

Figure 26 shows the track shift forces for different chosen equivalent conicities. It can be observed that for soft bumpstop there is a marginal variation in track shift forces as the vehicle is able to run with minimum flange contact and with curve 6 being more like a straight track, the steering issues are minimal.

For medium bumpstop the reduction in track shift forces with increasing conicity can be observed. This is because with increasing conicity, the flange contact is avoided for longer duration of the run (because of better steering) and hence the reduction in track shift forces.

Similar trend could be observed for a hard bumpstop with the exception at equivalent conicity of 0.1. This could be attributed to the fact that for hard bumpstop the track shift forces are reduced, if the wheel flange is always in contact with the rail but, at equivalent conicity of 0.1, the flange contact is reduced, which increases the oscillations of the wheelsets, leading to sudden impacts between wheel flange and rail and hence, an increase in the peak values of track shift forces.

In short, certain conicity is required to avoid contact between wheel flange and rail. Less track shift forces are observed if either the wheel flange is in complete contact with rail (for e.g. at equivalent conicity of 0.025 in Figure 26) or the wheel flange-rail contact is completely avoided (for e.g. at equivalent conicity of 0.3 in Figure 26). For an in between value, the track shift forces are high because of high impacts between wheel flange and rail. Moreover with hard bumpstop the track shift forces are much higher in comparison to soft bumpstop, as a soft bumpstop is able to absorb more energy.

Figure 25: Maximum track shift force in percent of permissible value for the different run cases, Failure case 3
Figure 26: Maximum track shift force in percent of permissible value as function of equivalent conicity - Curve 6, Failure case 3

Table 9 shows the contact geometry combinations used to obtain the equivalent conicity of 0.1 and 0.3 [27]. The equivalent conicity has been calculated for amplitude of ± 3 mm [18]. Comparison of Table 9 with Table 4 highlights the main differences, to obtain the required equivalent conicities.

Table 9: Contact geometry combinations, resulting in different values of equivalent conicity

<table>
<thead>
<tr>
<th>Wheel Profile</th>
<th>Rail Profile</th>
<th>Track gauge (mm)</th>
<th>Rail Inclination</th>
<th>Type</th>
<th>Equivalent conicity</th>
<th>Designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>BR_P8-dense</td>
<td>UIC 60</td>
<td>1440</td>
<td>1:40</td>
<td>1</td>
<td>0.1</td>
<td>01T1</td>
</tr>
<tr>
<td>S1002</td>
<td>R_EN_52E1</td>
<td>1432</td>
<td>1:40</td>
<td>1</td>
<td>0.3</td>
<td>03T1</td>
</tr>
</tbody>
</table>

6.7.3 Friction

Figure 27 shows the track shift forces for different chosen coefficient of friction. It can be observed that with increase of friction the track shift force increases. This is attributed to the fact that with less friction the wheel flanges stay locked to the rail longer, than with higher friction. As the friction increases the wheel is able to move away from flange contact and hit back the rail with much higher forces, as the vehicle runs along the track, (Figure 28). It should be noted that such high conicities as 0.4 and 0.5 are unlikely at 360 km/h speed.
Figure 27: Maximum track shift force in percent of permissible value as function of friction - Curve 6, Failure case 3

Figure 28: Track Shift forces for friction variation - Curve 6, Failure case 3, and second axle

6.7.4 Vehicle parameters

Figure 29 shows the track shift forces for different chosen vehicle parameters. It can be clearly observed that the chosen parameters have marginal effect on the track shift forces with the exception of the primary vertical damper. The convergence of all lines at soft bumpstop further strengthens the observation that bumpstop characteristic is the key dominating parameter. With soft bumpstop, irrespective of the changes in the other vehicle parameters, there is hardly any change in the track shift forces. Also for hard bumpstop the variation in track shift forces is marginal.
However, the trend in primary vertical damping is unlike the other parameters. The track shift forces appear to be least for primary damper with twice the damping, than the original value. Also the trend for half and twice primary damping is very different than the original one.

Figure 29: Track Shift forces - Curve6, Failure case 3

Figure 30 further highlights the trends of primary vertical dampers for different bumpstop stiffness. It can be observed that for the soft bumpstop, the variation in primary damper doesn’t affect the track shift forces whereas, for the hard bumpstop the track shift forces decrease with increase in damping. The exact reason for this trend is not known and needs to be further investigated.

Figure 30: Maximum track shift force in percent of permissible value as function of primary damping, Curve 6, Failure Case 3
With increase in primary vertical damping, a reduction in vertical forces is also observed, (Figure 31). This is attributed to the fact that a stiffer primary damper is able to absorb more energy and hence the primary vertical bumpstop contact is avoided. The absence of peak in the 200% vertical primary damping clearly depicts no primary vertical bumpstop contact.

![Figure 31: Vertical forces, 3rd axle, curve6, failure case 3, 3rd wheelset, left wheel](image)

### 6.7.5 Other parameters

All the other parameters, chosen for sensitivity analysis, have shown no significant impact on the vehicle safety and hence are neglected. Table 10 shows the resulting values in percentage of limit values for such parameters.

**Table 10: Track shift forces ($\sum Y$) 99.85%**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Soft bumpstop (%)</th>
<th>Medium bumpstop (%)</th>
<th>Hard bumpstop (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Failure case 3</td>
<td>89</td>
<td>113</td>
<td>113</td>
</tr>
<tr>
<td>Track stiffness and damping 50%</td>
<td>91</td>
<td>100</td>
<td>113</td>
</tr>
<tr>
<td>Track stiffness and damping 200%</td>
<td>90</td>
<td>91</td>
<td>106</td>
</tr>
</tbody>
</table>
### Discussion

It is very clear from the results that the track shift force limit is the critical criterion in the vehicle safety judgement, with vertical force limits also being overshot in some cases. The key technique to keep the track shift forces under the limits is to either maintain a flange contact all through the curve, which reduces the lateral bouncing of the wheelsets; or to avoid the flange contact as much as possible, while negotiating the curve.

The results show that secondary lateral bumpstop characteristic is the key parameter for the vehicle safety under the given conditions of vehicle speed, track plane acceleration and track conditions. The following sections discuss the results in greater detail.

#### 7.1 Soft bumpstop

The results clearly show that with soft bumpstop the vehicle always runs within safe limits. Also with soft bumpstop the effect of changing properties of the other identified passive components, on track shift forces is negligible. It should be noted that soft here refers to soft stiffness at the working point when the actuator pushes the carbody into the bumpstop.

#### 7.2 Medium bumpstop

For medium bumpstop stiffness, the results present a very interesting picture about the vehicle’s response to variation in passive component’s properties. At first glance, (Figure 19) it appears that softening primary lateral suspension and

<table>
<thead>
<tr>
<th>Condition</th>
<th>Force Limit 1</th>
<th>Force Limit 2</th>
<th>Force Limit 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air spring inflated</td>
<td>83</td>
<td>99</td>
<td>114</td>
</tr>
<tr>
<td>Air spring deflated</td>
<td>92</td>
<td>92</td>
<td>87</td>
</tr>
<tr>
<td>Centre of gravity shift (laterally outwards)</td>
<td>91</td>
<td>111</td>
<td>116</td>
</tr>
<tr>
<td>Normal run (no failure)</td>
<td>79</td>
<td>81</td>
<td>80</td>
</tr>
<tr>
<td>Failure scenario 4</td>
<td>89</td>
<td>84</td>
<td>85</td>
</tr>
<tr>
<td>Failure scenario 5</td>
<td>85</td>
<td>89</td>
<td>88</td>
</tr>
<tr>
<td>Trailer car (T4) unloaded</td>
<td>88</td>
<td>95</td>
<td>91</td>
</tr>
</tbody>
</table>
stiffening primary longitudinal suspension would keep the vehicle within safe limits but, a pensive glance at the back side of the Figure 19 (the so called mountain), (Figure 21) gives another but a very interesting view of the picture. It suggests that below a certain limit, softening the primary longitudinal stiffness actually reduces the track shift forces! This is due to the fact that softening the primary longitudinal stiffness, beyond the certain limit improves radial steering, which avoids flange contact of the wheel with the rail.

High vertical forces, beyond safe limits have been observed, for the medium bumpstop. This has been accounted to the primary vertical bumpstop contact.

Hence it is very important to choose the correct properties of primary stiffness in case of medium bumpstop in order to keep the vehicle operation within safe limits.

7.3 Hard bumpstop
With hard bumpstop the results show that the vehicle always runs above the safe limits for curve 6, (Table 3) and given track conditions. Moreover the effect of varying the properties of the identified parameters on the vehicle safety has been found to be negligible.

7.4 Primary vertical damper
The study reveals that stiffening the primary vertical damper does help in avoiding the primary vertical bumpstop contact hence, a reduction in the vertical forces is observed. However, increasing the primary vertical damping also reduces the track shift forces, with the exception of medium and soft bumpstop, (Section 6.7.4). The reason for this effect is not known and is proposed for further study.
8 Conclusions

In the present study a particular high speed train, V300 Zefiro, has been studied for a failed active lateral suspension and different track cant, curve radius and vehicle speed. Curve 6 (cant = 150mm, vehicle speed = 360 km/h, curve radius = 5097 m), (Table 3 & Figure 25), for failure case 3 (Full force on both actuators), (Figure 11), combination has been identified to be the worst case. Under the considered assumptions and failure cases, the following is concluded for a high speed vehicle with a failed active lateral suspension:

1. Bumpstop stiffness is the single dominating parameter to keep track forces below limit values.

2. With soft bumpstop the influence of all other parameters is small.

3. With medium bumpstop, primary longitudinal and lateral stiffness as well as other studied parameters may influence the track shift forces.

4. Low track shift forces are achieved for medium bumpstop and soft primary suspension (as Regina 250), by avoiding flange contact for the outer wheel.

5. For slightly higher primary suspension stiffness (24%) and medium bumpstop stiffness, the track shift forces become worse than those for stiff ones. This is due to the fact that the vehicle with stiff suspension tends to follow the outer rail rather than bouncing into it.

6. High vertical forces have been observed for some of the simulated cases due to the vertical primary bumpstop contact.

7. Increasing the primary vertical damping leads to reduction in vertical forces, by avoiding the primary vertical bumpstop contact.
9 Recommendations

From the above the following is recommended for a high speed vehicle with a failed active lateral suspension:

1. Soft lateral bumpstop, at the working point, is the most effective way to maintain the running safety of the vehicle. Hence it is recommended to maintain the lateral bumpstop stiffness as soft, at the working point.

2. With medium bumpstop a careful choice of other parameters could maintain the running safety of the vehicle.

3. No recommendation can be made on the choice of primary vertical damping as the study on primary vertical dampers is inconclusive. However, it is clear that increasing the primary vertical damping leads to reduction in vertical forces, by avoiding the primary vertical bumpstop contact.
10 Future work

In order to achieve a more general perspective more work needs to be done. The current results are based on a particular high speed train, V300 Zefiro, under certain conditions and assumptions (discussed above). This cannot represent the wide selection of high speed trains that exist and will exist in the future. In order to improve the research the following is proposed to be included in the future work:

1. A study needs to be conducted to include track transition for the analysis.
2. The methodology used in this report could be used to study the vehicle safety for other active suspension systems.
3. The effect of primary damper on track shift forces needs to be studied further.
4. The influence of different rail and wheel profiles need to be studied as well.
5. Since the simulations were carried out with the assumption of single point contact between rail and wheel, which is not always the case, simulations with two point contact need to be conducted.
6. An on track test to verify the results of simulations would further strengthen the conclusions.
7. The study could be conducted on other high speed trains including “Regina” train and results could be compared.

Including the track transition in the study could reveal an entire new picture as the vehicle is highly unstable during the transitions. Since a single point contact between rail and wheel is not always the case, simulations with two point contact between rail and wheel could reveal some interesting unknown results. Moreover an on track test would be highly beneficial to compare the simulation results with actual reality. However, it would be very expensive.
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