Multibody simulation model for freight wagons with UIC link suspension

by

Per-Anders Jönsson
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Preface and acknowledgements

The work reported in this doctoral thesis has been carried out as part of a research programme on vehicle-track interaction (called SAMBA) at the Royal Institute of Technology (KTH), Division of Rail Vehicles. The aim of the present work is to investigate the dynamic performance of freight wagons.

The financial and personal support from the Swedish National Rail Administration (Banverket), Bombardier Transportation (Sweden), Green Cargo and Interfleets Technology (Sweden) is gratefully acknowledged.

For the on-track tests additional support was received from Green Cargo, Interfleets Technology, Kockums Industrier, Dellner Dampers and Tikab Strukturnmekanik.

The support from Mr. Ingemar Persson at DEsolver regarding the multibody simulation software GENSYS is also gratefully acknowledged.

Finally, I would like to thank my colleagues at the division, in particular my supervisors Dr.-Ing. Sebastian Stichel and Prof. Evert Andersson as well as Prof. Mats Berg.

Stockholm, June 2007

Per-Anders Jönsson
Abstract

The previous freight wagon model developed at KTH is able to explain many of the phenomena observed in tests. In some cases, however, simulated and measured running behaviour differ. Therefore, in this paper a new simulation model is presented and validated with on-track test results. The performance of standard two-axle freight wagons is investigated. The most important parameters for the running behaviour of the vehicle are the suspension characteristics. The variation in characteristics between different wagons is large due to geometrical tolerances of the components, wear, corrosion, moisture or other lubrication. The influence of the variation in suspension characteristics and other parameters on the behaviour of the wagon on tangent track and in curves is discussed. Finally, suggestions for improvements of the system are made.
1 INTRODUCTION

1.1 Background

Running gear on the freight wagon fleet in Sweden as well as in the rest of Western Europe are usually one of the three UIC standard types, c.f. Figure 1:

- Y25 bogie.
- Link suspension bogie (G-type).
- Two-axle wagons with UIC double link suspension.

Wagons with these types of running gear exist in large numbers and will continue to be the backbone of the European rail freight transport system for a foreseeable future. Improving ride quality and increase axleload, loading gauge and speed are desirable in order to make rail freight traffic more competitive.

However, a majority of the traffic related cost for track deterioration originates from freight traffic. As heavier and faster freight trains are introduced the cost for track maintenance is likely to increase, at least if freight wagons with standard running gear are used and the track is not strengthened. In Sweden for example 25 tonnes and in some cases 30 tonnes axle load is tested on several lines, higher and wider wagons are introduced and mail services at speeds up to 160 km/h have been operating for several years.

Figure 1: Standard types of running gear.

1.2 Wheel-rail contact conditions

The wheel-rail interface is important for the dynamic interaction between vehicle and track. In Europe there are different approaches on how the interface should be managed. Wheel and rail profiles and rail inclination differs from country to country, e.g.,

- 1:20 rail inclination - UK, France, Italy,
- 1:40 rail inclination - Germany, Austria, Switzerland,
For freight wagons in international traffic it is important that they are designed to perform well for all variable wheel-rail contact conditions that can occur. The equivalent conicity is often used to characterise the wheel-rail interface. In Figure 2 a comparison between the S1002 and P8 wheel profiles on rails with 1:20 respectively 1:40 inclination is shown. The effect of variation in track gauge is also shown. The equivalent conicity for the S1002 wheel profile on rails with 1:20 inclination can be low and can cause bad ride comfort for freight wagons. Due to wear and maintenance operations the variation in contact conditions can be even greater. When testing vehicles for approval track sections may be disregarded if, for a maximum speed of 140 km/h, the equivalent conicity exceeds 0.5 [20]. This limit value is indicated in Figure 2. Contact geometry also determines the size of the contact patch and in turn the contact stresses which also are important for cost and safety of rail freight traffic.

![Figure 2: Wheel-rail contact conditions. Comparison between S1002 and P8 wheel profiles on UIC60 rail profiles with various inclination.](image)

1.3 This study

The overall aim with this study is to investigate the dynamic performance of two-axle freight wagons with double-link suspension. A multibody dynamic simulation (MBS) model is developed and the simulation results are compared with on-track test results and other simulation results. A new link suspension model is introduced that gives better results compared to measured vehicle behaviour than the previous model used at.

- In Chapter 2 the variation in lateral and longitudinal suspension characteristics is discussed.
The new multibody simulation model is described in Chapter 3.

Validation of the model is shown in Chapter 4.

Running behaviour on tangent track is discussed in Chapter 5.

Curving performance is shown in Chapter 6.

Overall behaviour is discussed in Chapter 7.
Section 1 - Introduction
2 SUSPENSION CHARACTERISTICS

2.1 UIC double link suspension

The main components of the UIC double link suspension system are the leaf spring, links and bearings as shown in Figure 3. The carbody is connected to the leaf spring via the suspension links. The system allows vertical, lateral and longitudinal relative motions between the axlebox and carbody. The longitudinal and to some extend the lateral motions are limited by the axle guard. The model for the vertical, lateral and longitudinal suspension and the horizontal bump stops are described in Chapter 3.

Figure 3: The UIC double-link suspension for two-axle wagons. a) Side view. b) Close up off double-links. c) View of links and bearings. (1) carbody, (2) wheelset, (3) leaf spring, (4) axle guard, (5) end bearing, (6) link, (7) intermediate bearing, (8) link pin.

2.2 Vertical characteristics

Parabolic as well as trapezoidal leaf springs are used for the vertical suspension. Both types exist as single and two-stage (progressive) springs. Typical force-displacement characteristics are shown in Figure 4.

Figure 4: Example hysteresis diagram single-stage and two-stage leafspring.
Section 2 - Suspension characteristics

The vertical characteristics of freight wagon leaf springs have been investigated by ORE and reported in [17] and [18] for trapezoidal and parabolic leaf springs respectively. The description of the vertical stiffness and kinematics for a trapezoidal spring follow these reports.

2.3 Lateral and longitudinal characteristics

The most important parameters for the running behaviour of the vehicle are the suspension characteristics. Measurements of lateral and longitudinal force-displacement characteristics are therefore performed on several two-axle freight wagons with link suspension running gear. Samples of the lateral measurements are shown in Figure 5a). We observe the typical non-linear behaviour with a considerable hysteresis. The variation of characteristics between different wagons is significant. Reasons for the variation are differences in contact conditions between link and bearing caused by geometrical tolerances of the components, wear, corrosion, moisture or other lubrication. In laboratory tests it is observed that the energy dissipation, i.e. the amount of hysteresis after a few hours of dynamic testing on new links and bearings, is considerably lower than what initially was measured. It is also observed that carbon is migrating from the components as the initial surface roughness is worn down and it is the authors’ opinion that the lubrication from the carbon can have caused the observed behaviour with low energy dissipation in the laboratory tests [9].

Several authors have investigated the characteristics of the link suspension [10],[21],[13] and [6].

In Table 1 lateral and longitudinal suspension parameters from different sources are shown. Some of them are measured. Others are calculated from mathematical models of the link suspension. As comparison are the values calculated with the model according to [21] are shown as well (cf. source 7 in Table 1). This model is derived from a
geometrical representation of the suspension assuming cylindrical shape of the components in contact. The values shown in the table are obtained with nominal dimensions of the components and with a value of 0.3 for the coefficient of friction in the contact between link pins and bearings.

**Table 1:** Longitudinal and lateral suspension characteristics for 2-axle freight wagons with link suspension. Parameters from different sources. $k_1$, $k_2$ and $F_d$ are defined in Figure 5b). The values are normalized with the vertical axle box load.

<table>
<thead>
<tr>
<th>Source*</th>
<th>Nr. of measure</th>
<th>$k_1$/Fbox [1/m]</th>
<th>$k_2$/Fbox [1/m]</th>
<th>$F_d$/Fbox [1]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Max</td>
<td>Min</td>
<td>Average</td>
<td>σ</td>
</tr>
<tr>
<td>Longitudinal direction</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. 8</td>
<td>42.9</td>
<td>10.6</td>
<td>19.2</td>
<td>12.4 (64%)</td>
</tr>
<tr>
<td>4. -</td>
<td>-</td>
<td>-</td>
<td>23.0</td>
<td>-</td>
</tr>
<tr>
<td>5. -</td>
<td>-</td>
<td>-</td>
<td>51.8</td>
<td>-</td>
</tr>
<tr>
<td>6. -</td>
<td>-</td>
<td>-</td>
<td>7.7</td>
<td>-</td>
</tr>
<tr>
<td>7. -</td>
<td>-</td>
<td>-</td>
<td>7.7</td>
<td>-</td>
</tr>
<tr>
<td>Lateral direction</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. 8</td>
<td>54.2</td>
<td>22.3</td>
<td>34.1</td>
<td>11.4 (33%)</td>
</tr>
<tr>
<td>2. 16</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>3. 4</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>4. -</td>
<td>-</td>
<td>-</td>
<td>13.8</td>
<td>-</td>
</tr>
<tr>
<td>5. -</td>
<td>-</td>
<td>-</td>
<td>60.0</td>
<td>-</td>
</tr>
<tr>
<td>6. -</td>
<td>-</td>
<td>-</td>
<td>8.0</td>
<td>-</td>
</tr>
<tr>
<td>7. -</td>
<td>-</td>
<td>-</td>
<td>10.4</td>
<td>-</td>
</tr>
</tbody>
</table>

* 1. Measurements performed 1997 and 2004 by KTH in the present research project.
  2. Measurements within ORE B56 [15].
  3. Measurements within ORE DT30 [16].
  4. Data used in simulations by INRETS [2].
  5. Data used in simulations by Alstom [23].
  6. Data used in simulations by DTU [7].
  7. Theoretical model of the UIC double link suspension by Piotrowski [21].

We observe a large variation of stiffness values between the different wagons.

The tests in the lateral direction according to source 2 in Table 1 were performed with new respectively worn links. Eight new and eight suspensions with worn components were tested. The difference between new and worn links from these tests are shown in Table 2. The break-out force $F_d$ increases 25-30% and the pendulum stiffness $k_2$ decreases 15-20% when the links are worn. The standard deviation of the measured characteristics is slightly less for the worn links. Probably changed geometry due to wear.
Section 2 - Suspension characteristics

is responsible for these differences. The effective pendulum length, which is inverse proportional to the pendulum stiffness, \( L_{\text{eff}} = \frac{F_{\text{box}}}{k_2} \) is 139 mm for new links and 169 mm for worn components. From Figure 6 we can derive the following geometrical expression for the effective lateral pendulum length:

\[
L_{\text{eff}} = (L_0 + 2 \cdot R) \cdot \cos(\alpha)
\]  

(8)

Table 2: Longitudinal and lateral suspension characteristics for 2-axle freight wagons with link suspension. Parameters from different sources. Stiffness and force are definitions in Figure 5b). The values are normalized with the vertical axle box load.

<table>
<thead>
<tr>
<th>Source*</th>
<th>Note</th>
<th>( k_2/F_{\text{box}} ) [1/m]</th>
<th>( F_{\text{box}}/k_2 ) [mm.]</th>
<th>( F_d/F_{\text{box}} ) [1]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Max</td>
<td>Min</td>
<td>Average</td>
<td>σ</td>
</tr>
<tr>
<td>Lateral direction</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>New links and bearings</td>
<td>10.8</td>
<td>3.3</td>
<td>7.2</td>
</tr>
<tr>
<td>2.</td>
<td>Worn links and bearings</td>
<td>8.3</td>
<td>3.9</td>
<td>5.9</td>
</tr>
</tbody>
</table>

Figure 6: Lateral pendulum length in the double-link suspension.

For new ideal components with \( L_0 = 140 \) mm, \( R = 12.5 \) mm and \( \alpha = \sin(150/288) = 31.4^\circ \) is \( L = 141 \) mm which in average corresponds well with the measured data. The considerable difference for the worn components needs some further explaining.
At overhaul in Sweden the length of the double-link suspension is measured. The nominal measure is 288 mm, c.f. Figure 6. When the length is more than 302 mm the links and bearings are exchanged. As the longitudinal dimension in the suspension is fix this will reduce the angle $\alpha$. Investigating worn components further reveals that $L_0$ and $R$ increases for worn components [10]. This can however; not fully explain the difference in suspension characteristics between new and worn components. Examining worn links we find that the bearings are not positioned in the centre of the link. Hence; the nominal gap of 5 mm between link and bearing, shown in Figure 6, is fully closed. This will further decrease the link inclination angle $\alpha$.

In the longitudinal direction the contact surface of the end bearings take a more cylindrical shape as they are worn. The reason for this is that the link pins, cf. number (8) in Figure 3, are machined to a cylindrical shape and manufactured from a material harder than the bearings; hence it is mainly the contact surface of the end bearings that are worn and the surface is formed by the pin.

In Table 3 typical values for lateral and longitudinal suspension characteristics are defined. Case 1, 3 and 5 respectively refer to min, average and maximum hysteresis loops and the values are set to reflect variation in measured suspension characteristics of which some measurements are shown in Table 1. The parameters in Table 3 are used in the parameter studies in the following chapters of this report.

**Table 3:** Typical longitudinal and lateral suspension characteristics for 2-axle freight wagons with link suspension. Stiffness and force are defined in Figure 5b). The values are normalized with the vertical axle box load.
Section 2 - Suspension characteristics
3 SIMULATION MODEL

The multibody simulation (MBS) model of the two-axle freight wagon consists of carbody, leafspring, wheelset, rail and track. The carbody is considered flexible whereas the other bodies are rigid. The wheel-rail contact is non-linear, both with respect to the wheel-rail geometry and the creep-creep force relations. The suspension, carbody and track models are described in Sections 3.1-3.3.

3.1 Coupling elements

The non linear force displacement characteristics present in the leaf spring link suspension is caused by the friction in the suspension elements. Several authors have developed models to describe friction in vehicle dynamics, see for instance Berg [3], Fancher [5] and Lange [13]. Lange proposed a linear spring in series with a Coulomb friction element, in parallel with a linear spring. Different functions where the friction force increases towards a limit value were proposed by Berg as well as Fancher.

In the model suggested here the total force over the coupling element is separated in piece wise elastic respectively friction force

\[ F = F_e + F_f. \]  \hspace{1cm} (9)

The elastic force is described by

\[ F_e = F_0 + K_e \cdot \delta, \]  \hspace{1cm} (10)

where \( F_0 \) is the static preload, \( K_e \) the stiffness and \( \delta \) the deformation over the spring.

An exponential expression similar to the one derived by Fancher is used to describe the friction force \( F_f \). However, the difference compared to Fanchers approach is that the force gradient, \( \frac{\partial F_f}{\partial \delta} \), is assumed to be constant at every point when the direction of loading is changed.

\[ F_f = F_f_1 + \frac{K_f A}{\alpha_A} \cdot (1 - e^{-\alpha_A \cdot (\delta_1 - \delta)}), \quad \delta \geq 0 \]

\[ \alpha_A = \frac{K_f A}{F_f - F_f_1}, \]  \hspace{1cm} (11)

\[ \left. \frac{\partial F_f}{\partial \delta} \right|_1 = K_f A, \]
Section 3 - Simulation model

\[ F_f = F_{f2} + \frac{kf_B}{\alpha_B} \cdot (1 - e^{-\alpha_B \cdot (\delta - \delta_2)}), \quad \delta < 0 \]

\[ \alpha_B = \frac{kf_B}{F_{fB} - F_{f2}}, \]

\[ \frac{\partial F_f}{\partial \delta} \bigg|_{\delta_2} = kf_B. \]

The characteristics described by Equation (11)-(12) is shown in Figure 7.

\[ F_f = F_{fA} \quad \delta_1 \quad \delta_2 \quad \delta_3 \]

**Figure 7:** Force-displacement characteristics for the friction force.

As mentioned earlier in the section the force gradient is the same every time the direction of the loading is changed. The reason for this assumption is that we in laboratory test always observe a hysteresis loop, even for small displacements. In Fanchers approach the force gradient is given by the following expression

\[ \frac{\partial F_f}{\partial \delta} = \frac{(F_{fA} - F_f)}{C}, \]

where \( C \) is a constant. The behaviour of the two different models are shown in principle in Figure 7. If the direction of loading is changed in point 1 the response from both models, \( \alpha_1 \) and \( \beta_1 \), form a closed loop, hence energy is dissipated. At point 2 the response from Fanchers model, \( \beta_2 \), does not form a closed loop. Furthermore the model approach is justified by physical interpretation of the link suspension. The contact between links and bearings can have three different states:
- Rolling contact (high stiffness).
- Sliding contact (low pendulum stiffness).
- Transition between rolling and sliding.
If the loading changes direction in the transition zone the state of the contact is also changed from partly sliding to pure rolling. The stiffness in the rolling contact is mainly given by the difference in rolling radius of the two components in contact, hence assuming constant stiffness for this state is reasonable independent on where in the hysteresis loop the direction of loading is changed.

A consequence is shown in Figure 8. We excite the model given by Equation (9) with a low frequency periodic triangular pulse and one superimposed with high frequency and low amplitude pulse. The hysteresis loop for the second case is larger than the first one. This is a consequence of the model approach. Similar behaviour is observed in tests on link suspensions [23].

![Figure 8: Example of force-displacement characteristics for coupling element according to Equation (9).](image)

**Figure 8:** Example of force-displacement characteristics for coupling element according to Equation (9).

a) Force-displacement characteristics.

b) Excitation signals.

### 3.1.1 Kinematics

The link inclination angle, $\alpha$, is decisive for the forces in the double links. In this section we estimate the influence of the vertical respectively longitudinal motions on the link inclination. The length and link angle for a trapezoidal leaf spring as function of spring camber, $p$ according to Figure 9, can be expressed as

\[
L = \frac{1}{2} \cdot \sqrt{L_s^2 - \frac{4}{3} \cdot (2 \cdot p - d_i)^2 - 8 \cdot (2 \cdot p - d_i) \cdot (d_i + h) + \frac{1}{4} \cdot L_s^2 - \frac{4}{3} \cdot (2 \cdot p - d_i)^2}
\]

(14)

and
Section 3 - Simulation model

\[
\tan(\alpha) = \frac{A - L}{2 \cdot I_g \cdot \sqrt{1 - \left(\frac{A - L}{2 \cdot I_g}\right)^2}} \tag{15}
\]

with

\[
p = p_0 - \delta_z. \tag{16}
\]

Figure 9: Leaf spring and link suspension.

The relation between the vertical deflection, \(\delta_z\), and spring length, \(L\), for a 9-leaf, 1200 mm spring with properties according to Table 2-1 is shown in Figure 11a).

If we simplify the system and not consider rolling contact between pins and bearings the relation between a longitudinal motion of the wheelset and the inclination angle of the links easily can be estimated. From Figure 10 we can derive

\[
x + \frac{A}{2} = l_g \cdot \sin(\alpha_2) + \frac{L \cdot \cos(\phi)}{2} + (h_0 + p) \cdot \sin(\phi), \tag{17}
\]

\[
A = l_g \cdot \sin(\alpha_1) + l_g \cdot \sin(\alpha_2) + L \cdot \cos(\phi), \tag{18}
\]

\[
\sin(\phi) = \frac{l_g \cdot (\cos(\alpha_1) - \cos(\alpha_2))}{L}. \tag{19}
\]

If we write the angles of link inclination as:
Multibody simulation model for freight wagons with UIC link suspension

\[ \alpha_1 = \alpha_0 + d\alpha_1 \]  \hspace{1cm} (20)

\[ \alpha_2 = \alpha_0 + d\alpha_2 \]  \hspace{1cm} (21)

and linearize Equations (17)-(19) assuming small displacements around \( \alpha_0 \), i.e. \( d\alpha_1 \) and \( d\alpha_2 \) are small, we can derive expressions for the relation between longitudinal displacement and link inclination

\[ \alpha_1 = \alpha_0 - \frac{\cos(\alpha_0) \cdot \sin(\phi) \cdot L + 2 \cdot l_g - 2 \cdot l_g \cdot \cos(\alpha_0)^2}{2 \cdot \cos(\alpha_0) \cdot \sin(\alpha_0) \cdot l_g} + \frac{L \cdot \cos(\phi) \cdot \sin(\alpha_0) - A \cdot \sin(\alpha_0)}{2 \cdot \cos(\alpha_0) \cdot \sin(\alpha_0) \cdot l_g}, \]  \hspace{1cm} (22)

\[ \alpha_2 = \alpha_0 - \frac{-\cos(\alpha_0) \cdot \sin(\phi) \cdot L + 2 \cdot l_g - 2 \cdot l_g \cdot \cos(\alpha_0)^2}{2 \cdot \cos(\alpha_0) \cdot \sin(\alpha_0) \cdot l_g} + \frac{L \cdot \cos(\phi) \cdot \sin(\alpha_0) - A \cdot \sin(\alpha_0)}{2 \cdot \cos(\alpha_0) \cdot \sin(\alpha_0) \cdot l_g}, \]  \hspace{1cm} (23)

\[ x = l_g \cdot \sin(\alpha_0) + l_g \cdot \cos(\alpha_0) \cdot d\alpha_2 + \frac{L \cdot \cos(\phi)}{2} + (h_0 + p) \cdot \sin(\phi). \]  \hspace{1cm} (24)

Figure 10: Principlesketch - link suspension.
In Figure 11b) the relation between spring camber and link inclination according to Equation (15) is shown. The difference in link inclination for a tare to laden vertical motion is rather moderate, approximately 1.5°. However, for a longitudinal motion the difference in link inclination is considerable. This is further discussed in Section 3.1.5.

**Figure 11:**  
a) Length of spring as function of vertical spring deflection according to Equation (14).  
b) Inclination of suspension link as function of spring deflection.  
   *Solid line* - Vertical motion according to Equation (15).  
   *Dashed* - Longitudinal motion according to Equation (24), tare.  
   *Dash-dot* - Longitudinal - laden.

**Table 2-1**  

<table>
<thead>
<tr>
<th>Notation</th>
<th>L_s</th>
<th>n</th>
<th>h</th>
<th>b</th>
<th>d_i</th>
<th>p_0</th>
<th>C_a</th>
<th>C_b</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring length.</td>
<td>1400</td>
<td>9</td>
<td>16</td>
<td>120</td>
<td>36</td>
<td>79</td>
<td>0.944</td>
<td>0.910</td>
</tr>
<tr>
<td>Number of leaves.</td>
<td>1200</td>
<td>9</td>
<td>16</td>
<td>120</td>
<td>36</td>
<td>46</td>
<td>0.586</td>
<td>0.555</td>
</tr>
<tr>
<td>Leaf thickness.</td>
<td>1200</td>
<td>8</td>
<td>16</td>
<td>120</td>
<td>36</td>
<td>64</td>
<td>0.662</td>
<td>0.631</td>
</tr>
<tr>
<td>Leaf width.</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm/kN</td>
<td>mm/kN</td>
</tr>
</tbody>
</table>
3.1.2 Vertical suspension

The vertical stiffness of the leaf spring is influenced by the double-link arrangement, see Figure 12. The reasons for this are changes in link inclination and elastic deformations in the components with increased load. Based on test results ORE derived the following expression for the relation between vertical spring flexibility measured in rolling carriage and double link arrangement

\[
\frac{C_a}{C_z} = 1.0 + 1.9 \cdot 10^{-3} \cdot (p - \frac{d_i}{2}),
\]

where spring camber, \(p\), and diameter of spring eye, \(d_i\), are given in mm. For the definition of \(C_a\) and \(C_z\) cf. Table 2-1.

The ORE tests also conclude that the influence of friction on the relation between the nominal flexibility and upper envelop of the hysteresis loop, \(C_b/C_a\), is 0.92 - 0.96 for new springs and 0.85-0.87 for ungreased reconditioned springs. The damping forces \(Ff_A\) and \(Ff_B\) respectively stiffness \(K_e\) in Equations (11)-(12) are calculated as

\[
Ff_A = \left(1 - \frac{C_b}{C_a}\right) \cdot F_z,
\]

\[
Ff_B = -\left(1 - \frac{C_b}{C_a}\right) \cdot F_z,
\]

\[
K_e = \frac{K_{ar}}{1.0 + 1.9 \cdot 10^{-3} \cdot (p - \frac{d_i}{2})},
\]

where \(F_z\) is the vertical force in the spring.

<table>
<thead>
<tr>
<th>Notation</th>
<th>(C_z)</th>
<th>(K_{ar})</th>
<th>(C_b/C_a)</th>
<th>(C_z/C_a)</th>
</tr>
</thead>
<tbody>
<tr>
<td>mm/ kN</td>
<td>1.018</td>
<td>1.059</td>
<td>0.964</td>
<td>1.078</td>
</tr>
<tr>
<td>Mean spring flexibility (in link suspension)</td>
<td>0.633</td>
<td>1.706</td>
<td>0.948</td>
<td>1.080</td>
</tr>
<tr>
<td>Stiffness ((1/C_a))</td>
<td>0.718</td>
<td>1.511</td>
<td>0.954</td>
<td>1.085</td>
</tr>
</tbody>
</table>
3.1.3 Suspension in the horizontal plane

The coupling between carbody and wheelset in the horizontal plane is modelled via one longitudinal element and two lateral elements at each axlebox. The lateral elements connect the carbody and leafspring and are placed in the position where the double-links are attached to the carbody, see Figure 10. The leafspring is an own rigid body in the MBS model and is connected to the wheelset via constraint restricting all motions except yaw between leafspring and wheelset. This is to limit the number of DOF’s in the model and to avoid high eigenfrequencies in the system. The coupling in yaw is modelled via a rotational spring in series with a friction slider, cf. section 3.1.6. The longitudinal and lateral models are described further in Section 3.1.4-3.1.5.
Figure 13: Horizontal coupling of wheelset and carbody. Principle sketch.

Lateral forces and resulting moment acting between leafspring and wheelset are shown in Figure 14 and can be derived accordingly

\[ F_{y12} = F_{1y} + F_{2y}, \]  
\[ F_{y34} = F_{3y} + F_{4y}, \]  
\[ M_{ls12} = (F_{1y} - F_{2y}) \cdot a_{al}, \]  
\[ M_{ls34} = (F_{3y} - F_{4y}) \cdot a_{al}. \]
Section 3 - Simulation model

Figure 14: Coupling forces.
   a) Forces between carbody and leafspring.
   b) Lateral forces and yaw moment on leafsprings.

Forces and moments on the wheelset write as

\[
F_x = F_{x12} + F_{x34} = 0, \quad (33)
\]

\[
F_y = F_{y12} + F_{y34} = 0, \quad (34)
\]

\[
M = M_{ls12} + F_{x12} \cdot b_l + M_{ls34} - F_{x34} \cdot b_l = 0. \quad (35)
\]

The displacements in the connection points for the carbody can be written as:

\[
x_{12c} = x_c - b_l \cdot \psi_c, \quad (36)
\]

\[
x_{34c} = x_c + b_l \cdot \psi_c, \quad (37)
\]

\[
y_{1c} = y_c + (a_{ca} + a_{al}) \cdot \psi_c, \quad (38)
\]
Multibody simulation model for freight wagons with UIC link suspension

\[ y_{2c} = y_c + (a_{ca} - a_{al}) \cdot \psi_c, \quad (39) \]

\[ y_{3c} = y_c - (a_{ca} + a_{al}) \cdot \psi_c, \quad (40) \]

\[ y_{4c} = y_c - (a_{ca} - a_{al}) \cdot \psi_c. \quad (41) \]

For the wheelset the displacements write as

\[ x_{12ws} = x_{ws} - b_l \cdot \psi_{ws}, \quad (42) \]

\[ x_{34ws} = x_{ws} + b_l \cdot \psi_{ws}, \quad (43) \]

\[ y_{1ws} = y_{ws} + a_{al} \cdot \psi_{ws} + a_{al} \cdot \psi_{ls12}, \quad (44) \]

\[ y_{2ws} = y_{ws} - a_{al} \cdot \psi_{ws} - a_{al} \cdot \psi_{ls12}, \quad (45) \]

\[ y_{3ws} = y_{ws} + a_{al} \cdot \psi_{ws} + a_{al} \cdot \psi_{ls34}, \quad (46) \]

\[ y_{4ws} = y_{ws} - a_{al} \cdot \psi_{ws} - a_{al} \cdot \psi_{ls34}, \quad (47) \]

where \( x_c, y_c, \psi_c, x_{ws}, y_{ws} \) and \( \psi_{ws} \) are the longitudinal, lateral and yaw displacements for carbody and wheelset respectively, and the yaw angles of the leafsprings \( \psi_{ls12} \) respectively \( \psi_{ls34} \).

If we only consider small displacements the suspension characteristics can be linearized and the forces in the coupling elements can be written as

\[ F_{1y} = F_z \cdot k_y \cdot (y_{1ws} - y_{1c}), \quad (48) \]

\[ F_{2y} = F_z \cdot k_y \cdot (y_{2ws} - y_{2c}), \quad (49) \]

\[ F_{3y} = F_z \cdot k_y \cdot (y_{3ws} - y_{3c}), \quad (50) \]
where $k_x$ and $k_y$ are the longitudinal respectively lateral stiffness normalized with the vertical load on the axlebox $F_z$.

If we consider the case where the break-out friction moment in the coupling between leafspring and axle-box is greater than the yaw moment acting on the leaf spring, i.e. $\psi_{ls12}$ and $\psi_{ls34}$ are zero, and insert Equations (29)-(32) and (36)-(53) in Equations (33)-(35) we can write the forces from the coupling elements on the wheelset as

$$ F_{4y} = F_z \cdot k_y \cdot (y_{4ws} - y_{4c}), \quad (51) $$

$$ F_{12x} = F_z \cdot k_x \cdot (x_{12ws} - x_{12c}), \quad (52) $$

$$ F_{34x} = F_z \cdot k_x \cdot (x_{34ws} - x_{34c}), \quad (53) $$

In the expression for the yaw stiffness for the wheelset in Equation (54) we can see that there is one additional term, $4k_y \cdot a_{al}^2$, stabilising the wheelset when the longitudinal dimensions of the leafspring and the friction moment in the coupling between leafspring and axlebox are considered. This term is not included in other models that are found in literature.

### 3.1.4 Longitudinal coupling

The longitudinal force in the coupling between axlebox and carbody is linearly depending on the vertical force on the axlebox $F_z$. The force is calculated using Equations (9)-(12) and the expression

$$ F_x = F_{x0} \cdot \frac{F_z}{F_{z0}}, \quad (55) $$

where $F_z$ is the vertical force on the axlebox and $F_{x0}$ is the longitudinal force corresponding to the vertical force $F_{z0}$.
3.1.5 Lateral coupling between carbody and leafspring

The lateral coupling between car and leafspring is modelled with two elements according to Equations (9)-(12). The lateral characteristics is linearly dependent on the forces acting in the direction of the double-links, i.e. forces $F_1$ and $F_2$ in Figure 15.

![Figure 15: Relations between forces in links and vertical respectively longitudinal forces in the UIC double-link suspension.](image)

From Figure 15 we can derive the following equations of equilibrium:

\[
X: \quad F_{x1} - F_{x2} + F_x = 0 \tag{56}
\]

\[
Z: \quad F_{z1} + F_{z2} - F_z = 0 \tag{57}
\]

\[
x: \quad -F_{z1} \cdot A + F_z \cdot \left(\frac{A}{2} + x\right) - F_x \cdot (H_{box} + H_{spring} - l_s \cdot \cos(\alpha_0)) = 0 \tag{58}
\]

In order to get a simple expression easy enough to be used in multibody dynamic simulations we assume that the relation between the longitudinal forces can be written as

\[
F_{x1} = F_{x0} - \frac{F_x}{2} \tag{59}
\]

\[
F_{x2} = F_{x0} + \frac{F_x}{2} \tag{60}
\]
where $F_{x0}$ is the longitudinal component of the force in the double-links when $F_x=0$.

From Equations (57) and (58) the vertical components of the forces are found

\[
F_{z1} = \frac{F_z \cdot \left(\frac{A}{2} + x \right) - F_x \cdot (H_{box} + H_{spring} - l_g \cdot \cos(\alpha_0))}{A}, \quad (61)
\]

\[
F_{z2} = \frac{F_z \cdot \left(\frac{A}{2} - x \right) + F_x \cdot (H_{box} + H_{spring} - l_g \cdot \cos(\alpha_0))}{A}, \quad (62)
\]

\[
F_1 = \sqrt{F_{x1}^2 + F_{z1}^2}, \quad (63)
\]

\[
F_2 = \sqrt{F_{x2}^2 + F_{z2}^2}. \quad (64)
\]

3.1.6 Coupling between leaf spring, axlebox and axle

The leaf spring is standing on top of the axlebox. Longitudinal and lateral relative movements are restricted as a vertical pin on the leaf spring is mounted in a hole in the top surface of the axlebox, see Figure 16.

The x, y and z translation and the roll motions of the leaf spring are coupled to the motions of the wheelset via constraints. The coupling in yaw is modelled via a rotational spring in series with a friction slider. The contact area between leaf spring and axlebox is square, 100x120 mm, and the diameter of the vertical pin $\phi_d=50$ mm. The break-out friction moment is calculated as:

\[
M_{frz} = \mu_{lsa} \cdot r \cdot F_z \quad (65)
\]

where $\mu_{lsa}$ is the coefficient of friction between leaf spring and axlebox, $r$ is the average radius in the contact surface and $F_z$ is the vertical force on the axlebox.
3.1.7 Lateral and longitudinal bumpstops

Lateral and longitudinal motions between carbody and wheelset are restricted via mechanical stops. The axle guard, shown in Figure 3, consists of two steel plates that are attached to the carbody and connected below the axlebox via a tie bar. The main reason for use of the tie bar is to maintain the wheelset close to the carbody in case of derailment. The principle of the bumpstops is shown in Figure 17a). When the lateral or longitudinal play is exceeded the stops becomes active. The lateral and longitudinal play in the suspension is approximately 20 mm. The play, however, can differ considerably due to wear and plastic deformation of the axle guard.

The bumpstops are modelled via non-linear springs connecting the wheelset and carbody as shown in Figure 17b). Properties are obtained from suspension characteristics measurements. The lateral flexibility is mainly obtained through bending of the axle guard. In the longitudinal direction the axle guard is very stiff. Here the flexibility is mainly caused by the flexibility of the connection between carbody and axle guard. The stiffnesses in the bump stops are set to \( k_x=10 \) [MN/m] and \( k_y=1.5 \) [MN/m]. When the bump stop becomes active the stiffnesses initially are lower as shown in Figure 17b).
Section 3 - Simulation model

3.1.8 Lateral play in bearings

In order to accommodate variation in temperature a lateral play is needed in the bearings. This is realized through a dead band spring laterally connecting the leafspring and axlebox. However, if the lateral play is zero the leafspring is connected to the wheelset via a lateral constraint, cf. Section 5.9.

3.1.9 Previous simulation model

In the following chapters comparisons with a simulation model developed by Lange [13] are shown. This model is in the following text referred to as the previous model.

The main differences between the new and previous model are the type of suspension element used and position of the lateral elements. Lange used a linear spring in series with a friction element in parallel with a linear spring to describe the hysteresis loop. The influence of different types of coupling elements is investigated in [11], and the effect is small, at least in comparison to the effect of the large variation in suspension characteristics. Therefore only the influence of the longitudinal position and friction yaw moment is investigated in this chapter. For the previous model the longitudinal distance $a_{sl}$ in Figure 18 and the break-out friction in the yaw coupling between the leafspring and axlebox are set to zero.

---

Figure 17: Lateral and longitudinal stop.

a) Top view [13].

b) Suspension model.
3.2 Carbody and Load

The flexible properties of the carbody are considered and the eigenmodes of the first torsional and first bending eigenmodes are incorporated in the carbody model.

Primary data to a simulation are the axleload and height of carbody mass centre. The mass of the carbody is calculated as

\[ m_{\text{car}} = 2 \cdot (\text{AXLELOAD} \cdot 1000 - m_{\text{ws}} - 2 \cdot m_{\text{ls}}), \]  

where \( \text{AXLELOAD} \) is the axleload in tonnes, \( m_{\text{ws}} \) and \( m_{\text{ls}} \) masses of the wheelset and leafspring respectively. The height of the carbody is given by

\[ H = 2 \cdot (h_{\text{ccg}} - h_{\text{low}}), \]  

where \( h_{\text{ccg}} \) is the height of the carbody mass centre and \( h_{\text{low}} \) the height from top of rail to the lower parts of the carbody that significantly contribute to the mass of the car. For the calculation of inertia moments the masses of the carbody and the load are evenly distributed in a box with dimension \( L \times W \times H \),

\[ J_{\text{cxy}} = \frac{m_{\text{car}}}{12} \cdot (H^2 + W^2), \]  

\[ J_{\text{cyy}} = \frac{m_{\text{car}}}{12} \cdot (L^2 + H^2), \]  

\[ J_{\text{czz}} = \frac{m_{\text{car}}}{12} \cdot (L^2 + W^2). \]
3.3 Track

The track is a so called moving track model. Three rigid masses are located under each wheelset connected via linear springs and viscous dampers, c.f. Figure 19. The track model introduces 5 DOFs per wheelset. The non-linear wheel-rail geometry is precalculated within GENSYS and the creep forces are interpolated from a four-dimensional table generated using the FASTSIM algorithm of Kalker.

Figure 19: Track model.
4 VALIDATION OF SIMULATION MODEL

The simulation model is validated by comparing simulation results with on-track tests and with simulation results from other authors.

4.1 Comparison between simulations and on-track measurements

In August 2004 on-track acceleration measurements on two different types of freight wagons with link suspension were carried out.

- Two axle wagon, Littera *Kbps*\textsuperscript{741}.
- Bogie wagon Littera *Rs*\textsuperscript{691}.

The test were performed within the SAMBA1-project and the two-axle wagon was equipped with a yaw-damper arrangement and lateral hydraulic axlebox dampers as shown in Figure 20a). Axlebox adapters were designed specifically for the test in order to mount the dampers to the axlebox. The yaw-damper arrangement consists of a hydraulic damper and a linkage connecting the two axleboxes and the damper is active only for a yaw motion of the wheelset. A similar yaw-damper design has been used on high speed freight wagons in Germany and Holland [14]. The bogie wagon is equipped with lateral and longitudinal primary dampers and secondary yaw-dampers. This arrangement is shown in Figure 20b).

Tests were performed with empty and loaded wagons and at speeds between 100 and 170 km/h. Also different hydraulic damper configurations were tested. In this report only results concerning the *Kbps* wagon without supplementary hydraulic dampers are presented.

![Figure 20: Axlebox adapter, lateral damper and yaw damper arrangement.](image)

a) Two axle wagon.

b) Bogie wagon.

The wagons were loaded with steel weights to 18 tonnes axleload. The height of the combined mass centre for carbody and load is 1.3 meter. The loading is shown in Figure 21. The axleload for the loaded testcase was chosen with respect to the high speeds and
acceleration respectively breaking capacity of the test train. Hence, the axleload was not chosen with respect to the vertical and lateral dynamic performance of the wagons.

Vertical respectively lateral accelerations were measured in the carbody above the centre of the wheelsets. Also lateral accelerations on the axleboxes and bogiframes were measured.

**Figure 21:**  *Loaded bogie wagon.*

The wagon had newly reprofiled S1002 wheels with a flange thickness of 30-31 mm and an inner distance between the wheels of 1458 mm respectively 1459 mm. Wheel profiles were measured by Interfleet Technology with the SPAK profile measuring device. Also lateral and longitudinal suspension characteristics were measured.

The track selected for the on-track tests is located between Uppsala and Gävle, between km 13 and km 28, consisting mainly of tangent sections and large radius curves. The rails are mainly continuously welded BV50 rails (50 kg/m) with an inclination of 1/30. However, on some sections UIC60 rails were used. The rails are mounted with Pandrol fastening to concrete sleepers via 10 mm thick rubber pads. The sleeper distance is 0.65 m. The track geometry, track irregularities and rail profiles were measured by Banverket with the STRIX-recording coach in July 2004. The rails are worn and the geometry deviates considerably from the nominal rail profile. The track gauge varies between 1433 and 1438 mm and track geometry quality is QN2 for speeds up to 160 km/h according to prEN 14636 [20]. Track design data and irregularities were transformed to a GENSYS format and implemented as excitation to the vehicle model.

The results were evaluated on ten approximately 500 meter long sections, seven tangent track and three curve sections. Contact point functions for the wheel and rail profile combinations were determined for each track section combining the measured wheel and rail geometry. The equivalent conicity according to UIC519 [25] for all track sections is in the range of 0.01-0.1 for an amplitude of ±3 mm.

In Figure 9 examples of time series and power spectra of lateral and vertical accelerations are shown. The time series are low-pass filtered at 10 Hz. The agreement is good with regards to amplitudes and the hunting frequency (approximately 1.8 Hz) is clearly shown in test results as well as the simulation results. The agreement in the power
spectra is good up to frequencies of 12-15 Hz. The main reason for the deviation above 15 Hz is that track irregularities with short wavelength are not considered in the simulations. Further only the two first flexible eigenmodes of the carbody are considered.

From the comparison with the measured results it can be concluded that the model represents reality quite well, at least in the frequency range up to 15-20 Hz.

Figure 22: Comparison between measured and simulated carbody accelerations above trailing wheelset. Loaded wagon. 18 tonnes axleload. Speed 100 km/h.
   a) Time series - Lateral acceleration.
   b) Time series - Vertical acceleration.
   c) Power spectra - Lateral acceleration.
   d) Power spectra - Vertical acceleration.

4.2 Comparison with simulations performed at DTU

The dynamic performance of two-axle freight wagons with link suspension has been investigated by Hoffmann [7]. Data for an empty covered wagon Littera: Hbbills\textsuperscript{310} is used and our simulation results are compared with the results of Hoffmann. The main data for the wagon are given in Table 4. The following adjustments were made to the model to achieve reasonable comparable conditions:
Section 4 - Validation of Simulation Model

- The previous primary suspension model, described in Section 3.1.9, is used, i.e.
  - The longitudinal position of the lateral coupling elements $d_{lsa}=0$.
  - The yaw friction in the yaw coupling between leavespring and axlebox is set to zero.
- A rigid carbody is used.
- The degrees of freedom for the track are removed.
- UIC60 rails with 1:30 inclination are used in combination with the S1002 wheel profile.
- The coefficient of friction between wheel and rail is 0.3.
- The reduction factors for the Kalker-coefficients are set to 1.

Data for the wagon is given in Table 4.

**Table 4: Data for the comparison, $Hbbills$** [7].

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Axleload</td>
<td>9.1</td>
<td>tonnes</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>10</td>
<td>meter</td>
</tr>
<tr>
<td>COM height</td>
<td>1.57</td>
<td>meter</td>
</tr>
<tr>
<td>Carbody mass</td>
<td>15176</td>
<td>kg</td>
</tr>
<tr>
<td>Mass inertia - roll</td>
<td>32675</td>
<td>kgm²</td>
</tr>
<tr>
<td>Mass inertia - pitch</td>
<td>422084</td>
<td>kgm²</td>
</tr>
<tr>
<td>Mass inertia - yaw</td>
<td>413250</td>
<td>kgm²</td>
</tr>
<tr>
<td>Axial characteristics</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$k_f_{Ax}$</td>
<td>7.67</td>
<td>[1/m]</td>
</tr>
<tr>
<td>$K_{ex}$</td>
<td>5.51</td>
<td>[1/m]</td>
</tr>
<tr>
<td>$F_f_{Ax}$</td>
<td>0.059</td>
<td>[1]</td>
</tr>
<tr>
<td>Lateral characteristics</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$k_f_{Ay}$</td>
<td>10.38</td>
<td>[1/m]</td>
</tr>
<tr>
<td>$K_{ey}$</td>
<td>3.41</td>
<td>[1/m]</td>
</tr>
<tr>
<td>$F_f_{Ay}$</td>
<td>0.08</td>
<td>[1]</td>
</tr>
<tr>
<td>Vertical characteristics</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$k_f_{Az}$</td>
<td>4.9</td>
<td>[MN/m]</td>
</tr>
<tr>
<td>$K_{ez}$</td>
<td>1.1</td>
<td>[MN/m]</td>
</tr>
<tr>
<td>$F_f_{Az} / F_z$</td>
<td>13%</td>
<td>[1]</td>
</tr>
</tbody>
</table>

In Figure 23a) the own simulation results are shown and compared with the bifurcation diagram by Hoffmann [7] in Figure 23b). The speeds at the bifurcation points are given...
in Table 5. The overall agreement is good. The main reason for deviations is believed to be differences in the wheel-rail contact model. Hoffmann uses the theory by Shen-Hedric-Elkins to represent the relation between creep and creep-forces whereas the present model uses the FASTSIM algoritm by Kalker.

**Figure 23:** Validation of simulation model. Lateral amplitude of wheelset.  
a) Simulation result, with present model.  
b) Bifurcation diagram by Hoffmann [7].

**Table 5:** Comparison of speeds at bifurcation points a, b and c in Figure 23.

<table>
<thead>
<tr>
<th>Point</th>
<th>Speed Present model.</th>
<th>Speed Hoffmann.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[m/s]</td>
<td>[km/h]</td>
</tr>
<tr>
<td>a</td>
<td>12.5</td>
<td>45.0</td>
</tr>
<tr>
<td>b</td>
<td>28.1</td>
<td>101.2</td>
</tr>
<tr>
<td>c</td>
<td>39.7</td>
<td>142.9</td>
</tr>
</tbody>
</table>

4.3 Validation of the software

The multibody simulation software GENSYS [19] and the predecessor SIMFO have continuously been developed since the beginning of the 1970:s. The simulation results have on numerous occasions been compared to measured quantities and results obtained with other software. A survey of validation and comparisons is presented by Kufver [12].

A vast variety of vehicle types have over the years been investigated. In 1977 and 1984 were test and simulation results of lateral Y-forces on passenger coaches with stiff respectively resilient wheelset guidandes compared where most differences were found to be less than 10%. During the last 15 years several vehicle models, freight wagons as well as passenger vehicles, were developed and implemented in GENSYS.

In the benchmark tests from ERRI respectively Manchester is the simulation results from GENSYS compared to simulation results from ADAMS, MEDYNA, NUCARS, SIMPACK and VAMPIRE. The results are found to agree overall.
5 RUNNING BEHAVIOUR ON TANGENT TRACK

In this chapter the influence of vehicle and track parameters on the running behaviour on tangent track is studied.

An approximate method to display limit cycles is used. The running behaviour on an ideal tangent track is simulated. We start at 160 km/h and excite the vehicle with a single lateral disturbance and integrate until we have an oscillation with constant amplitude. The speed is reduced with 1 km/h and the solution in the last time instant in the first simulation is used as initial condition. This procedure is repeated until the speed reaches 50 km/h.

With this approximative method, however, it is not possible to exactly follow the solution branches. The system also can have coexisting attractors of which not all are found. Sometime multiple attractors are seen as the solution jumps from one attractor to another.

A standard configuration which represents suspension characteristics in an intermediate worn state, i.e. $k_x^3$ and $k_y^3$ according to Table 3 is used as reference. The wheel-rail contact conditions, where nothing else is stated, are given by ideal S1002 wheel profiles and BV50 rail profiles with 1 over 30 rail inclination and 1435 mm gauge. The coefficient of friction between wheel and rail is 0.4 and the so called Kalker coefficients are reduced with the factor $\mu/0.6$.

5.1 General running behaviour

The dynamic behaviour of two-axle freight wagons with links suspension is complex and is first generally discussed. Railway vehicles are so called parameter dependent systems. A typical parameter is the vehicle speed $v$ and the equilibrium solution to the system depends on the value of the parameter. As the system is non-linear several equilibrium solutions can exist for the same set of parameters. Looking at a typical bifurcation diagram for rail vehicle dynamics in Figure 24 we see that if the vehicle speed is below $v_{nlin}$, the so called non-linear critical speed, only the equilibrium solution without oscillations exists. However, if the speed is between $v_{nlin}$ and $v_{lin}$ we have two stable attractors separated by an unstable branch and the response of the system depends on the initial conditions. If the initial conditions are above the unstable branch the equilibrium solution is given by the non-zero attractor. The same applies to speeds above $v_{lin}$.

In Figure 25 results from simulations with an empty two-axle freight wagon are shown. For speeds below 73 km/h all oscillations vanish. The zero attractor exists for speeds up to around 100 km/h. At 73 km/h another attractor arises. The lateral amplitude of the wheelset declines with increasing speed. However, the frequency of the motion is relatively constant, i.e. 2.3 - 2.6 Hz. The reason for this behaviour is a resonance between the lateral excitation frequency from the wheelset and different eigenmodes in the vehicle.

For speeds above 120 km/h also the wheelset hunting mode or flange-to-flange attractor is present. The lateral motion of the wheelset is restricted by the flange contact and the frequency for this motion is between 4.5 and 6 Hz. We can see that the lateral track
forces can be considerable even though the axleload in this case is only 6.5 tonnes and the vehicle is running on an ideal track without track irregularities.

**Figure 24:** Typical bifurcation (limit cycle) diagram in rail vehicle dynamics.

**Figure 25:** Simulation results with an empty freight wagon. 6.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_e=0.12$.
Suspension characteristics $k_x^3$ and $k_y^3$.
+ simulation starting at 160 km/h with decreasing speed.
 o simulation starting at 50 km/h with increasing speed.
a) Lateral amplitude of leading wheelset.
b) Lateral carbody acceleration amplitude above leading wheelset.
c) 99.85-percentile of the lateral track force $Y_{11r}$.
d) Dominating frequency of the lateral carbody acceleration.
The frequency of the lateral motion of the wheelset can be estimated using the Klingel formula for a free or stiffly connected wheelset to a bogie frame

\[ f_{\text{free}} = \frac{v}{2\pi} \sqrt{\frac{\lambda_e}{b_o \cdot r_o}}, \]  
(71)

\[ f_{\text{stiff}} = \frac{v}{2\pi} \sqrt{\frac{\lambda_e}{b_o \cdot r_o} \cdot \frac{1}{1 + \left(\frac{a}{b_o}\right)^2}}, \]  
(72)

where \( a \) is half of the axle semi distance, \( b_o \) half of the semi axlebox distance, \( r_o \) is the wheel radius and \( v \) the speed of the vehicle. Equivalent conicities according to UIC519 are shown in Figure 26. In the example shown in Figure 25 the lateral amplitude of the wheelset is between 3.8 and 6.2 mm. If we insert the corresponding equivalent conicity in Equations (71) and (72) the hunting frequencies according to Klingel can be calculated. In Figure 27 a comparison with simulated values is shown. The simulation results are below the values obtained by Klingels equation for a free wheelset. Reasons for this are that mass forces lead to sliding between wheel and rail. Hence, the wavelength of the sinusodial motion of the wheelset is increased. The motion is not free as the coupling of the wheelset via the primary suspension adds constraints to the motion. We can form a relation \( Q_{\text{Klingel}} \) between the simulated frequencies and estimations with Klingels formulas

\[ Q_{\text{Klingel}} = \frac{f_{\text{sim}} - f_{\text{stiff}}}{f_{\text{free}} - f_{\text{stiff}}}. \]  
(73)

Comparing simulation results with Kilingel frequencies in Figure 27, in average the quotient \( Q_{\text{Klingel}} \) is 61%.

The yaw motion of the carbody is the dominating eigenmode in the frequency range 1.5 - 3 Hz for this particular loadcase. The yaw eigenfrequency of the vehicle can be calculated as

\[ f_{\text{yaw}} = \frac{1}{2\pi} \sqrt{\frac{k_{\text{yaw}}}{J_{\text{yaw}}}}, \]  
(74)

where

\[ k_{\text{yaw}} = 4 \cdot a^2 \cdot k_y, \]  
(75)

and \( k_y \) is the secantial stiffnesses for the non-linear suspension characteristics, cf. \( K_s \) in Figure 5. The secantial stiffness decreases with increasing amplitude. With \( k_y = 0.53 \text{ MN/m} \) and \( J_{\text{yaw}} = 104000 \text{ kgm}^2 \) the yaw eigenfrequency becomes 2.8 Hz.
Figure 26: Equivalent conicity according to UIC519. Ideal S1002 wheel profile. Ideal and worn BV50 rail profiles with 1:30 rail inclination.

Figure 27: Comparison between simulation and Klingels equations.
Now return to the attractor in Figure 25 that arises at 73 km/h. The reason for this attractor is a resonance between the lateral excitation from the wheelset and, for this loadcase, the yaw eigenmode of the carbody. If we follow the attractor with decreasing speed we see that the lateral amplitude of the wheelset increases. Hence, with the increased equivalent conicity that follows with larger amplitude of the motion the excitation frequency is relatively constant for the entire speed range 80 to 160 km/h. From Figure 25 we can calculate the quotient between the frequency at 80 km/h respectively 160 km/h as $f_{80}/f_{160} = 2.28/2.63 = 0.87$. The increased lateral amplitude of the wheelset increases the lateral displacement in the primary suspension, hence, the yaw eigenfrequency of the carbody is reduced. At 86 km/h the maximum lateral displacement of the wheelset is reached. When the speed is further reduced the excitation from the wheelset is not sufficient to fully drive the hunting motion, the amplitude of the carbody acceleration is reduced, and the carbody yaw eigenfrequency constitutently increased. Due to the non-linear suspension characteristics the yaw stiffness is considerably higher for small displacements. When the excitation and the eigenfrequency become to diverse the attractor disappears. At speeds between 73 and 78 km/h the dominating hunting mode is not the carbody yaw but the lower sway, where the wheelsets are displaced laterally and the carbody is displaced laterally and roll in phase with the wheelsets.

5.2 Comparison with previous simulation model

In this section a comparison with a simulation model developed by Lange [13] is made. This model, in the following text called previous model, and the new model is described in Chapter 3.

5.2.1 Suspension characteristics

In Figure 28 the vehicle reaction between the new and previous model is compared. The wheelsets are fixed and a yaw motion with amplitude 1 mrad is applied to the carbody. In a) and b) force-displacement characteristics in the lateral elements are shown. When friction is present in the connection between leafspring and axlebox, $\mu_{lsa} > 0$, the characteristics between inner and outer element differ. However, for the case with $\mu_{lsa} = 0$, not shown in the figure, the new and previous model are identical.

The effect on the yaw moment of the wheelset is considerable. In the previous model only the longitudinal elements contribute to stiffness and damping.
Section 5 - Running behaviour on tangent track

Figure 28: Comparison new and previous model.  
Suspension characteristics $k_x3$ and $k_y3$ according to Table 3:.

- Solid - Previous model.
- Dash dot - new model $\mu_{lsa}=0.1$ (Friction between leafspring and axlebox).
- Dashed - new model $\mu_{lsa}=0.4$.

a) Lateral force-displacement characteristics - inner element.
b) Lateral force-displacement characteristics - outer element.
c) Longitudinal force-displacement characteristics - right element.
d) Yaw moment on wheelset as function of carbody yaw angle.

5.2.2 Comparison on tangent track

In Figure 29 a comparison between simulations with the new respectively previous model on tangent track is shown. We observe a considerable difference in critical speed between the two models. The difference in lateral amplitude for the wheelset is considerable for speeds above 110 km/h. Even though the wheelset lateral amplitude is higher for the new model, i.e. higher equivalent conicity, the frequency of the motion is slightly lower. Calculating $Q_{Klingel}$ according to Equation (73) gives 35% for the previous model and 24% for the new. This is considerably less than what was found for the empty wagon, cf. Section 5.1. Hence, the hunting motion for the loaded wagon produces more sliding motions between wheel and rail than the empty wagon.
5.3 Loading and wheel-rail contact conditions

In this section the dynamic behaviour for different loading and wheel-rail contact conditions is investigated. Equivalent conicity according to UIC519 for the two different wheel-rail combinations is shown in Figure 26.

In Figure 30 a comparison between 6.5 and 22.5 tonnes axleload is shown. Ideal S1002 wheels and BV50 rail profiles with 1:30 inclination are used. The critical speed is shifted from 73 to 100 km/h when the wagon is loaded. The flange-to-flange attractor is shifted even further from 120 km/h to speeds above 300 km/h, i.e. far above present and future operational speeds for freight wagons. The lateral amplitude of the wheelset does not decline as much with speed when the wagon is loaded as for the empty wagon. However, the dominating frequency for the loaded wagon is lower for the loaded wagon even though the equivalent conicity is higher. This is discussed further in Section 5.4.

The behaviour on a track with high equivalent conicity is shown in Figure 31. Here the flange-to-flange attractor is shifted to speeds above 240 km/h for the loaded wagon. The resonance with the carbody yaw eigenfrequency exists for the empty as well as the
loaded wagon. This attractor drops to zero for the loaded wagons at speeds above 124 km/h.

**Figure 30:** Comparison between different loading for low conicity. Suspension characteristics $k_x$ and $k_y$.
+ 22.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_c=0.12$.
- 6.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_c=0.12$.
  a) Lateral amplitude of leading wheelset.
  b) Lateral carbody acceleration amplitude above leading wheelset.
  c) 99.85-percentile of the lateral track force $Y_{11r}$.
  d) Dominating frequency of the lateral carbody acceleration.
5.4 Influence of carbody flexibility

A principal view of the coupling between wheelset and carbody is shown in Figure 32. The equations of motion for the lateral and roll motions are given by

\[
\begin{bmatrix}
    m & 0 \\
    0 & J_{cf}
\end{bmatrix}
   \begin{bmatrix}
         \dot{y} \\
         \dot{\psi}
\end{bmatrix}
+ \begin{bmatrix}
    4k_y & -4k_yh \\
    -4k_yh & 4(k_yh^2 + k_yb^2)
\end{bmatrix}
   \begin{bmatrix}
         \dot{y} \\
         \dot{\psi}
\end{bmatrix}
= \begin{bmatrix}
    0 \\
    0
\end{bmatrix}
\]

(76)

The eigenfrequencies depend on the amplitude of lateral motion in the primary suspension. In Figure 33 a typical lateral force-displacement characteristics is shown. From this figure the secantial stiffness, \( K_s \), can be calculated for different values of the lateral displacement, c.f. Figure 5. The eigenmodes and eigenfrequencies can now be calculated for discrete amplitudes of the lateral displacement. The corresponding eigenmode to the lowest eigenfrequency is a combined lateral and yaw motion of the carbody, from her on denoted the lower sway eigenmode.
In Table 6 carbody yaw and lower sway eigenfrequencies according to Equation (74) and (76) are shown. The frequencies for both eigenmodes are in the range 1.7 - 3.5 Hz depending on the lateral amplitude over the primary suspension. The carbody yaw eigenfrequency is, if we assume a rigid carbody, independent of the axleload. For the lower sway eigenfrequency we see a slight dependency of the axleload. The secantial lateral stiffness $k_y$ increases linearly with increasing carbody mass as shown in Figure 33, if we assume that the carbody is box shaped, $B \times W \times L$. The mass inertia in yaw and roll are calculated according to Equations (69) and (70). Further we consider a case with low height of mass centre, 1.2 meter at 6.5 tonnes axleload and 1.4 meter at 22.5 tonnes axleload, i.e. the mass inertia and lateral stiffness increase approximately linearly with increasing mass.

**Figure 32:** Principle sketch vertical and lateral coupling between carbody and wheelset.

**Table 6:** Carbody yaw and lower sway eigenfrequencies according to Equation (74) and (76). Rigid carbody.
The torsional stiffness of the carbody is an important parameter when designing two-axle freight wagons. As it is decisive when it comes to safety against derailment when the wagon runs over twisted track the limit values are regulated. The torsional stiffness is defined as

\[ c_t^* = \frac{2a \cdot 2b \cdot \Delta F}{\phi} \quad (77) \]

The parameters are shown in Figure 34. The torsional stiffness for a two-axle wagon shall be between \(0.4 \times 10^{10}\) and \(8.4 \times 10^{10}\) kNmm\(^2\)/rad according to UIC 530-2 [24]. For wagons with progressive vertical springs torsional stiffness up to \(10.4 \times 10^{10}\) kNmm\(^2\)/rad is allowed. The torsional stiffness for an open standard two-axle freight wagons is typically in the range \(5-8 \times 10^{10}\) kNmm\(^2\)/rad.

The equivalent lateral stiffness between the centre of mass and the connection point for the lateral primary suspension is shown in Figure 32 and can be calculated as

\[ k_{ycar} = \frac{c_t^*}{2a \cdot h^2} \quad (78) \]
Figure 34: Carbody torsional flexibility. Principle sketch.

The total lateral stiffness in the connection between wheelset and carbody mass centre can now be calculated

\[ k_y = \frac{1}{\frac{1}{k_{ycar}} + \frac{1}{K_s}} \]

where \( K_s \) is the secantial stiffness in the primary suspension according to Figure 33. In Figure 35 the yaw respectively lower sway eigenfrequencies are shown. The influence of the structural stiffness on the eigenfrequencies is considerable and must be included when multibody dynamic simulations are performed. We can see that the eigenfrequency for the lower sway is lower than the carbody yaw eigenfrequency and that torsional stiffness of the carbody has greater influence for the loaded wagon.
Figure 35: Eigenfrequencies for carbody yaw and lower sway according to Equation (74) and (76). Suspension characteristics $k_{x_3}$.

Solid lines - Flexibility for upper respectively lower limit for carbody torsional stiffness included.
Dashed line - Rigid carbody.

a) 6.5 tonnes axleload - 1.2 m height of carbody mass centre.
b) 22.5 tonnes axleload - 1.4 m height of carbody mass centre.
In this section, up to now, the principle influence of torsional carbody flexibility on yaw respectively lower sway eigenmodes has been discussed. In the MBS model however, the first torsional respectively first bending eigenmodes are included.

In Figure 36 a comparison between simulations with flexible respectively rigid carbody and track is shown. For the empty wagon the differences between simulations with flexible respectively rigid carbody are small. Mainly carbody accelerations at frequencies above 5 Hz are influenced.

For the loaded wagon however, the influence of carbody flexibility is considerable. Here mainly the torsional flexibility influences the running behaviour whereas the bending mode affect vertical track forces and ride comfort. The motion patterns for the vehicle with rigid respectively flexible carbody are quite different.

- For the wagon with rigid carbody the wheelsets are displaced laterally almost in phase and the carbody moves laterally and rolls.
- For the wagon with flexible carbody the wheelsets are displaced laterally out of phase and the carbody moves laterally, yaws and rolls.

The driving mechanism for both cases is a resonance between the kinematic motion of the wheelset and the lower sway eigenmode of the carbody. With the torsional flexibility, however, one degree of freedom is added to the system and the lower sway motion can occur independently over leading respectively trailing wheelsets and interact with the yaw motion of the carbody. Hence, the lateral acceleration and track forces increase.

In Figure 37 lateral carbody accelerations from on-track tests with a two-axle freight wagon are shown. The wagon is on test section 4 mainly hunting over the trailing wheelset, hence, the motion is driven by a resonance with the lower sway eigenmode. On section 3 the lateral accelerations above leading respectively trailing wheelset are almost 180 degrees out of phase. The amplitude of the acceleration is larger. This is a typical difference between the hunting modes, c.f. Figure 36. The main difference between the two sections is the track gauge, that is in average slightly tighter on section 4, 1434 mm compared to 1436 mm on section 3.
**Figure 36:** Comparison between flexible and rigid carbody and track. Suspension characteristics $k_{x3}$ and $k_{y3}$. S1002 BV50i30, 1435 mm gauge, $\lambda_e=0.12$.

- + Flexible carbody.
- o Rigid carbody.
- a) Lateral amplitude of leading wheelset.
- b) Lateral carbody acceleration amplitude above leading wheelset.
- c) 99.85-percentile of the lateral track force $Y_{11r}$.
- d) Dominating frequency of the lateral carbody acceleration.
Figure 37: Test result. Lateral carbody acceleration lowpass filtered at 10 Hz. 18 tones axleload. Dashed line - above leading wheelset. Solid line - above trailing wheelset.

a) Tangent test section 3. Mean gauge 1436 mm.
b) Tangent test section 4. Mean gauge 1434 mm.

5.5 Variation in suspension characteristics

As discussed in Chapter 2 the variation in suspension characteristics between different wagons is considerable. Three typical suspension characteristics for various status of the suspension components are:

- Case 1: $k_x^4$ and $k_y^1$ - New components.
- Case 2: $k_x^3$ and $k_y^3$ - Intermediate worn components.
- Case 3: $k_x^1$ and $k_y^5$ - Worn components.

A comparison is made for two different wheel-rail contact conditions at 6.5 respectively 22.5 tonnes axleload. The resulting diagrams at 22 tonnes and 6.5 tonnes axleload are shown in Figure 38 respectively Figure 39. In Table 7 the speeds at the bifurcation points for the carbody respectively wheelset hunting attractors are shown. The variation between the different cases is considerable. For many combinations non-zero attractors are present at typical operational speeds, i.e. 90-100 km/h. For the empty wagon the wheelset hunting, or flange-to-flange attractor is present for speeds above 100-120 km/h. Hence further increasing the speed for empty wagons is not advisable as this hunting mode is safety critical.

In Appendix A and B results from several other parameter variations with the new respectively the previous model are shown. The variation in the longitudinal suspension characteristics have most influence on the running behaviour.

Today wheelset hunting limits the possibility to increase the speed of empty wagons. If new materials are introduced in the suspension components that are resistant to wear and give more precise friction behaviour the suspension characteristics can be designed more
precisely. Then the variation of suspension characteristics would be less over the life span of the suspension components. With increased initial stiffness and damping in the links the critical speed is increased. However, as long as the longitudinal pendulum stiffness is kept at today's levels the influence on the curving performance would be moderate.

**Table 7:** Variation in suspension characteristics.
*Speeds at bifurcation points in Figure 38 and Figure 39.*

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbody hunting</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>λ = 0.12</td>
<td>22.5</td>
<td>109</td>
<td>100</td>
<td>68</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>6.5</td>
<td>71</td>
<td>73</td>
<td>50*</td>
<td>122</td>
<td>119</td>
<td>106</td>
</tr>
<tr>
<td>Wheelset Hunting</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>λ = 0.42</td>
<td>22.5</td>
<td>-</td>
<td>77</td>
<td>-</td>
<td>159</td>
<td>-</td>
<td>149</td>
</tr>
<tr>
<td></td>
<td>6.5</td>
<td>62</td>
<td>56</td>
<td>50*</td>
<td>105</td>
<td>103</td>
<td>93</td>
</tr>
</tbody>
</table>

Note:  
- No attractor present in the speed range 50 to 160 km/h.  
* The speed for the bifurcation is below 50 km/h.
Figure 38: Variation of suspension characteristics. Lateral amplitude of wheelset, lateral carbody acceleration amplitude, lateral track force and dominating frequency.
+ Suspension characteristics $k_{x4}$ and $k_{y1}$ (“New”).
o Suspension characteristics $k_{x3}$ and $k_{y3}$ (“Intermediate”).
* Suspension characteristics $k_{x1}$ and $k_{y5}$ (“Worn”).
a) 22.5 tonnes axle load, S1002 BV50i30, 1435 mm gauge, $\lambda_e=0.12$.
b) 22.5 tonnes axle load, S1002 BV50i30w, 1431 mm gauge, $\lambda_e=0.42$. 
Figure 39: Variation of suspension characteristics. Lateral amplitude of wheelset, lateral carbody acceleration amplitude, lateral track force and dominating frequency:
+ Suspension characteristics $k_{x4}$ and $k_y1$ (“New”).
 o Suspension characteristics $k_{x3}$ and $k_y3$ (“Intermediate”).
* Suspension characteristics $k_{x1}$ and $k_y5$ (“Worn”).
a) 6.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_e=0.12$.
b) 6.5 tonnes axle load. S1002 BV50i30w, 1431 mm gauge, $\lambda_e=0.42$. 

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5.6 Stiffness and damping in the leaf springs

The energy dissipation in the leaf springs varies with maintenance status of the springs, cf. Section 3.1.2. The frictional break-out force is,

- ±4-8% of the vertical axlebox load for new springs,
- ±13-15% of the vertical axlebox load for reconditioned ungreased springs.

As a typical value generally in this report 14% is used. In Figure 40 a comparison between 6% and 14% vertical damping is made. The critical speed is shifted from 100 to 114 km/h, however, also the hunting mode is changed. With 14% vertical damping the dominating hunting mode is the lower sway. The vertical break-out force in the leaf spring is high and the lower sway hunting motion is relatively undamped. For the case with 4% damping the break-out force is reached and the lower sway motion is damped out. However, now the carbody yaw motion is dominating instead.

Figure 40: Comparison between high and low vertical damping. Suspension characteristics $k_{x3}$ and $k_{y3}$. 22.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_e=0.12$. + Reference case with high amount of vertical damping (14%). o Low amount of vertical damping (6%). a) Lateral amplitude of leading wheelset. b) Lateral carbody acceleration amplitude above leading wheelset. c) 99.85-percentile of the lateral track force Y11r. d) Dominating frequency of the lateral carbody acceleration.

The properties of the leafspring influence to great extent the vertical track forces and ride comfort. To investigate this medium sized track irregularities are introduced. The irregularities are representative for a continuous welded track intended for 80-120 km/h,
Multibody simulation model for freight wagons with UIC link suspension

maintenance level QN1 according to prEN14636 [20]. A 500 m long track section is used for the simulations. Initially the vehicle runs over a section including large lateral track irregularities in order to excite possible limit cycles in the system. We compare four different leafspring configurations.

- Case 1: 14% damping, initial stiffness $k_f = 5 \, [\text{MN/m}]$.
- Case 2: 14% damping, initial stiffness $k_f = 15 \, [\text{MN/m}]$.
- Case 3: 4% damping, initial stiffness $k_f = 5 \, [\text{MN/m}]$.
- Case 4: 4% damping, initial stiffness $k_f = 15 \, [\text{MN/m}]$.

In Figure 41 the 99.85-percentiles of the vertical track forces and RMS-values of the vertical carbody acceleration are shown. We observe a considerable influence of the suspension parameters on the track forces as well as ride comfort.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure41.png}
\caption{Influence of leafspring properties. 22.5 tonnes axleload.
\begin{itemize}
  \item [a)] 99.85-percentiles - Vertical track forces.
  \item [b)] RMS-value. Vertical carbody acceleration.
\end{itemize}}
\end{figure}

5.7 Height of carbody mass centre

Up to now we have considered loadcases with low centre of gravity, i.e. the wagon is loaded with high density goods. In this section the influence of different loading conditions is discussed. Simulations with 1.4 m, 1.8 m and 2.2 m height of carbody centre are compared. The moment of inertia around the x-axis is changed from 3400 kgm$^2$ till 40500 kgm$^2$ respectively 59000 kgm$^2$. In Figure 42 simulation results from the three different loadcases are shown. The carbody yaw and lower sway eigenfrequencies are reduced as the height of mass centre is increased. The lower sway is affected more as also the roll mass inertia influences this eigenfrequency, cf. Equation (76). We have two dominating carbody eigenmodes, carbody yaw and lower sway.

However, they can result in three principally different hunting modes:

- Carbody yaw - The wheelsets are displayed laterally out of phase and the carbody is yawing.
Section 5 - Running behaviour on tangent track

- Symmetric lower sway - The wheelsets are displayed laterally in phase and the carbody is displayed laterally and is rolling.
- Antisymmetric lower sway - The wheelsets are displayed laterally out of phase and the carbody is yawing, displayed laterally and is rolling. The lower sway motion occurs antisymmetric over leading respectively trailing wheelset as the roll motion of the carbody is decoupled via the torsional stiffness. When the carbody yaw and the lower sway eigenfrequencies are in the vicinity of each other the antisymmetric lower sway motion is excited.

For the cases with 1.4 m respectively 1.8 m height of carbody mass centre the results are relatively similar. The dominating motion is the antisymmetric lower sway. When the height is increased to 2.2 meter the yaw respectively lower sway eigenfrequencies become to diverse and the motion is dominated by the symmetric lower sway.

Figure 42: Comparison between different height of carbody mass centre
Suspension characteristics $k_x$ and $k_y$.
22.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_c=0.12$.
+ 1.4 m height of carbody mass centre.
o 1.8 m height of carbody mass centre.
* 2.2 m height of carbody mass centre.
a) Lateral amplitude of leading wheelset.
b) Lateral carbody acceleration amplitude above leading wheelset.
c) 99.85-percentile of the lateral track force $Y_{11r}$.
d) Dominating frequency of the lateral carbody acceleration.
5.8 Influence of continuously variable suspension characteristics

So far we have assumed the lateral and longitudinal suspension characteristics to linearly depend on the static vertical load on the axlebox. However, during operations large variations in the vertical axlebox load can occur. For instance when the vehicle is running through a curve with cant deficiency or excess or in situations leading to derailment caused by wheel unloading.

In this section the influence of continuously variable suspension characteristics, i.e. the suspension parameters are at every timestep updated due to changes in the vertical load on the axlebox, is investigated. The influence on the running behaviour on tangent track is limited as shown in Figure 43.

![Comparison between constant respectively variable horizontal suspension characteristics. Suspension characteristics kx and ky.](image)

**Figure 43:** Comparison between constant respectively variable horizontal suspension characteristics. Suspension characteristics $k_x$ and $k_y$.
- 1.4 m height of carbody mass centre.
- 22.5 tonnes axle load. S1002 BV50i30,1435 mm gauge, $\lambda_e=0.12$.
- a) Lateral amplitude of leading wheelset.
- b) Lateral carbody acceleration amplitude above leading wheelset.
- c) 99.85-percentile of the vertical track force $Y_{11r}$.
- d) Dominating frequency of the lateral carbody acceleration.

5.9 Axlebox play

In order to accommodate thermal expansion due to variation in temperature lateral play is needed in the bearings. The influence of the bearing play on the running behaviour is discussed in this section. The suspension model is modified to accommodate 0.4 mm lateral play in the coupling between leafspring and wheelset. A comparison at 22.5 tonnes axleload is shown in Figure 44 and for an empty wagon in Figure 45. The overall
influence on the running behaviour is limited. The reason might be that the link suspension is soft. The difference is probably more significant for stiffer wheelset guidance.

Figure 44: Influence of lateral play in the bearings. Suspension characteristics $k_{x3}$ and $k_{y3}$. 1.4 m height of carbody mass centre. 22.5 tonnes axle load. S1002 BV50i30, 1435 mm gauge, $\lambda_e = 0.12$.

+ Rigid connection. Starting at 160 km/h with decreasing speed.
0 0.4 mm lateral play. Starting at 160 km/h with decreasing speed.
* Rigid connection. Starting at 50 km/h with increasing speed.
+ 0.4 mm lateral play. Starting at 50 km/h with increasing speed.

a) Lateral amplitude of leading wheelset.
b) Lateral carbody acceleration amplitude above leading wheelset.
c) 99.85-percentile of the vertical track force $Y_{11r}$.
d) Dominating frequency of the lateral carbody acceleration.
Figure 45: Influence of lateral play in the bearings.
Suspension characteristics $k_{x3}$ and $k_{y3}$.
1.2 m height of carbody mass centre.
6.5 tonnes axle load. S1002 BV50i30,1435 mm gauge, $\lambda_e=0.12$.
+ Rigid connection.
0 0.4 mm lateral play.
a) Lateral amplitude of leading wheelset.
b) Lateral carbody acceleration amplitude above leading wheelset.
c) 99.85-percentile of the vertical track force $Y_{11r}$.
d) Dominating frequency of the lateral carbody acceleration.
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6 QUASISTATIC CURVING

6.1 Quasistatic curving behaviour

In this section the quasistatic behaviour of a vehicle in curves is investigated. Indicators of the steering capability of a vehicle are the yaw angle of the wheelset, quasistatic lateral track force, \( Y_{\text{qst}} \) and the energy dissipation in the contact patch. In this report the energy dissipation is used. It is calculated as the sum of products between the creep forces and the corresponding creepage,

\[
E = F_\zeta \cdot v_\zeta + F_\eta \cdot v_\eta + M_\zeta \cdot \phi. \tag{80}
\]

In case of ideal steering the wheelsets have a radial position, i.e. the axles points to the centre of the curve radius and the yaw angle is equal to zero.

Decisive parameters for the steering capability are:

- The wheel-rail profile combination that must admit sufficient difference in rolling radius between outer and inner wheel.
- Properties of the primary suspension.
- Friction between wheel and rail.

The needed difference in rolling radius between outer and inner wheel can be calculated according to Andersson et.al. [1] as

\[
r_{\text{out}} - r_{\text{in}} = 2r_0 \frac{b_0}{R}, \tag{81}
\]

where \( r_0 \) is the nominal wheel radius, \( b_0 \) half the lateral distance between the contact points and \( R \) the curve radius. Typical values for a wheel with 920 mm diameter are given in Table 8.

<table>
<thead>
<tr>
<th>( R )</th>
<th>250</th>
<th>300</th>
<th>400</th>
<th>600</th>
<th>[m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_{\text{out}} - r_{\text{in}} )</td>
<td>2.8</td>
<td>2.3</td>
<td>1.8</td>
<td>1.3</td>
<td>[mm]</td>
</tr>
<tr>
<td>( x ) (2a=9 m)</td>
<td>18.0</td>
<td>15.0</td>
<td>11.3</td>
<td>7.5</td>
<td>[mm]</td>
</tr>
<tr>
<td>( x ) (2a=1.8 m)</td>
<td>3.6</td>
<td>3.0</td>
<td>2.3</td>
<td>1.5</td>
<td>[mm]</td>
</tr>
</tbody>
</table>

However, the wheelset is connected to the carbody or a bogie frame and the primary suspension must allow the axlebox to move longitudinally. In Figure 46a) the geometrical relations are shown. The longitudinal displacement in the primary suspension for radial alignment of the wheelset in a curve can be calculated to
Section 6 - Quasistatic Curving

\[ x = \frac{a \cdot b_1}{R}. \]  

(82)

Longitudinal displacements for a two-axle freight wagon with 9 meter axledistance and for a bogie with 1.8 m semi axle distance are shown in Table 8. In order to achieve radial alignment of the wheelset the restrictive yaw moment from the primary suspension has to be balanced by longitudinal creep forces

\[ 2 \cdot b_0 \cdot F_\xi = 2 \cdot b_1 \cdot F_x + 2 M_{lsa}, \]

(83)

where \( F_x \) is the longitudinal force in the primary suspension and \( M_{lsa} \) is the friction moment in the yaw coupling between leafspring and axlebox.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure46.png}
\caption{Quasistatic curving.}
\begin{itemize}
  \item[a)] Geometric data.
  \item[b)] Origin of lateral creepage.
\end{itemize}
\end{figure}

Consider a two-axle wagon at 22.5 tonnes axleload running through curve a with 400 meter radius. The longitudinal forces in the primary suspension with three different force-displacement characteristics are given in Figure 47. The longitudinal wheel-rail contact forces, \( F_\xi \), are calculated according to Equation (83) assuming that \( M_{lsa} \) is zero. In Table 9 the results for three different suspension characteristics are shown.

In Figure 48 the energy dissipation on the leading outer wheel on a two-axle wagon running in an ideal circular curve with 400 metre radius under quasistationary conditions is given. Three typical suspension characteristics are compared. The curving
performance with suspension characteristics $kx_1$ and $kx_3$ with $\mu=0.4$ is nearly ideal. When the friction is reduced it is not possible to build up sufficient amount of longitudinal forces and the radial alignment of the wheelset is lost. A deviation from a radial alignment, i.e. increased yaw angle of the wheelset, leads to lateral creepage because the direction of the motion of the wheelset is not the same as the rolling direction, c.f. Figure 46b). Hence, the energy dissipation increases as the lateral creepage increases. However, when the friction is further reduced the creep forces and with them also the energy dissipation become lower.

In Figure 48 it is shown that the radial alignment of the wheelset for $kx_3$ with $\mu=0.4$ is good. However, this is not the case for $\mu=0.3$. Comparing these simulation results with the estimations in Table 9 we can as a rule of thumb say that for small wheelset yaw angles not more than half, in this case approximately 40%, of the available friction can be used to build up longitudinal forces. However, for non-favourable wheel-rail contact geometry considerable less amount of the friction in the contact between wheel and rail can be utilised in the longitudinal direction.

![Figure 47: Longitudinal force $F_x$ versus displacement. Suspension characteristics $kx_1$, $kx_3$ and $kx_5$. 22.5 tonnes axleload.](image-url)
Figure 48: Energy dissipation. Leading outer wheel. Ideal track. 22.5 tonnes axleload. Curve R400 meter.

Table 9: Utilization of static wheel load $Q_0$. Curve R400 m. 22.5 tonnes axleload. $2a=9$ meter.

<table>
<thead>
<tr>
<th>Longitudinal characteristics</th>
<th>$kx_1$</th>
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<tr>
<td>$F_x$</td>
<td>7.39</td>
<td>11.12</td>
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<td>$F_\xi$ ($M_{lsa}=0)$</td>
<td>9.85</td>
<td>14.83</td>
<td>20.73</td>
</tr>
<tr>
<td>$F_\xi / Q_0$</td>
<td>0.09</td>
<td>0.13</td>
<td>0.19</td>
</tr>
<tr>
<td>$F_\xi / (Q_0^*\mu), \mu=0.3$</td>
<td>0.30</td>
<td>0.43</td>
<td>0.63</td>
</tr>
<tr>
<td>$F_\xi / (Q_0^*\mu), \mu=0.4$</td>
<td>0.23</td>
<td>0.33</td>
<td>0.48</td>
</tr>
</tbody>
</table>

Figure 49 shows the vehicle entering a curve with 400 m radius. The simulations start on tangent track going through a 120 meter long transition curve and ends in a circular curve with 150 mm cant. The axleload is 22.5 tonnes and speed 92 km/h, i.e. the vehicle is running with 100 mm cant deficiency. The longitudinal creep forces are for the case with $\mu=0.4$ slightly higher than what is given in Table 9, 19.53 kN compared to 14.83 kN. The reason for this deviation is that the friction moment, $M_{lsa}$, contributes to the quasistatic restrictive yaw moment in the primary suspension. The height of carbody mass centre is assumed to be low, 1.4 meter. However, still the quasistatic contribution to the vertical force on the outer wheel is $128.7/110.4=1.17$. For the case with lower friction, $\mu=0.3$, it is not possible to build up sufficient amount of longitudinal creep forces. Hence, the yaw angle and the lateral force increase.
6.2 Influence of track irregularities

The simulations in Section 6.1 are performed in time domain. The wagon is running on ideal track geometry starting on a tangent section continuing through the transition curve and finally a circular curve. For simulations on ideal track hydraulic dampers are used between wheelset and carbody to damp out the dynamic contribution that arises when the wagon is entering or leaving the transition curve.

In Figure 50 a comparison between simulations with respectively without track irregularities is shown. No hydraulic dampers are used for the simulations with track irregularities. For the simulations without track irregularities in the 400 meter curve it is not possible to build up sufficient amount of longitudinal forces. The longitudinal displacement in the primary suspension is approximately 6.5 mm and the wheelset takes an under radial position increasing the lateral creepage. The energy dissipation for this
case is 467 Nm/m as shown in Figure 51. When track irregularities are included we see two interesting phenomena. The additional oscillations introduced by the track irregularities give the wheelset a nearly radial position. This effect is known as friction climbing and can be found in various oscillating friction damped systems. The pendulum stiffness, $k_2$ in Table 3, is indicated by the dash-dotted line in the upper graph. It is clearly shown that the longitudinal force, in average, is given by the displacement and the pendulum stiffness. Hence, the force is considerably lower than shown in Figure 47. The energy dissipation is reduced to 53 Nm/m calculated as an average value for the circular curve. In the following comparison track irregularities are included in the simulations.

**Figure 50:** Comparison simulations with and without track irregularities. Curve R400 m, 22.5 tonnes axleload, 2a=9 meter. Suspension characteristics $k_x$, $k_y$. Longitudinal displacement between carbody and left respectively right axlebox. Upper plot - Longitudinal force - displacement characteristics. Lower plot - Longitudinal displacement versus time.
6.3 Comparison with previous simulation model.

In Figure 52 results with the new and the previous simulation model are compared. The yaw resistance for the new model is higher, as discussed in Section 5.2, resulting in slightly higher energy dissipation.
Section 6 - Quasistatic Curving

Figure 52: Energy dissipation. Comparison between simulations with new and previous model. 22.5 tonnes axleload. Suspension characteristics $k_x^3$ $k_y^3$. \( \mu = 0.3 \).

6.4 Variation in suspension characteristics

In Figure 53 a comparison between three typical suspension characteristics is shown:

- $k_x^5$ and $k_y^1$ - New components.
- $k_x^3$ and $k_y^3$ - Intermediate worn components.
- $k_x^1$ and $k_y^5$ - Worn components.

For $k_x^1$ and $k_x^2$ the energy dissipation is below 200 Nm/m for curves with radius above 300 m and \( \mu \) above 0.3. Hence, the quasistatic curving performance for wagons with link-suspension is very good. For $k_x^5$ the values are higher. However, with slightly increased friction, \( \mu = 0.4 \), the energy dissipation is considerably reduced.
6.5 Influence of continuously variable suspension characteristics

Due to the quasistatic wheel load redistribution, shown in Figure 49, the characteristics of the horizontal suspension change. Hence, the stiffness and hysteresis increase in the outer suspensions and decrease in the inner suspension as the vehicle, in this case, is running with 100 mm cant deficiency. However, computing the parameters for the suspension characteristics at every timestep increase the simulation time with a factor of 2.5. Hence, generally in this report it is assumed that the suspension characteristics linearly depend on the static vertical load on the axlebox.

In Figure 54 simulation results with constant respectively continuously variable suspension characteristics are compared. The curving performance is slightly improved due to the variable characteristics. It shall be pointed out that here 1.4 m height of carbody mass centre is considered. The vertical wheel load redistribution would be larger with higher carbody mass centre.
Figure 54: Influence of continuously variable suspension characteristics. 22.5 tonnes axleload. Curve R300 meter.
7 CONCLUSIONS AND SUGGESTIONS FOR IMPROVEMENTS

7.1 Conclusions

Simulation results with the presented new model for two-axle freight wagons with UIC link suspension agree well with on-track test results. Hence, the model represents reality quite well, at least in the frequency range up till 15-20 Hz. The new model approach has considerable influence on the running behaviour. For the loaded wagon the critical speed is increased by up to 50% compared to the old model.

The most important parameters for the running behaviour are the suspension characteristics. At the same time the variation in the lateral and longitudinal suspension characteristics is considerable. The reasons for the variation are differences in contact conditions between link and bearing caused by geometrical tolerances of the components, wear, corrosion, moisture or other lubrication. The influence of the variations in suspension characteristics on the dynamic performance of two-axle freight wagons is considerable. The variation in critical speed can be as much as ± 20 km/h depending on the suspension characteristics.

There are two principle different hunting modes for these types of wagons,

- wheelset hunting (usually in the frequency range 4-8 Hz),
- carbody hunting (usually in the frequency range 1.5-3 Hz).

However, carbody hunting can be caused by a resonance with several carbody eigenmodes,

- carbody yaw,
- symmetric lower sway,
- anti-symmetric lower sway.

The flexible carbody has a strong influence on the running behaviour of loaded wagons. The frequency of the carbody yaw eigenmode is decreased when flexible properties of the carbody is considered. Further more the torsional flexibility allows a hunting motion pattern where the trailing and leading wheelsets move independently, the anti-symmetric lower sway.

For the loaded wagon it is generally carbody hunting that causes unfavourable ride conditions, at least at normal operational speeds. Carbody hunting causes increased track forces and ride discomfort. At high conicity wheelset hunting can occur for speeds above 140-150 km/h.

For empty wagons carbody hunting occurs at lower speeds than for the loaded wagon. Since the track forces are low and it is generally not safety critical, other than under special conditions, it may be acceptable. However, for the empty wagons wheelset hunting can occur at speeds around current operational speeds on track with low as well as high conicity. To increase speed is not advisable without improving the suspension.

The quasistatic curving performance of two-axle freight wagons with link suspension is good due to the relatively low pendulum stiffness. Radial steering down to curve radii of 300 metre is possible under favourable conditions.
70

Section 7 - Conclusions and Suggestions for Improvements

The vertical damping force in the trapezoidal leaf spring is high. Hence, the vertical stiffness is high for small amplitudes which leads to high vertical track forces and bad ride behaviour.

7.2 Suggestions for improvements

The wheel-rail interface can be improved from a national Swedish perspective. The S1002 wheel profile is optimised for 1:40 rail inclination and the P8 profile for a rail inclination of 1:20. In Sweden the track is laid with a rail inclination of 1:30. It should be possible to develop a wheel profile that is better adapted to this condition. The wheel-rail interface is also an important issue from a European perspective, e.g. for a freight wagon going from Germany to France, cf. Figure 2.

With enhanced wheel-rail contact geometry, i.e. initially a slightly higher conicity then today, and sharpened requirements for maintenance of the suspension components the ride comfort for loaded wagons can probably be improved.

Today wheelset hunting limits the possibility to increase speed of empty wagons. If new materials are introduced in the suspension components that are resistant to wear and give more precise friction behaviour the suspension characteristics can be designed more precisely. Then the variation of suspension characteristics would be less over the life span of the suspension components. With increased initial stiffness and damping in the links the critical speed is increased. As long as the longitudinal pendulum stiffness is kept at today’s level the influence on the curving performance would be moderate, the good curving performance would remain.
REFERENCES


Reference


## APPENDIX A  STABILITY MAP - NEW MODEL

<table>
<thead>
<tr>
<th>Lateral amplitude</th>
<th>Lateral amplitude</th>
<th>Amplitude [mm]</th>
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</thead>
<tbody>
<tr>
<td>Leading wheelset</td>
<td>Trailing wheelset</td>
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</tr>
<tr>
<td>Yaw angle</td>
<td>Yaw angle</td>
<td>Amplitude [mrad]</td>
</tr>
<tr>
<td>Leading wheelset</td>
<td>Trailing wheelset</td>
<td></td>
</tr>
<tr>
<td>Lateral acceleration</td>
<td>Lateral acceleration</td>
<td>Amplitude [m/s²]</td>
</tr>
<tr>
<td>Carbody above</td>
<td>Carbody above</td>
<td></td>
</tr>
<tr>
<td>Leading wheelset</td>
<td>Trailing wheelset</td>
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<td>Dominating frequency</td>
<td>Dominating frequency</td>
<td>Frequency [Hz]</td>
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<td>Lateral acceleration above</td>
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<tr>
<td>Leading wheelset</td>
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<td>Vertical track force</td>
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<td>99.85-percentile [kN]</td>
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<td>Leading right wheel</td>
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<tr>
<td>Lateral track force</td>
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<td>99.85-percentile [kN]</td>
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<tr>
<td>Leading right wheel</td>
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### Table A1: Speeds at bifurcation points.

<table>
<thead>
<tr>
<th>Axleload</th>
<th>Carbody hunting</th>
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<tr>
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<td>kx₁</td>
<td>kx₃</td>
</tr>
<tr>
<td>λ=0.12</td>
<td>22.5</td>
<td>ky₁</td>
</tr>
<tr>
<td></td>
<td>22.5</td>
<td>ky₃</td>
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<tr>
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<td>ky₅</td>
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<tr>
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<tr>
<td></td>
<td>22.5</td>
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<td>ky₅</td>
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<td></td>
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<td>ky₃</td>
</tr>
<tr>
<td></td>
<td>6.5</td>
<td>ky₅</td>
</tr>
</tbody>
</table>

Note: - No attractor present in the speed range 50 to 160 km/h.
* The speed for the bifurcation is below 50 km/h.
Appendix A - Stability map - New model

New model

22.5 tonnes axelload
S1002 - Bv50i30
Gauge - 1435 mm
λ_e=0.12

ky1  +  - kx1
  o  - kx3
  *  - kx4

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
New model

22.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
\( \lambda_c = 0.12 \)

\[ ky_3 \quad + \quad kx_1 \]

\[ o - kx_3 \]

\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Appendix A - Stability map - New model

**New model**

- 22.5 tonnes axleload
- S1002 - Bv50i30
- Gauge - 1435 mm
- $\lambda_c = 0.12$

- $k_y5 + k_{x1}$
- $k_{x3}$
- $k_{x4}$

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Multibody simulation model for freight wagons with UIC link suspension

New model
22.5 tonnes axleload  
S1002 - Bv50i30-worn  
Gauge - 1431 mm  
$\lambda_e = 0.42$

ky1  +  - kx1  
0  - kx3  
*  - kx4

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Appendix A - Stability map - New model

New model
22.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
$\lambda_e=0.42$

$ky_3 + - kx_1$
$O - kx_3$
$* - kx_4$

Lateral displacement
and yaw angle of
wheelset.

Lateral carbody acceleration
and dominating
frequency.

Vertical and lateral track
force on leading wheelset.
**New model**

22.5 tonnes axleload  
S1002 - Bv50i30-worn  
Gauge - 1431 mm  
$\lambda_e=0.42$

$k_y5 + k_x1$  
$- k_x3$  
$* - k_x4$

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Appendix A - Stability map - New model

New model

6.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
\( \lambda_c = 0.12 \)

\[ ky_1 + - kx_1 \]
\[ o - kx_3 \]
\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Multibody simulation model for freight wagons with UIC link suspension

Appendix A - 9

New model

6.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
$\lambda_e=0.12$

$ky_3 + - kx_1$
$0 - kx_3$
$* - kx_4$

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.

Appendix A - 9
Appendix A - Stability map - New model

**New model**

6.5 tonnes axleload  
S1002 - Bv50i30  
Gauge - 1435 mm  
\( \lambda_c = 0.12 \)

\[ ky_5 + ky_1 \]
\[ - ky_3 \]
\[ - ky_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
New model

6.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
\( \lambda_e = 0.42 \)

\[ ky_1 + - kx_1 \]
\[ o - kx_3 \]
\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Appendix A - Stability map - New model

**New model**

6.5 tonnes axleload  
S1002 - Bv50i30-worn  
Gauge - 1431 mm  
$\lambda_e=0.42$

ky₃  +  kx₁  
₀ - kx₃  
* - kx₄

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Multibody simulation model for freight wagons with UIC link suspension

New model

6.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
\( \lambda_e = 0.42 \)

\[ ky_5 + kx_1 \]
\[ o - kx_3 \]
\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
## APPENDIX B  STABILITY MAP - OLD MODEL

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<td>Frequency [Hz]</td>
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<td>Leading right wheel</td>
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<tr>
<td>Lateral track force</td>
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<td>99.85-percentile [kN]</td>
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<td>Leading right wheel</td>
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<td>99.85-percentile [kN]</td>
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### Table B1: Speeds at bifurcation points.

<table>
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<th>Wheelset Hunting</th>
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Note: - No attractor present in the speed range 50 to 160 km/h.
* The speed for the bifurcation is below 50 km/h.
Appendix B - Stability map - Old model

**Old model**

22.5 tonnes axleload  
S1002 - Bv50i30  
Gauge - 1435 mm  
$\lambda_c = 0.12$

ky1 $+$ - kx1  
$\theta$ - kx3  
$*$ - kx4

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Old model

22.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
\( \lambda_e = 0.12 \)

\[ ky_3 + kx_1 \]
\[ o - kx_3 \]
\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Old model

22.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
$\lambda_c = 0.12$

ky5  + - kx1
O - kx3
* - kx4

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Old model

22.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
\( \lambda_c = 0.42 \)

\( k_{y1} + - k_{x1} \)
\( o - k_{x3} \)
\( * - k_{x4} \)

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Appendix B - Stability map - Old model

Old model

22.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
$\lambda_c=0.42$

$ky_3 + - kx_1$
$- - kx_3$
$* - kx_4$

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Multibody simulation model for freight wagons with UIC link suspension

Old model
22.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
\( \lambda_e = 0.42 \)

\( ky_5 \) + - \( kx_1 \)
\( o - kx_3 \)
\( * - kx_4 \)

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Old model

6.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
\( \lambda_c = 0.12 \)

ky1 \( + \) kx3
kx1 \( - \) kx4

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Multibody simulation model for freight wagons with UIC link suspension

Appendix B

Old model

6.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
\( \lambda_c = 0.12 \)

\[ ky_3 + kx_1 \]
\[ o - kx_3 \]
\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
**Old model**

6.5 tonnes axleload
S1002 - Bv50i30
Gauge - 1435 mm
\( \lambda_e = 0.12 \)

\( ky_5 \) + - \( kx_1 \)

\( o \) - \( kx_3 \)

\( * \) - \( kx_4 \)

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Old model

6.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
$\lambda_e=0.42$

$ky_1 \; + \; kx_1$

$0 \; - \; kx_3$

$* \; - \; kx_4$

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
**Old model**

6.5 tonnes axleload  
S1002 - Bv50i30-worn  
Gauge - 1431 mm  
$\lambda_c=0.42$

$ky_3 + - kx_1$  
$0 - kx_3$  
$* - kx_4$

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.
Old model

6.5 tonnes axleload
S1002 - Bv50i30-worn
Gauge - 1431 mm
\( \lambda_e = 0.42 \)

\[ ky_5 \quad + - kx_1 \]
\[ \circ - kx_3 \]
\[ * - kx_4 \]

Lateral displacement and yaw angle of wheelset.

Lateral carbody acceleration and dominating frequency.

Vertical and lateral track force on leading wheelset.