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Development of Track-Friendly Bogies for High Speed

A Simulation Study

by

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Preface

This simulation study is part of the research and development program "GrönaTåget" (GreenTrain), financed by the Swedish National Rail Administration (Banverket). It has been carried out at the Royal Institute of Technology (KTH) in Stockholm, Sweden, in close co-operation with Bombardier Transportation.

The simulation model had been built by Björn Roos and Olaf Kämmerling at Bombardier Transportation and their support and advice are gratefully acknowledged.

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Abstract

In 2004 the reasearch and development program “GrönaTåget” (GreenTrain) was initiated in order to develop the next generation of high-speed trains for Nordic conditions, aiming at speeds up to 250 à 300 km/h, still being “friendly” to the track.

The purpose with this simulation study is to investigate what measures to be taken in order to modify a bogie that guarantees a stable running behaviour at higher speeds, but still generates acceptably low track forces and amount of wear in curves with smaller curve radii. Simulations are performed with a one-car Regina train modelled in the simulation tool SIMPACK.

The simulations have been performed with track geometry cases with varying curve radius, cant and speed (i.e. varying cant deficiency and hence track plane acceleration). The assessment of track forces and ride comfort has been performed on straight track and in curves with radii between 300 and 4900 metres.

In order to maintain a stable running behaviour at higher speed the wheelset guidance stiffness of the bogie, as well as the yaw damping, had to be increased, compared with the original design intended for 200 km/h. The purpose of the present study has been to investigate the difference between two different types of bogie configurations with soft and medium wheelset guidance. For comparison, original soft (original Regina for $v \leq$ 200 km/h) and stiff wheelset guidances have been included in the evaluation. The results indicate that the difference between the two bogie types (“soft” and “medium”) is not that evident on straight track and in large-radius curves. However, in small-radius curves the soft bogie configuration generates lower lateral track forces and hence lower energy dissipation (related to wheel and rail wear) than the medium bogie configuration.

In the present simulation study the influence of equivalent conicity on the vehicle’s running behaviour has been examined. Ten different contact conditions have been defined, with varying wheel and rail profiles, track gauge and rail inclination. Furthermore, two different types of conicity have been defined, simply called Type 1 and Type 2, where the latter has a high conicity at small amplitudes. Generally, higher values of equivalent conicity generate higher track forces and accelerations. Additionally, simulations performed with conicity Type 2 generally cause higher track forces and accelerations than those with Type 1.

During the summer 2006 high-speed tests have been performed on Swedish tracks. A small amount of track force results from these tests have been used in order to validate the simulation model. Test and simulation results show good agreement when studying the track forces in small-radius curves. Also the principal behaviour on straight track shows good agreement.

Content

1 Introduction

As a part of the technical development, the tendency in the railway sector is, among other things, to aim towards higher speeds for passenger trains. With a faster and more comfortable railway travel, the train can be a respectable competitor to other means of transportation. In order to permit and achieve higher speeds, measures must be taken to develop and improve both the vehicle and the track. This simulation study concentrates on the modifications and improvements that have to be made to the vehicles.

One of the necessary measures is to have a dynamically stable running behaviour, i.e. to avoid self-generated lateral vibrations, often called "hunting". Furthermore, the trains should not generate too high track forces and rail wear in curves, i.e. trains should still be "track-friendly". All this is to be achieved even when the speed is increased.

1.1 The Green Train

In 2004 the research and development program "GrönaTåget" (GreenTrain) was initiated by the Swedish National Rail Administration (Banverket). Leading companies in the railway industry as well as universities and consultants in Sweden are co-operating with Banverket in order to develop the next generation of high-speed trains for Nordic conditions. The expected and desired speed is in the range of 250–300 km/h. The project includes an aim for building up and maintaining a wide competence, as well as an extensive exchange of knowledge and information. There is also an ambition of being able to strengthen the possibilities to influence and contribute to the European research and standardization. The project is due to continue until at least 2011 [8].

One part of the project is to develop and design a bogie that enables higher speeds and yet is considered to be "track-friendly", particularly in curves. Suspension and damping parameters should be developed in order to fulfil the requirements of safety and comfort, according to international standards. This work is mainly performed at the Royal Institute of Technology (KTH) in Stockholm, Sweden, in close co-operation with Bombardier Transportation.

1.2 Track friendliness

Even if higher speeds are desired on an existing track the vehicles should be track-friendly, i.e. they should not deteriorate the track too fast. Accordingly, vehicles should most likely have modest axle-load and unsprung masses.

Furthermore, the suspension in the bogie must be designed to maintain running stability at higher speeds, i.e. to avoid violent lateral vibrations – so-called hunting. Normally, this is done by introducing a quite stiff guidance of wheelsets in the bogies, combined with an appropriate yaw damping between bogie and carbody. Sometimes also the sprung and unsprung masses of the bogie have to be reduced. However, the increased guidance stiffness will reduce the curving ability of the vehicle, thus contributing to higher lateral forces between vehicle and track, and also an increased amount of wear and rolling contact fatigue (RCF) of rails (and wheels).

If the vehicles run on a perfect and robust track with very large curve radii (say ≥ 2500 m), specially built for dedicated high-speed trains, the increased forces and wear would not be a major problem. However, if the vehicles have to run to a large extent on existing older tracks, sometimes shared with heavy freight trains, the situation would become worse.

Firstly, on existing track there are still curves in the "normal" range, i.e. with curve radii from 250 m up to 1500 m, which may generate excessive wheel and rail wear with stiffly guided wheelsets, in particular at dry conditions. Secondly, too high axle-loads or unsprung masses would generate high dynamic vertical forces which, in turn, would generate excessive deterioration of the track (geometrical errors and component fatigue). Thirdly, the high-speed trains combined with heavy freight trains would generally cause worse track conditions that would lead to a more deteriorated track and higher track forces, eventually resulting in track maintenance problems and possibly also safety problems. Further, the Nordic winter climate with frost and frost upheaval would generate additional problems.

The conclusion is that track-friendly high-speed rail vehicles should generate modest track forces (laterally and vertically) as well as a modest amount of rail wear. These rail vehicles should also be capable of running on a "non-perfect" track without adverse effects on vehicle and passengers. This is particularly important if the vehicles have to run at high curving speeds (i.e. high cant deficiency, resulting in high lateral accelerations), where the most demanding situation is to run with tilting trains (cant deficiency up to about 300 mm).

1.3 Aim of the study

The aim of the present study is to find a bogie configuration that maintains a stable running behaviour on straight track and in larger curve radii at higher speeds, but also acceptably low track forces and amount of wear in curves with smaller curve radii. This report will describe the simulations being performed in the multi-body simulation tool SIMPACK. Furthermore, conclusions from the results will be presented. Finally, the simulation model and methodology are validated through comparisons with test data.

The issues of vertical track forces, axle-load and unsprung mass are not covered in the present study.

2 Assessment quantities and related limit values

In the process of evaluating and approving a new rail vehicle there are many requirements that have to be fulfilled in order to get it qualified to run on track, without any risks of exceeding limit values for safety, track fatigue and running behaviour. According to the international standards specified in the UIC code 518 there are, above all, requirements on track forces and accelerations that have to be met when evaluating running behaviour and lateral dynamic stability. All quantities and limit values in Section 2.1 refer to UIC 518 [15] and EN 14363 [4]. The vehicle model used in the present simulation study has a total mass of 61 tonnes, equally distributed on four axles.

2.1 Track forces and running stability

As running safety criterion the sum of lateral guiding forces per axle, ΣY (also known as track shift force S), is filtered with a sliding mean over 2 m in 0.5 m increments;

$$(\Sigma Y_{2m})_{lim} = \alpha\left(10 + \frac{P_0}{3}\right) \quad [\text{kN}] \tag{2-1}$$

where P_0 (axle-load) is expressed in kN. The constant α is set to 1, since a passenger vehicle is assumed. With the total vehicle mass of 61 tonnes, resulting in a static load on each axle of $61/4 \cdot 9.81 \approx 150$ kN, the limit value of the safety criterion is set to 60 kN.

Another measure for safety is the derailment criterion $(Y/Q)_{2m}$, describing the ratio between the lateral and vertical wheel forces for the guiding wheelset, calculated as a sliding mean over 2 m in 0.5 m increments. The limit value is 0.8 for radii larger than 250 m.

In order to minimize track fatigue and wheel/rail wear, limit values of lateral and vertical forces are specified in the international standards as well. The dynamic vertical force is restricted by $Q_{dyn} = 90 + Q_0$, where Q_0 is the static wheel-load, in this case 75 kN. Hence, Q_{dyn} shall not exceed 165 kN. In small-radius curves the quasi-static lateral guiding force Y_{qst} is restricted to 60 kN and the quasi-static vertical force Q_{qst} is limited to 145 kN. According to UIC 518, small-radius curves are defined as 250 m $\leq R \leq$ 600 m.

One of the main issues in UIC 518 (and EN 14363) is track forces in small-radius curves, while medium-radius curves (600–1500 m) are mainly left without attention. However, in a yet not published study on track deterioration in Sweden it is shown that *medium radius curves have a significant contribution to rail wear, due to the quite common occurrence of such curves.* About 60 % of the calculated cost for wear and rolling contact fatigue, due to passenger trains, occur in these curves. In particular, the Swedish rail network has a lot of curves within the radius intervals 580–600 m and 980–1000 m. Curve radii up to 400 m only contribute to 10-15 % of the total cost for rail surface damage. This is due to the simple fact that only about 0.5 %

of the passenger traffic volume (in ton-km) is run in curves with radii less than 400 m. It is likely that future Swedish track-access charging will reflect these circumstances, i.e. a considerable amount of the "track deterioration" charge will be due to medium-radius curves, being differentiated with respect to the actual vehicle's characteristic performance.

Furthermore, assessment of the running stability of a vehicle is implemented on straight and/or large-radius curve track. The parameter $(S\Sigma Y)_{\mathrm{lim}}$, which is band-pass filtered around the instability frequency $f_0 \pm 2$ Hz, in combination with a sliding *rms* (root mean square) over 100 m in 10 m increments, is determined as half of the safety criterion mentioned above, i.e. 30 kN in this case.

Two types of lateral periodic motions

At higher speeds, in combination with high equivalent conicity (explained in Section 3.6), there is an imminent risk of unstable running behaviour, due to inertia forces of the wheelsets and coupled masses. This phenomenon is regarded as high-frequency instability, normally occurring at frequencies between 4 and 10 Hz, at vehicle speeds above 100 km/h, usually on straight track.

Another, not so familiar, type of instability is the low-frequency instability, caused by a relatively low wheelset hunting frequency that coincides with a natural frequency of the carbody and its suspensions. Also this phenomenon occurs at vehicle speeds above 100 km/h, but with a frequency of 1–2 Hz, and more commonly at lower conicities, typically 0.02 to 0.05 [1]. However, this type of instability has largest impact on the ride comfort in the carbody and is not obviously detected when assessing track forces. The term low-frequency instability can, however, be somewhat misleading, so henceforth the phenomenon will be referred to as "low-frequency periodic motions" and is further evaluated in Section 4.3, concerning ride comfort. These motions may occur both on straight track and in large-radius curves.

Lower track forces or higher degree of stability

When increasing the vehicle speed (and hence the risk of instability) the margin for a stable running behaviour must be large enough. However, a vehicle with wheelset guidance properties that ensures the most stable running behaviour may generate larger track forces and wheel/rail wear. Hence, when reaching an acceptable degree of stability of the vehicle the generated track forces and amount of wheel/rail wear should be crucial for determining the properties of the wheelset guidance.

2.2 Wear and energy dissipation

The major problems in small-radius curves are the contact conditions combined with high lateral forces, generating fatigue as well as wheel and rail wear. A common way of describing the relative wheel/rail wear for different conditions is to use the wheel/rail energy dissipation as a *wear index*, normally used as a comparative number. The energy dissipation $\bar{E}$ is defined as the energy loss per metre travelled distance, relating to creep forces (F_ξ and F_η) and spin moment M_ζ, as well as sliding velocities (v_ξ and v_η) and rotational sliding velocity ω in the wheel/rail contact [1];

$$\bar{E} = F_\xi \cdot \frac{v_\xi}{v} + F_\eta \cdot \frac{v_\eta}{v} + M_\zeta \cdot \frac{\omega}{v} \qquad \text{[Nm/m]} \tag{2-2}$$

where v is vehicle forward speed [m/s]. The relations v_ξ/v and v_η/v are the so-called creepages in longitudinal and lateral directions and ω/v is the so-called spin, or rotational creep.

However, the spin moment contribution (M_ζ, with corresponding rotational sliding velocity ω) is not included in the present simulations performed by SIMPACK, since its influence is considered to be negligible (however, studies at KTH indicate that the spin contribution could be in the order of 10 % in typical cases [17]). The energy dissipation, except the spin moment contribution, is usually called the *wear number*.

Pearce and Sherratt [12] have developed a wear predicting method, assuming that the amount of flange wear (worn-off area per rolled km) is depending on the wear number of the contact patch. This has been assessed by Roger Enblom at KTH and compared with a comprehensive method developed by Tomas Jendel, which is based on Archard's wear model. The methods agree remarkably well [6]. The model of Pearce and Sherratt indicates that the transition from mild to severe wear occurs between wear numbers 100 and 200 Nm/m [12].

Further British research has developed a methodology for predicting rolling contact fatigue (RCF) by means of the wear number of the contact patch between wheel and rail. A good correlation between predicted and observed damage on rails has been shown [3]. This model is also concidering wheel-rail abrasive wear on a similar (but not exactly the same) way as the previosly mentioned model [12].

It should be pointed out that too little flange wear is not desired either. In such cases there is a risk that the wheel tread is worn instead of the flange (developing a "hollow" wheel tread), resulting in risks of rapidly developing high values of equivalent conicity; see Section 3.6. A "well balanced" wear between wheel flange and tread is desired.

2.3 Ride comfort

There are different methods for measuring and evaluating ride quality and comfort, e.g. discomfort. A very common approach, which also is used in the present simulation work, is to use the *Wz* (Wertungszahl), which originates from German research [14].

Wz is evaluated from lateral and vertical accelerations in specific measuring points on the carbody floor. It is a frequency-weighted root mean square-value of accelerations, in a logaritmic scale, which can be calculated according to

$$Wz = 4.42(a^{wrms})^{0.3} \tag{2-3}$$

where a^{wrms} is the *rms*-value of the frequency-weighted acceleration.

For passenger traffic at longer distances *Wz* in the range of 2.0–2.6 are regarded as acceptable [1]. *Wz* = 2 is considered as "good" while *Wz* = 3 is considered as "tolerable" according to [14].

Another method for assessing the ride comfort is to evaluate the five-second *rms*-values of the frequency-weighted accelerations in longitudinal, lateral and vertical directions, according to the European standards EN 12299;

$$a_{Xj}^{Wi}(t) = \left[\frac{1}{T} \cdot \int_{t-T}^{t} (\ddot{x}_{Wi}^{*}(\tau))^2 \, d\tau\right]^{0.5} \tag{2-4}$$

$$a_{Yj}^{Wi}(t) = \left[\frac{1}{T} \cdot \int_{t-T}^{t} (\ddot{y}_{Wi}^{*}(\tau))^2 \, d\tau\right]^{0.5} \tag{2-5}$$

$$a_{Zj}^{Wi}(t) = \left[\frac{1}{T} \cdot \int_{t-T}^{t} (\ddot{z}_{Wi}^{*}(\tau))^2 \, d\tau\right]^{0.5} \tag{2-6}$$

where T = 5 s and t is a multiple of 5 seconds and $\ddot{x}_{Wi}^{*}$, $\ddot{y}_{Wi}^{*}$ and $\ddot{z}_{Wi}^{*}$ are the frequency-weighted accelerations. On the floor level, *rms*-values of frequency-weighted accelerations of 0.2–0.3 m/s^2, in lateral and vertical direction, are considered to be comfortable [5].

The frequency-weighting in the EN is somewhat different from *Wz*, in particular in lateral direction. *Wz* considers the most sensitive frequency (for lateral and vertical vibrations) to be 3–8 Hz, whereas EN regards the "sensitivity peak" at 0.5–2 Hz in the lateral direction.

3 Methodology and conditions

3.1 Simulation software

The vehicle model evaluated by simulations is built-up in the multi-body systems (MBS) simulation software SIMPACK. The bodies of the structure are interconnected to each other with spring and damping elements, and can be modelled as either rigid or elastic bodies. The model is then used to automatically generate the equations of motion, which are part of producing a solution (e.g. time integration) [9].

The wheel-rail contact is a very essential and complex issue when performing rail vehicle simulations. In order to simplify and save simulation time, it can be regarded as partly a normal contact problem, partly a tangential. However, the normal and tangential forces depend on and influence one other, but in order to be able to treat them separately the Hertzian contact theory is normally used, meaning that the contact area is regarded as an ellipse [7]. The tangential forces in the contact patch are calculated when the corresponding normal forces have been found. The method for calculating the normal forces depends on which contact conditions being chosen in the vehicle model. SIMPACK uses Kalker's simplified non-linear theory, with the numerical algorithm FASTSIM, when calculating the tangential forces [10][11].

3.2 Vehicle model

The multi-body dynamics vehicle model used in the present study has originally been developed by Bombardier Transportation and constitutes a one-car vehicle with two bogies, and models a Swedish Regina train, see Figure 3-1 (which here is a two-car train, with the working title Regina 250). The main part of the simulation work has been performed with a rigid carbody, what also will be shown in the results, in Chapter 4 and in Appendix B.

Figure 3-1 The Regina 250 two-car train unit. *Photo: Evert Andersson.*

The carbody is connected to two motor bogies (A and B) through the secondary suspension, modelled by a non-linear airspring. Additionally, the secondary suspension consists of,

- in lateral direction; two non-linear dampers per bogie, having linear spring-damper elements in series, and two bumpstops in each bogie, represented by non-linear spring elements with damping,
- in vertical direction; two non-linear dampers per bogie, having linear spring-damper elements in series,
- in longitudinal direction; one traction rod in the middle of each bogie, represented by spring-damper components in parallel,
- in yaw direction; one non-linear yaw damper at each side of the bogie, having linear spring-damper elements in series,
- in roll direction; one anti-roll bar per bogie with linear stiffness.

The yaw dampers have a so-called blow-off level, which means that there is a specific damping rate up to a certain force acting on the damper. Above the blow-off level the damping rate is decreased.

The wheelsets and the bogie frame are interconnected by the primary suspension. Between each end of the wheelset (i. e the axle journals) and the bogie frame there are non-linear spring elements with damping. Furthermore, the primary suspension consists of four separate non-linear axel-box dampers per bogie, having linear spring-damper elements in series, connecting axel-boxes and bogie frames.

In the present simulation study principally two different bogie configurations (called "soft" and "medium") are tested in order to evaluate their influence on running stability on straight track and their track friendliness in smaller curves. These variations regard the guidance stiffnesses of the wheelset on the bogie frame, in particular in the longitudinal direction. Further, two reference bogie configurations have been included in the evaluation for comparisons ("original soft" and "stiff"), where the acronyme "original soft" is used for the original soft Regina (bogie for $v \leq 200$ km/h). The resulting longitudinal stiffness values of the primary suspension for the mentioned configurations are listed in Table 3-1.

Table 3-1 Resulting longitudinal primary suspension stiffness for the four different bogie configurations.

	Primary suspension stiffness (MN/m)			
	Original soft	Soft	Medium	Stiff
Longitudinal stiffness	6.6	8.0	15	40

The mass properties of carbody, bogies and transformer of the modelled vehicle can be seen in Table 3-2. The bogie masses include wheelsets and brakes, suspension components, traction motors and transmission gears. Bogie A is the leading motor bogie, including the weight of a magnetic brake, whereas bogie B is the trailing motor bogie without magnetic brake.

Table 3-2 Mass properties of the modelled vehicle.

	Carbody (service)*	Transformer	Bogie A	Bogie B
Mass (kg)	41422	3200	8605	7906

* excl. weight of transformer

3.3 Track model

The track model used in the simulation program SIMPACK is a discrete track model, moving with the wheelsets and flexibly connected to a ground frame by lateral and vertical stiffness and damping elements. There is also allowance for a linear "contact stiffness" and damping between wheels and rails. This is a quite simple and rough model of the real matters, neglecting, for example, the lateral flexibility of both the rails and the wheels (including the effects of axle bending). Therefore, a careful tuning of the model is necessary in order to reflect reality as far as possible.

The tuning was made at the initial stage, in order to generate both realistic lateral stability of the bogies as well as realistic lateral impact forces that coincide with previous test results from the original Regina train.

After this "tuning" the lateral stiffness between track body and ground is set to 30 MN/m and the "lateral contact stiffness" between wheel and rail is set to 22 MN/m, which together generate a resulting lateral stiffness between wheel-axle and gound at 12.7 MN/m. In the "contact stiffness" now also approximate effects of rail and wheel lateral flexibilities are included.

Partly thanks to "tuning" of the simple wheel-track interface, the SIMPACK simulations have later shown good agreement with real vehicle behaviour on the track; see further Section 4.4.

3.4 Track geometry

For the evaluation of running behaviour, different cases of track geometry have been implemented in the simulations. Assessment of the vehicle's stability has been performed at high speed on straight track and in large-radius curves ($R \geq 2400$ m), whereas track friendliness and wheel/rail wear have been evaluated in curves with smaller radii ($R \leq 1500$ m). In all track geometry cases the simulated ride comfort has been observed as well.

The modified vehicle is supposed to have the ability to run stable at higher speeds. For the present work the goal is to have a permissible speed of 250 km/h. As a safety limit the vehicle is going to be classified for 250 km/h + 10 % = 275 km/h, according to [15]. At an early stage of the project the running stability of the vehicle has been simulated on straight track at higher speeds. The simulations on straight track have been performed at distances between 760 and 1500 metres.

The running stability has also been evaluated in large-radius curves, with radii of 2460, 3200 and 4900 metres. Apart from straight track and different curve cases with varying speed, cant and track plane acceleration (i.e. cant deficiency) the simulations evaluating the vehicle's running behaviour have also included different cases of equivalent conicity (explained further in Section 3.6).

In the cases of small-radius curves the track forces and wear numbers (i.e. simplified relation to wheel/rail wear) have been evaluated and thus part of the track friendliness of the vehicle. The small radii evaluated in the simulations are 300 and 600 metres. Furthermore, curves with somewhat larger radii; 900 and 1200 metres, here defined as medium-radius curves, have also been included in the assessment of track friendliness. The curve cases have been evaluated with speed and cant corresponding to approximate track plane accelerations of 0.8, 1.1 and 1.2 m/s^2 (i.e. cant deficiency of 122, 168 and 183 mm, respectively, on standard track gauge).

For comparison, the original soft Regina train is made and approved for 150 mm cant deficiency, while a future "GreenTrain" should be approved for 165 mm, both with a margin of +10 %.

The curve cases with small- and medium-radius curves are listed in Table 3-3 and the evaluated large-radius cases are shown in Table 3-4.

Table 3-3 Curve cases for small- and medium-radius curves used in the simulations.

	Radius (m)	Speed (km/h)	Cant (mm)	Track plane acc. (m/s^2)	Cant deficiency (mm)
Small radius	300	83	150	0.8	121
	300	89	150	1.1	162
	300	92	150	1.2	183
	600	117	150	0.8	119
	600	127	150	1.1	167
	600	130	150	1.2	182

Table 3-3 Curve cases for small- and medium-radius curves used in the simulations.

	Radius (m)	Speed (km/h)	Cant (mm)	Track plane acc. (m/s^2)	Cant deficiency (mm)
Medium radius	900	144	150	0.8	122
	900	155	150	1.1	165
	900	159	150	1.2	182
	1200	166	150	0.8	121
	1200	179	150	1.1	165
	1200	184	150	1.2	183

Table 3-4 Curve cases for large-radius curves used in the simulations.

	Radius (m)	Speed (km/h)	Cant (mm)	Track plane acc. (m/s^2)	Cant deficiency (mm)
Large radius	2460	250	150	1.0	150
	2460	260	150	1.14	174
	3200	202	150	0	0
	3200	260	150	0.65	100
	3200	275	130	1.0	150
	3200	275	115	1.1	164
	4900	250	150	0	0
	4900	275	83	0.65	100

3.5 Track irregularities

In the present simulation study a thorough classification of the track irregularities and the related limit values has been performed. Two different types of track irregularity levels have been defined, namely (1) *safety* and (2) *comfort track*.

The *safety track* is specified according to UIC 518 (i.e. QN1, QN2, QN3), where the track irregularities are band-pass filtered within the wavelength interval 3–25 m. Thereafter, they are scaled to reach the limit value for the quality level QN3, for each speed range, corresponding to a cant deficiency of 165 mm (track plane acceleration of approximately 1.1 m/s^2) when the cant is 150 mm. However, the maximum nominal permissible speed is

250 km/h, since this is the limit speed for which the vehicle will be classified in the present project. QN3 is defined as QN3 = 1.3 · QN2, according to UIC 518 and Table 3-5.

For example, if the curve radius is R = 900 m, cant D = 150 mm and permissible cant deficiency I = 165 mm the vehicle speed v is calculated to 155 km/h (43 m/s), according to the equation $v^2 = (D + I) \cdot R \cdot g / 2b_0$, where g = 9.81 m/s^2 and b_0 = 750 mm [1]. Hence, according to Table 3-5 the maximum isolated track error is QN3 = 1.3 · 10 = 13 mm in vertical direction and QN3 = 1.3 · 8 = 10.4 mm in lateral direction, for the speed range in question.

Table 3-5 Permissible peak values for isolated track irregularities, vertical and lateral alignment, according to UIC 518, for assessment of "safety track". *QN3 = 1.3 · QN2 [15]*

	Maximum isolated error (mm): Vertical alignment			Maximum isolated error (mm): Lateral alignment		
Speed (km/h)	QN1	QN2	QN3	QN1	QN2	QN3
$200 < v \leq 300$	4.0	8.0	10.4	4.0	6.0	7.8
$160 < v \leq 200$	5.0	9.0	11.7	5.0	7.0	9.1
$120 < v \leq 160$	6.0	10.0	13.0	6.0	8.0	10.4
$80 < v \leq 120$	8.0	12.0	15.6	8.0	10.0	13.0
$v \leq 80$	12.0	16.0	20.8	12.0	14.0	18.2

The *comfort track* is specified according to Banverket standards [2]. Track irregularities with maximum values kept under the limits are considered to provide good or at least acceptable ride comfort. The track irregularities are band-pass filtered, as in the previous case, but here scaled with respect to standard deviations within each speed range, see Table 3-6. The speed ranges correspond to trains specified as Category A, i.e. an allowed cant deficiency of 100 mm (track plane acceleration 0.65 m/s^2) when the cant is 150 mm, according to Banverket [2]. Also in the case of evaluating the comfort track the maximum nominal permissible speed is 250 km/h, which corresponds to track quality class K0 for vehicle speed above 145 km/h.

The present study has also taken peak amplitudes into consideration, when assessing the comfort track, by adopting the following condition; if the corresponding maximum peak amplitudes of the track irregularities turn out to exceed the limit values with more than 33 %, according to Table 3-7, the track irregularities are scaled according to these values (i.e. limit amplitude plus 33 %).

For example, if the curve radius is R = 900 m, cant D = 150 mm and permissible cant deficiency I = 100 mm the vehicle speed is calculated to 138 km/h (38 m/s), according to the reasoning and figures mentioned above. This corresponds to the track quality class K1 (according to Table 3-6) with maximum permissible standard deviation $\sigma_H = 1.3$ and $\sigma_P = 1.2$, and isolated peaks at 6 and 4 mm, in vertical and lateral directions, respectively. Hence, the isolated peaks should not exceed $6 \cdot 1.3 \approx 8.0$ mm in vertical and $4 \cdot 1.3 \approx 5.3$ mm in lateral direction, according to Table 3-7.

Table 3-6 Permissible standard deviations for track irregularities, vertical and lateral, according to Banverket, for assessment of "comfort track" [2].

Quality class	Speed (km/h) Category A	Standard deviation (mm) σ_H, vertical	σ_P, lateral
K0	145-	1.1	1.1
K1	125-140	1.3	1.2
K2	105-120	1.5	1.3
K3	75-100	1.9	1.7

Table 3-7 Permissible peak values for track irregularities, vertical and lateral, according to Banverket, for assessment of "comfort track" [2].

Quality class	Speed (km/h) Category A	Isolated peaks (mm) Vertical	Lateral
K0	145-	6	3
K1	125-140	6	4
K2	105-120	7	5
K3	75-100	10	6

3.6 Equivalent conicity

The contact condition between wheel and rail is described through the dimensionless parameter *equivalent conicity*. It is a function of wheel profile (i.e. the wheel contour in contact with the rail), wheel inside gauge, flange thickness, rail profile (i.e. the contour of the rail head), rail inclination, track gauge and relative lateral displacement. Usually, the equivalent conicity is a non-linear function, particularly for smaller track gauge than the nominal, and is defined as

$$\lambda_{eq} = \frac{\Delta r_r - \Delta r_l}{2\Delta y} \tag{3-1}$$

where Δr is the change in rolling radius and Δy is the lateral displacement (indices r and l denote right and left wheels, respectively). The wheelset oscillates laterally with plus-minus some millimetres in relation to the centre line of the track. Therefore, the equivalent conicity should be determined at a reasonable lateral displacement, for example ±3 mm. The equivalent conicity according to Equation (3-1) should, according to UIC 519, be further processed by a special algorithm in order to take the non-linearity into consideration and to produce a representative "average" over a lateral motion cycle (limited by the desired maximum lateral amplitude) [16].

Normally, a higher value of equivalent conicity may generate stability problems on straight track at higher speeds. On the other hand, a high λ_{eq} can be more advantageous when negotiating small-radius curves. According to UIC 518 the equivalent conicity should not exceed 0.25 when travelling on straight track and in large-radius curves in the permissible speed range $250 < v \leq 280$ km/h. In the speed range $230 < v \leq 250$ km/h equivalent conicity is allowed up to 0.3 [15]. However, in the present simulation study values up to 0.4 have been studied in the high-speed range, intended as a safety margin. These equivalent conicities refer to the "averaged" conicities over a lateral motion cycle with an amplitude of ±3 mm, according to UIC 518 and UIC 519.

Two different types of equivalent conicity have been taken into consideration [13] and are in the present simulation study simply called Type 1 and Type 2. Characteristic for Type 1 is a principally constant value for smaller lateral displacements, but growing as the lateral amplitude increases. Type 2 is distinguished by high values of conicity for small values of lateral displacement. Consequently, the two types can have the same conicity value at a given lateral displacement (3 mm according to UIC 518), but show significant differences at other amplitudes, as shown schematically in Figure 3-2. Equivalent conicity is in the simulation tool SIMPACK calculated by a harmonic linearization method, with quasi-elastic contact, according to UIC 519.

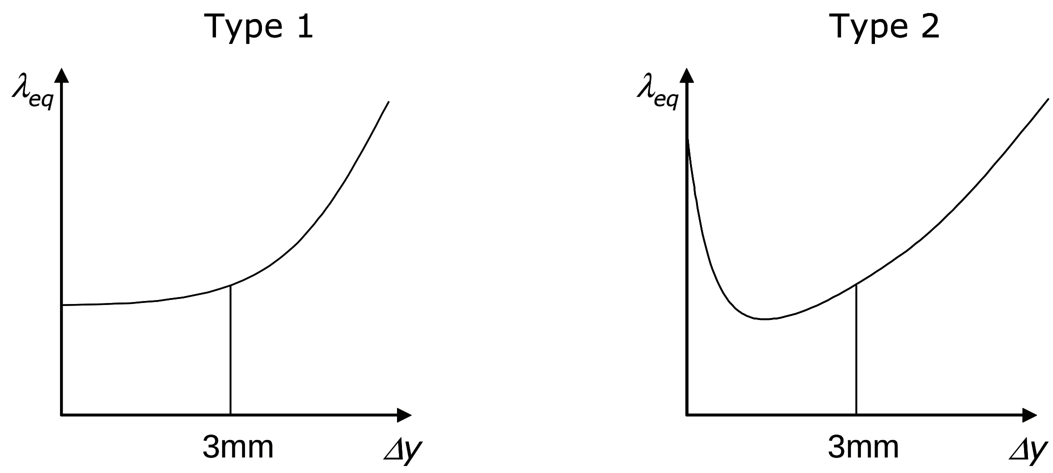


Figure 3-2 Two different types of equivalent conicity, having the same value at 3 mm amplitude, but with significant differences at other amplitudes.

Ten different contact conditions (with various wheel and rail profiles, track gauge and rail inclination) have been regarded in the present simulation study, when assessing running stability on straight track and in large-radius curves. These are presented in Table 3-8.

It should be pointed out that the actual conditions (wheel and rail profiles, track gauge and rail inclination) generating a certain equivalent conicity is of minor interest; the interesting matter is the resulting conicity value of either Type 1 or Type 2.

Table 3-8 Contact geometry combinations, resulting in different values of equivalent conicity.

Wheel profile	Rail profile	Track gauge (mm)	Rail inclination	Type	λ_{eq}	Design-ation
S1002	UIC 60	1440	1:30	1	0.01	001T1
S1002	UIC 60	1437	1:30	1	0.02	002T1
S1002	UIC 60	1436	1:30	1	0.067	0067T1
BR_P8-dense	UIC 60	1440	1:40	1	0.1	01T1
S1002	R_EN_52E1	1438	1:40	1	0.2	02T1
S1002	R_EN_52E1	1432	1:40	1	0.3	03T1
S1002	R_EN_52E1	1430	1:40	1	0.4	04T1
S1002	UIC 54E	1436	1:30	2	0.2	02T2
WornWheel07	UIC 60	1437	1:30	2	0.3	03T2
S1002	UIC 60	1430	1:30	2	0.4	04T2

For simulations in small- and medium-radius curves other wheel/rail combinations have been tested. Three different rail profiles with different degree of wear (slightly, moderately and heavily worn rail profiles). However, it turned out that the different combinations did not significantly affect the results. Therefore, the wheel/rail combination with original, unworn profiles (S1002, UIC 60) were chosen for simulations in small- and medium-radius curves. The track gauge has been varied in these curves with 1445 mm in curves with radius 300 and 600 m, 1442 mm for 900 m and 1440 mm for 1200 m curve radius, respectively.

4 Evaluation of parameter studies and simulation results

This chapter summarizes the evaluation of parametric studies that have been performed in the present simulation study. A huge amount of simulations have been performed, with varying parameters, such as bogie configurations (varied wheelset guidance stiffness), track geometry, track irregularity, equivalent conicity, yaw damping and other damping. A representative selection will be presented with references to illustrating diagrams in this chapter as well as in Appendix B. It should be noticed that the simulations evaluating track forces, wear and corresponding wear number have been performed on the so-called *safety track*, whereas ride comfort is evaluated according to *comfort track* (see Section 3.5 for track definitions).

4.1 Track forces and running stability

Simulations with varying wheelset guidance stiffness in the bogies have been performed at higher speeds on straight track in order to evaluate the dynamic behaviour for the different bogie configurations and hence detect possible instability. The two different bogie types being compared (as well as the two reference bogie configurations) have already been defined in Section 3.2.

With a stiffer wheelset guidance, in combination with increased yaw damping, a more stable running behaviour can be secured on straight track at higher speeds, which is at least true in the lower stiffness range. When reaching an acceptable stability margin the generated track forces and wheel/rail wear are instead regarded as design criteria (as already discussed in Section 2.1). This is illustrated with the diagram in Figure 4-1, where the simulations are performed on straight track at a vehicle speed of 275 km/h. The track irregularities are scaled according to *safety track*, defined in Section 3.5, and the equivalent conicity is 0.3, Type 2.

After approximately 2 seconds of the simulation time the track irregularities consist of a rather large lateral peak, which initiates the lateral motion. However, the bogies with soft and medium guidance stiffness manage to regain a stable running behaviour as a result of a stiffer wheelset guidance, whereas the instability pattern continues for the original soft bogie configuration. It was easily proven that the stiffness parameters in the original soft bogie configuration are not sufficient for stable running behaviour at higher speeds; therefore, only the soft and the medium bogie configurations will be regarded and compared in the following examples.

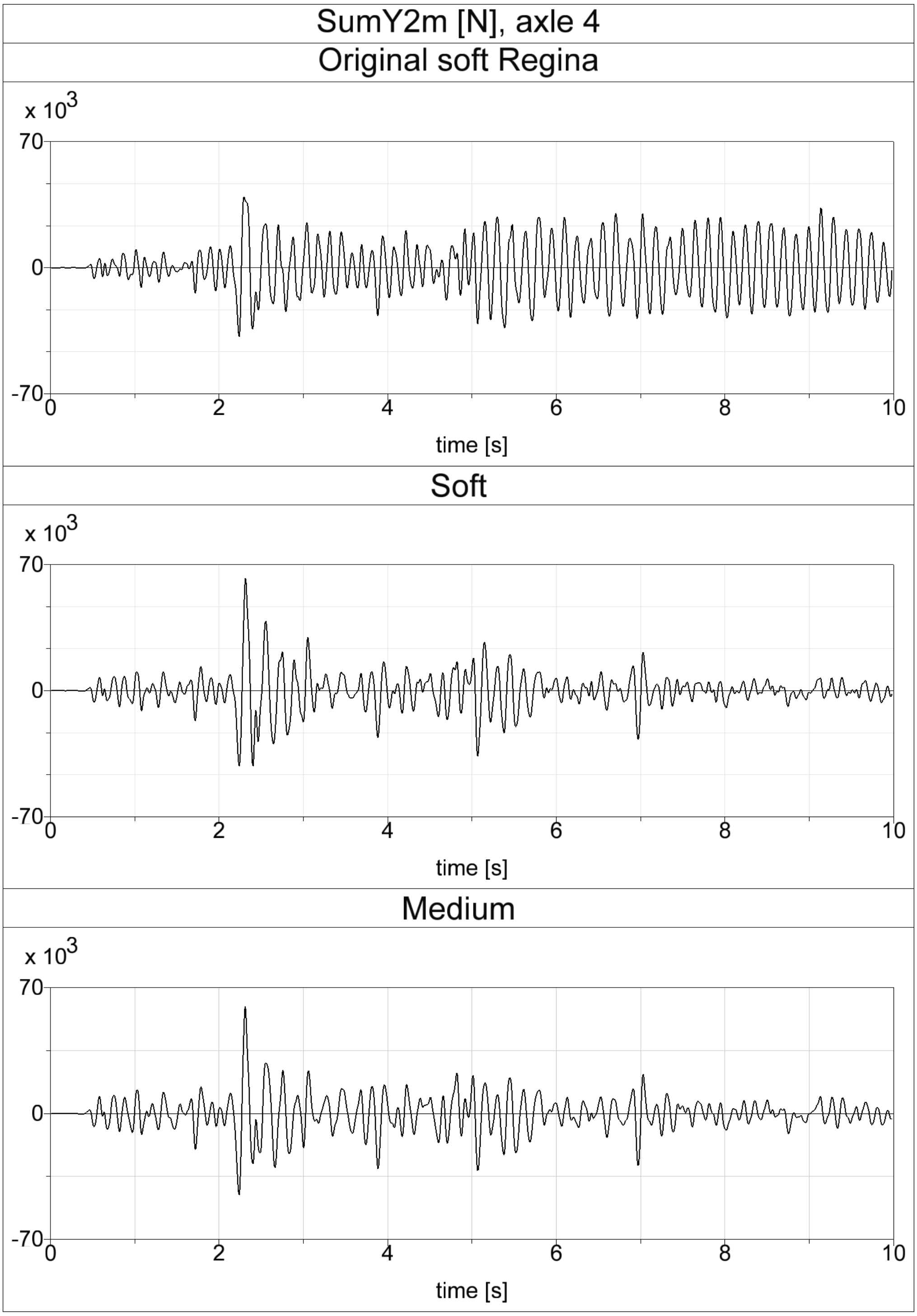


Figure 4-1 Simulations of track shift force (ΣY low-pass 20 Hz, sliding mean 2 m) on axle 4 for bogie configurations "original soft" Regina, "soft" and "medium". *Simulations are performed on straight safety track at 275 km/h with equivalent conicity 0.3, Type 2.*

Accordingly, the running stability of the vehicle is secured not only by increasing the stiffness in the primary suspension, but also by increasing the yaw damping forces. A parametric study has been performed on the influence of the blow-off level in the yaw dampers, which has been varied between 13 and 25 kN. The sliding *rms*-value over 100 m of the track shift forces (measure of instability, see Section 2.1) decreases with increasing blow-off level in the yaw dampers, which is illustrated in Figure 4-2. The example is shown for the soft bogie configuration running on straight safety track.

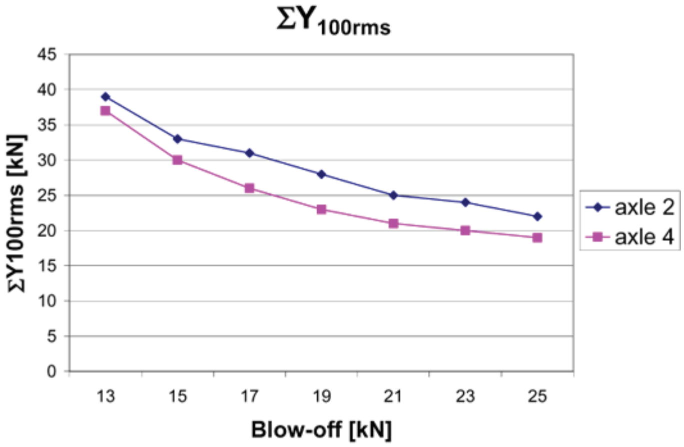


Figure 4-2 Maximum track shift forces (ΣY_{100rms}) with varying blow-off level in the yaw dampers. *Simulations are performed with soft bogie configuration on straight safety track at a vehicle speed of 275 km/h with equivalent conicity 0.3, Type 2.*

With an increased yaw damping there could be a risk of too high track forces when passing short switches. However, through simulations the influence of an increased yaw damping has been proven not to generate neither particularly high track forces nor rail wear.

The vehicle's running behaviour is strongly dependent on the equivalent conicity, explained in Section 3.6, where a high conicity value tends to generate higher track forces and accelerations. However, occasionally even a lower conicity value can generate rather high track forces. This can be seen in the diagrams presented in Figure B-1 (Appendix B), which represent simulations on straight *safety track* and at a vehicle speed of 275 km/h, with varying equivalent conicity of Type 1, from 0.01 to 0.4.

Additionally, comparisons between results from simulations performed with the two types of conicity (Type 1 and Type 2), with the soft bogie configuration, show that Type 2 generates slightly higher track forces. In Figure 4-3 results from six simulations are plotted, showing that despite the same conicity value at lateral amplitude ±3 mm (see Section 3.6), the track forces are higher for the cases with equivalent conicity Type 2.

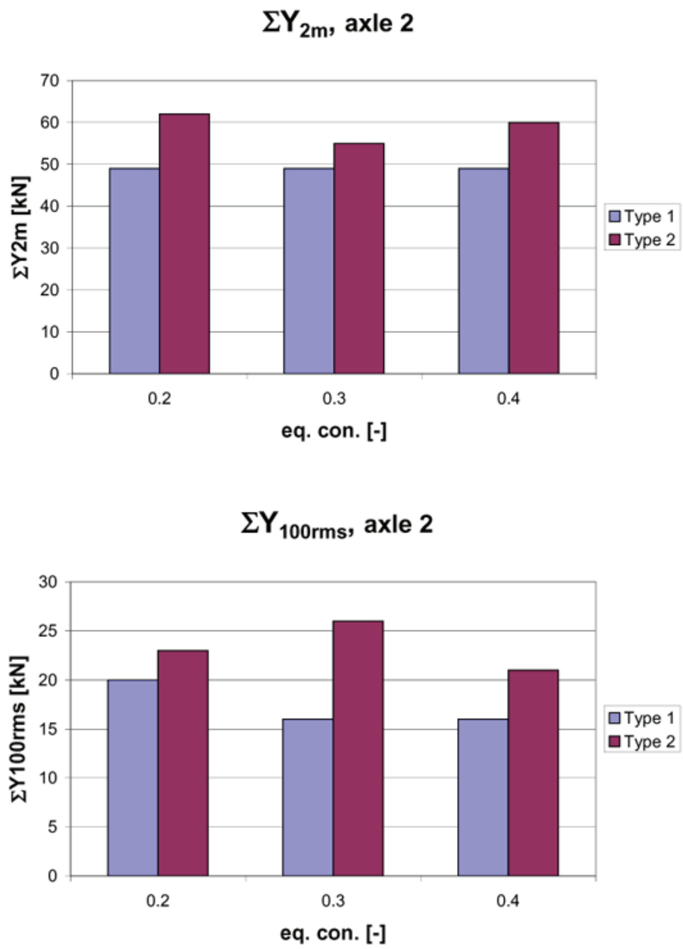


Figure 4-3 ***ΣY_{2m} (99.85-percentiles) and ΣY_{100rms} for simulations with varying equivalent conicity; 0.2, 0.3 and 0.4, Type 1 and Type 2.*** *Simulations with soft bogie configuration on straight safety track at a vehicle speed of 275 km/h.*

In order to compare the running behaviour and track forces of the soft and medium bogie configurations (in combination with increased yaw damping), the sliding mean over 2 m and the sliding *rms* over 100 m of the track shift forces have been evaluated. Results in Figure B-2 (Appendix B) show that the medium bogie configuration in general generates higher ΣY_{2m}-values than the soft bogie configuration. This could most likely be related to the stiffer coupling between bogie and wheelset, generating higher impact track forces when the bogie moves laterally on the quite rough track.

Furthermore, it can be observed that the medium bogie configuration generates higher ΣY_{100rms}-values for three cases with comparatively high equivalent conicity, but lower for the other cases. According to simulation results the soft bogie configuration shows slightly higher lateral bogie accelerations but generally somewhat lower track shift forces (ΣY) than the medium bogie configuration. This leads to a somewhat ambiguous analysis of the vehicle's running behaviour, since track shift forces as well as lateral bogie acceleration can be used as stability criteria, according to UIC 518. Hence, a general conclusion whether the medium bogie configuration is more stable than the soft one on straight track at higher speeds can not be drawn.

From a certain level, a stiffer wheelset guidance (in combination with increased yaw damping) generates larger track forces and has thus a negative impact on the vehicle's track friendliness, from a dynamic impact point of view. This is illustrated in Figure 4-4, showing track shift forces on all wheelsets from simulations performed in small- and medium-radius curves (300, 600, 900 and 1200 metres). The track irregularities are scaled according to *safety track* and the track plane acceleration is 0.8 m/s^2 in all curves. Mostly, the medium bogie configuration generates somewhat higher track shift forces in small- and medium-radius curves compared to the soft bogie configuration. However, there is some "spread" and uncertainty in the results, due to the stocastic nature of the track irregularities and the way the wheelsets hit these irregularities.

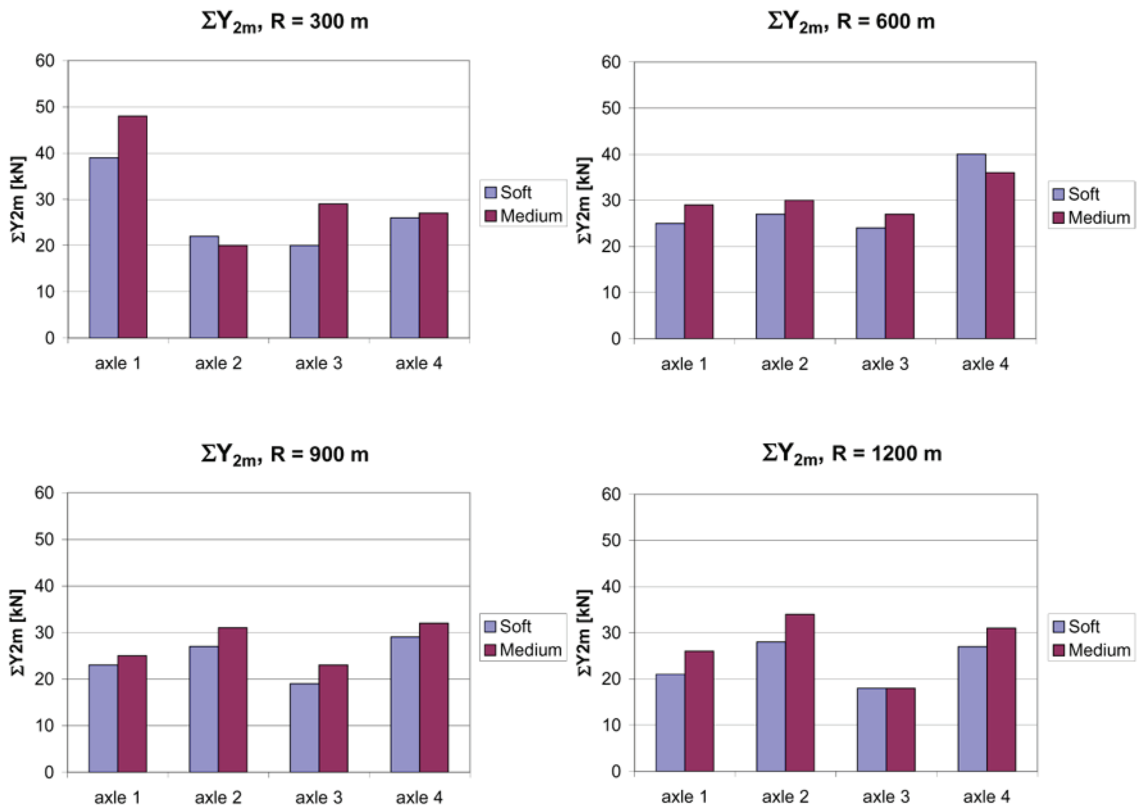


Figure 4-4 ***ΣY_{2m} (99.85-percentiles) on all four wheelsets of the soft and medium bogie configurations.*** *Simulations in curves with radii 300, 600, 900 and 1200 metres and track plane acceleration 0.8 m/s^2.*

4.2 Wear and corresponding wear number

In order to assess the track friendliness of the two bogie configurations, the wear number of the leading outer wheel is evaluated (where the energy dissipation is the highest). Here, also the stiff bogie configuration is included in the evaluation. The track irregularities are scaled according to *safety track* and the track plane acceleration is 0.8 m/s^2 in all curves. According to Section 2.2 the limit value where the transition from mild to severe wear starts from about 100 Nm/m and up. The results in Figure 4-5 indicate that the soft bogie configuration generates quite acceptable wheel and rail wear down to curve radius R = 600 m, whereas the medium bogie configuration merely manages R = 900 m for the same value of wear number. Additionally, the wear number for the medium bogie configuration in the curves with radii 600, 900 and 1200 metres is 50–90 % higher than for the soft bogie configuration. The corresponing increase in wear number from soft to stiff bogie configuration is usually 100–150 %. In this context it is worth mentioning that 600 and 900 m curves are the most important curve radius classes when considering the total accumulated abrasive wear and rolling contact fatigue damage on Swedish railway tracks; as earlier mentioned in Section 2.1.

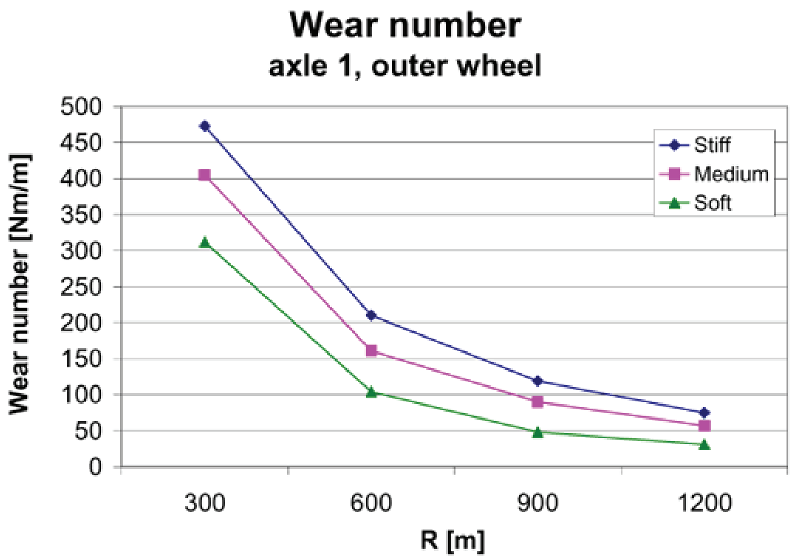


Figure 4-5 ***Wear number of the leading outer wheel for the soft, medium and stiff bogie configurations.*** *Simulations in curves with radii 300, 600, 900 and 1200 metres at track plane acceleration 0.8 m/s^2.*

4.3 Ride comfort

All simulations mentioned above have also been performed on the so-called *comfort track* (defined in Section 3.5), in order to evaluate the ride index Wz, thus the ride comfort. Wz is evaluated on the carbody floor above both bogie centres in lateral and vertical directions. However, the modifications of the wheelset guidance and yaw damping concern mainly longitudinal and lateral directions, so the ride comfort in vertical direction is not evaluated in this report.

Results from simulations being performed on straight track and in a large-radius curve with R = 3200 m, at a vehicle speed of 275 km/h and with track plane acceleration 1.1 m/s^2 in the wide curve, can be seen in Figure B-3 (Appendix B). A comparison of the two bogie configurations, "soft" and "medium", shows that they generate similar results on straight track as well as in the large-radius curve concerning lateral ride comfort. Remarkable is, however, that in the wide curve the Wz-levels are in the range of what is considered to be uncomfortable or even unpleasant, although the simulations are performed on *comfort track*. However, on straight track the lateral Wz indicates acceptable values for both bogie types.

The quite high lateral Wz in the wide curve is related to the so-called low-frequency periodic motions, mentioned in Section 2.1. Two cases with different equivalent conicity have been selected (0.067, Type 1 and 0.4, Type 2) for evaluation. When observing the diagrams in Figure B-4 (Appendix B), concerning conicity case 0.067, it is noticeable that the frequency content of the track forces is concentrated not only to 5.5 Hz, but also to the lower frequency 2 Hz. Moving upwards in the vehicle, to the carbody, a rather high lateral acceleration is generated at a dominant frequency of 2 Hz (generating the high Wz, seen in the previous example). Accordingly, low-frequency periodic motions are not obviously detected by evaluating track forces.

For the case with conicity 0.4, cf. Figure B-5 (Appendix B), the more dominant frequency for track forces is 6 Hz, whereas the distribution between 2 and 6 Hz is more equal for the lateral acceleration on the carbody floor. Hence, the generated high Wz for this case originates not only from low-frequency periodic motions, but also from the more common high-frequency motions.

Evaluation of lateral Wz has been implemented in small-radius curves as well, where the results can be seen in Figure B-6 (Appendix B). In these cases the difference between the two bogie types is more clear; the soft bogie configuration generates consistently higher Wz than the bogie with medium guidance stiffness. However, Wz is small and "comfortable" for all cases.

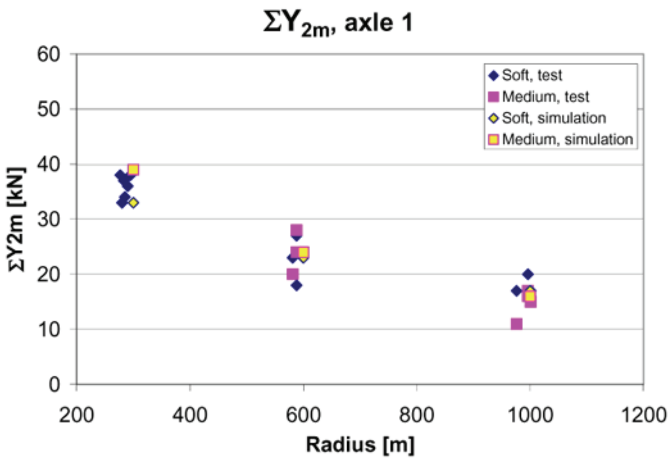
4.4 Validation

High-speed tests have been performed on various Swedish tracks during the summer 2006, using the soft and medium bogie configurations, respectively. During the tests a new Swedish high-speed record of 281 km/h was set, managed by the Regina 250 train unit equipped with a soft bogie configuration. The track gauge was 1430–1437 mm on straight track. A huge amount of data has been collected and the evaluation work of the results is highly time-consuming. Therefore, within the limits of this simulation study, a selection of results has been chosen in order to validate the simulation model.

In the present study, comparisons between test and simulation results have been made in small-radius curves (curve radii of approximately 300, 600 and 1000 metres). Several test results for each curve radius (evenly distributed between right- and left-hand curves) have been selected in order to illustrate the spread. Track shift forces, ΣY_{2m}, for tests as well as for simulations have been evaluated and are presented in Figure 4-6. In small-radius curves the track shift forces are the largest on axle 1 (shown here), in contrast to larger track shift forces on axle 2 or 4 on straight track and in large-radius curves. The limit value according to Equation (2-1) is in this case $\Sigma Y_{2m} < 60$ kN. The simulations are performed on *comfort track*, which should be representative of the real test conditions. Notable is that test results for the medium bogie configuration in the smallest curve, R = 300 m, are missing, due to instrumentation problems. The corresponding simulation results are, however, included in the comparison diagram. The simulation results for the track shift forces seem to agree remarkably well with the test results, although the track geometrical irregularities are not exactly the same.

Furthermore, the quasi-static lateral guiding forces on wheels, Y_{qst}, from the same test and simulation cases are plotted in Figure 4-7. It can be observed that the simulated results are consistently somewhat lower than the test results, although in the right order. The differences depend most likely on difficulties in imitating real weather conditions, and hence the friction coefficient in the contact point. Also, in the smallest curve radii ($R \approx 300$ m) the test track was usually lubricated at the high rail, which is not the case in simulations. Further tests with various track lubrication are planned for summer 2007 to resolve these issues.

Furthermore, if a more accurate agreement is desired the model should be provided with measured wheel and rail profiles (original unworn wheel and rail profiles, S1002 and UIC60, have been used for simulations, since it turned out that there was no significant difference to simulations with standard worn profiles). The virtual track conditions with appropriate track irregularities could also be included in the model.



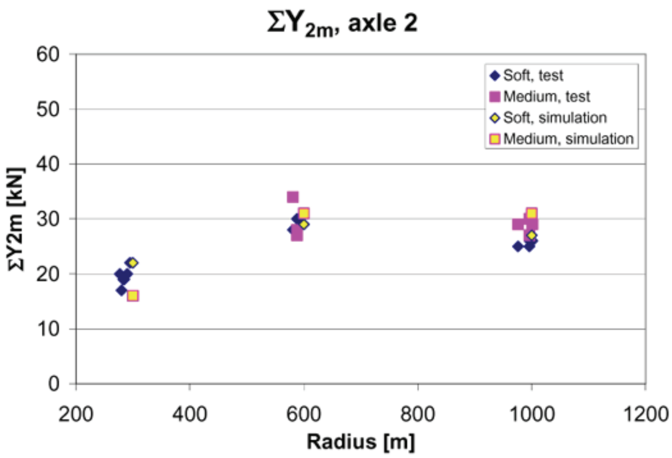
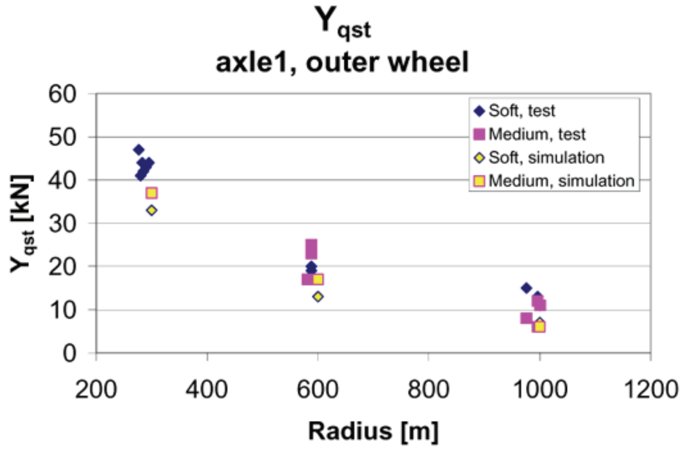


Figure 4-6 Track shift forces (ΣY_{2m}, 99.85-percentiles) of the leading bogie (axle 1 and 2) in small-radius curves. *Test and simulation results.*



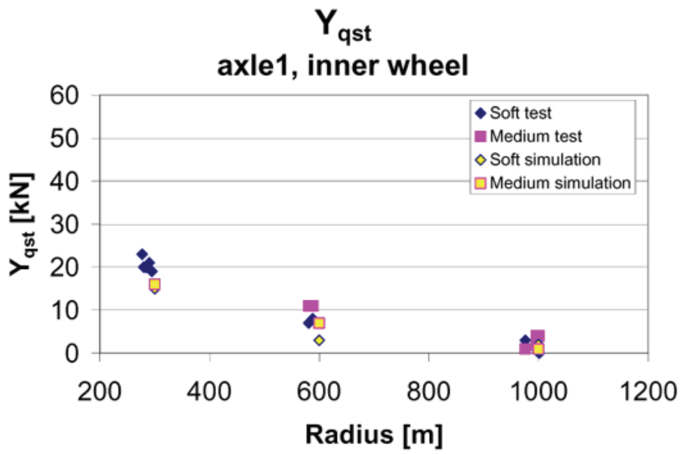


Figure 4-7 Quasi-static lateral guiding forces (Y_{qst}) of the leading wheelset (outer and inner wheel) in small-radius curves. *Test and simulation results.*

Comparisons between test and simulation results on straight track are not meaningful in this sense, since the maximum of the track shift forces is highly dependent on the actual track irregularities. If the virtual track irregularities were included in the simulations they could have been more exactly comparable. However, according to the test report [17] from Interfleet Technology the general behaviour is principally the same in tests and simulations. Both the soft and medium bogie configurations showed similar results regarding high-speed stability, with slightly higher lateral track forces for the medium bogie configuration.

Also the low-frequency periodic motions, observed in the simulations in large-radius curves at high cant deficiency, turned up in the tests. These motions caused considerable carbody accelerations, close to the limits set by UIC 518 [17].

The overall conclusion is that the simulation model and the simulation methodology used have the ability to predict real dynamic behaviour both qualitatively and quantitatively.

5 Conclusions and future work

Conclusions

An MBS vehicle model has been built up in order to perform simulations of running behaviour at higher speeds. The intension is to find a bogie configuration that guarantees stability on straight track and simultaneously acceptably low lateral track forces and wheel and rail wear in curves. The requirements established by international and national standards regarding safety, track fatigue and running behaviour must be fulfilled.

Simulations have been performed on track geometry with varying curve radii, track cant and cant deficiency. Ten different cases of equivalent conicity have been defined and used in order to evaluate stability, track forces and ride comfort on straight track; see Table 3-8.

Up to a certain level of wheelset guidance stiffness a more stable running behaviour at higher speeds can be achieved. However, higher stiffness parameters cause higher track forces in small-radius curves. The difference between the soft and medium bogie configurations is, however, quite modest according to running behaviour at higher speeds, but the more obvious when it comes to small-radius curves.

Further, the dissipated energy in the contact area between wheel and rail is more extensive in small-radius curves, in particular with a stiffer wheelset guidance. This leads to a risk of a high amount of wear on wheels and rails as well as rolling contact fatigue (RCF). A vehicle, which generates acceptably low track forces and hence a moderate amount of wear is considered to be track-friendly.

Parameter studies have been performed in order to assess the influence of equivalent conicity on the vehicle's running behaviour. Generally, higher values of equivalent conicity cause a worse case, concerning track forces and accelerations. It could also be observed that simulations performed with equivalent conicity Type 2 generally cause higher track forces and accelerations compared with Type 1.

It turned out that merely increasing the wheelset guidance stiffness was not enough to secure running stability at higher speeds. The influence of the yaw dampers was examined and it could be stated that the running behaviour could be stabilized if parameters of the yaw dampers were adjusted.

Any significant differences in lateral ride comfort could not be detected between the two bogie types ("soft" and "medium") on straight track or in large-radius curves, whereas the medium bogie configuration shows slightly lower ride index values in small-radius curves.

An interesting observation is the tendency of low-frequency periodic motions in large-radius curves, which considerably affects the ride comfort, more than the track forces and their safety criteria. Both bogie configurations show the same motion behaviour in the frequency range 1.5–2 Hz.

Although the GreenTrain testing has not been fully evaluated, there are strong indications that the principles of radial self-steering bogies (i.e. bogie with a quite soft wheelset guidance) can be extended into the speed range of at least 250 km/h. The lateral track forces are low, typically 50–65 % of the limit values. Simulations of wheel and rail wear (e.g. the so-called wear number) show significant advantages for the "softer" wheelset guidance. The difference between the soft and medium bogie configurations is quite small when it comes to running behaviour at higher speeds, but more obvious when it comes to small- and medium-radius curve negotiation. In particular the soft bogie configuration has the potential to run at further increased lateral accelerations (cant deficiency) which could be important when used for tilting trains.

Furthermore, comparison between simulations and on-track testing shows good agreement, as far as the tests have yet been evaluated.

Future work

The vehicle model used in the simulation study comprises a rigid carbody. However, in future research a vehicle model with a flexible carbody should be evaluated and validated more thoroughly in order to receive even more accurate simulation results.

High-speed tests have been performed on tracks in Sweden during the summer 2006. The measurements should be evaluated more thoroughly in order to be able to draw conclusions and new directions for the project in general.

By means of measurements from the high-speed tests the vehicle model in SIMPACK used for the simulation study can be validated more accurately.

So far, the phenomenon low-frequency periodic motions has not been thoroughly investigated and deserves thus special attention in further research work.

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Appendix A - Notations

Latin symbols

a^{wrms}	*rms*-value of frequency-weighted acceleration (m/s^2)
a_{Xj}^{Wi}	*rms*-value of longitudinal frequency-weighted acceleration (m/s^2)
a_{Yj}^{Wi}	*rms*-value of lateral frequency-weighted acceleration (m/s^2)
a_{Zj}^{Wi}	*rms*-value of vertical frequency-weighted acceleration (m/s^2)
b_0	half distance for definition of track cant (m)
D	cant (m)
$\bar{E}$	energy dissipation (Nm/m)
F_η	lateral creep force (N)
F_ξ	longitudinal creep force (N)
f_0	instability frequency (Hz)
g	gravitational acceleration (m/s^2)
I	cant deficiency (m)
M_ζ	spin moment (Nm)
P_0	axle-load (N)
Q	vertical contact force (N)
Q_{dyn}	dynamic vertical contact force (N)
Q_{qst}	quasi-static vertical contact force (N)
R	curve radius (m)
Δr	difference in rolling radius (m)
S	track shift force (N)
T	time (s)
t	time (s)
v	vehicle speed (m/s)
v_η	lateral sliding velocity (m/s)
v_ξ	longitudinal sliding velocity (m/s)
$\ddot{x}_{Wi}^{*}$	longitudinal weighted acceleration (m/s^2)

Y	lateral contact force (N)
Y_{qst}	quasi-static lateral contact force (N)
Y/Q	ratio between lateral and vertical wheel force (-)
ΣY	sum of guiding forces per wheelset (N)
Δy	lateral displacement (m)
$\ddot{y}^*_{Wi}$	lateral weighted acceleration (m/s^2)
$\ddot{z}^*_{Wi}$	vertical weighted acceleration (m/s^2)

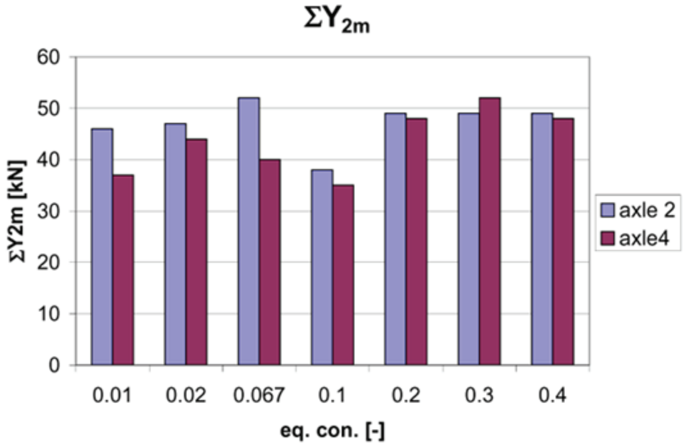
Greek symbols

α	constant (-)
λ_{eq}	equivalent conicity (-)
σ_H	standard deviation of vertical track irregularity (m)
σ_P	standard deviation of lateral track irregularity (m)
ω	rotational sliding velocity (rad/s)

Abbreviations

KTH	Kungliga Tekniska Högskolan (Royal Institute of Technology)
MBS	multi-body system
RCF	rolling contact fatigue
rms	root mean square
UIC	Union Internationale des Chemins de fer
Wz	Wertungszahl (comfort index)

Appendix B - Simulation results



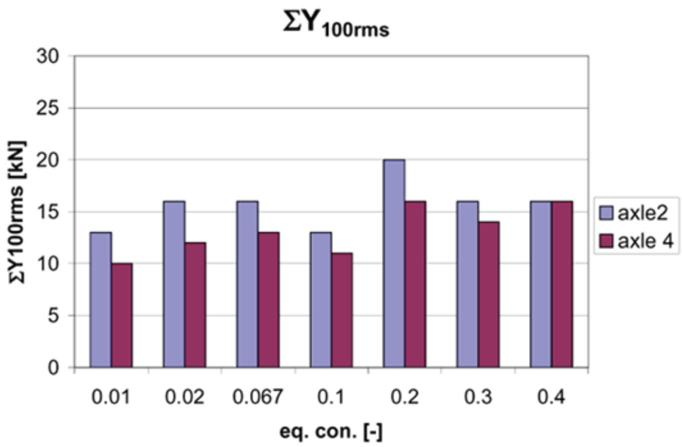


***Figure B-1** **ΣY_{2m} (99.85-percentiles) and ΣY_{100rms} of axle 2 and 4 for equivalent conicity values between 0.01 and 0.4, Type 1.** Simulations with soft bogie configuration on straight safety track at a vehicle speed of 275 km/h.*

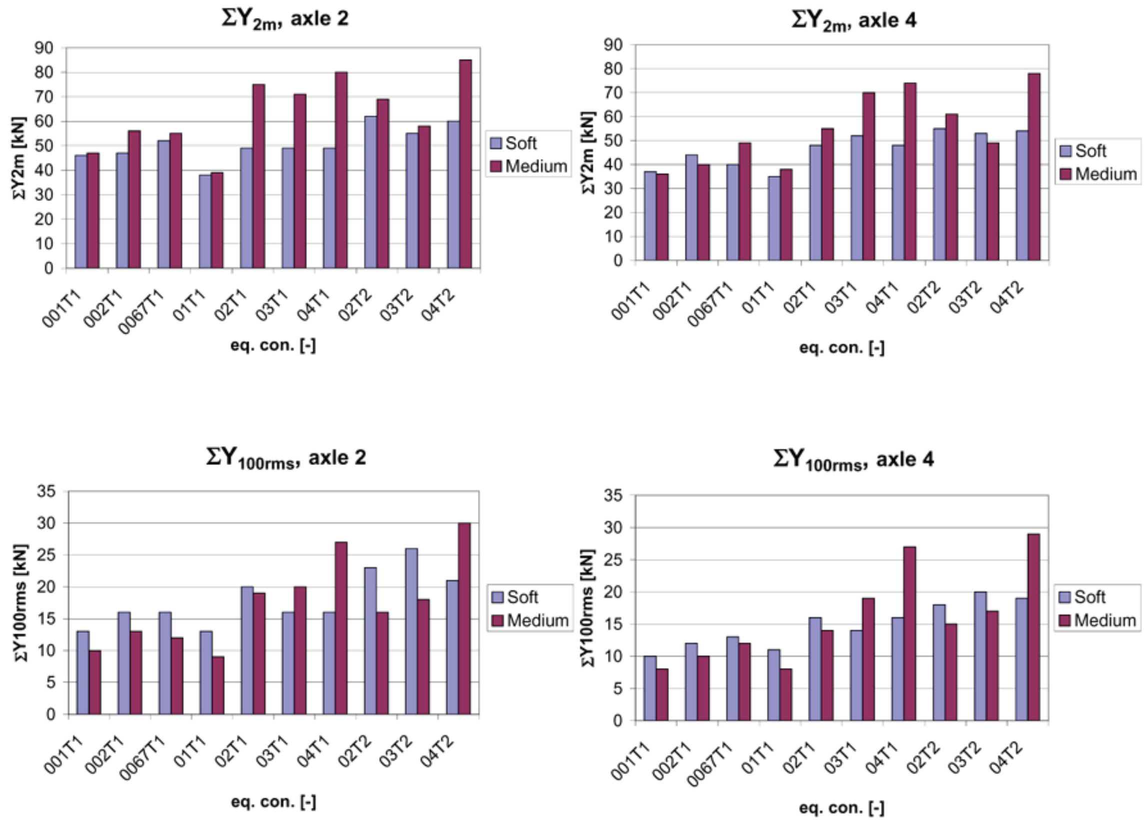


Figure B-2 ***ΣY_{2m} (99.85-percentiles) and ΣY_{100rms} of the soft and medium bogie configurations for different equivalent conicity.*** *Simulations on straight safety track at a vehicle speed of 275 km/h.*

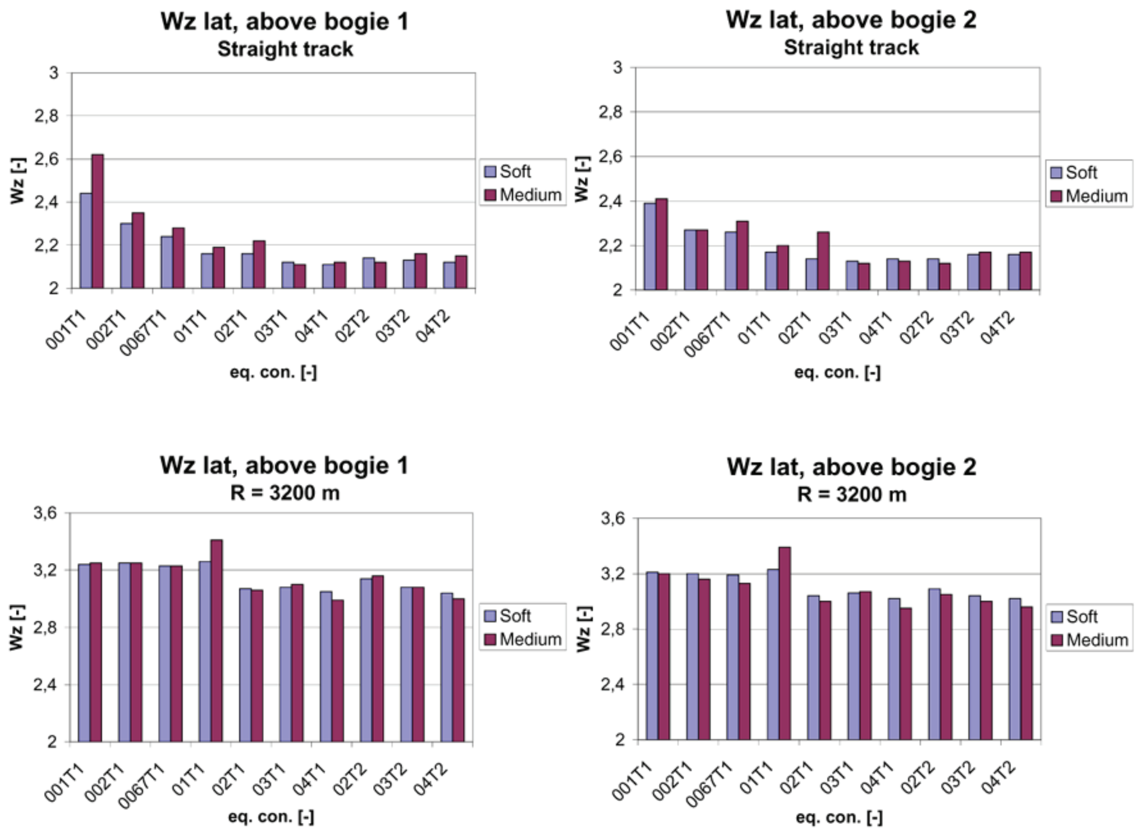


Figure B-3 Lateral Wz on the carbody floor above the two bogies of the soft and medium bogie configurations. *Simulations are performed on straight comfort track and in curve with* R *= 3200 m, at a vehicle speed of 275 km/h. The track plane acceleration is 1.1 m/s*^{2} *in the curve.*

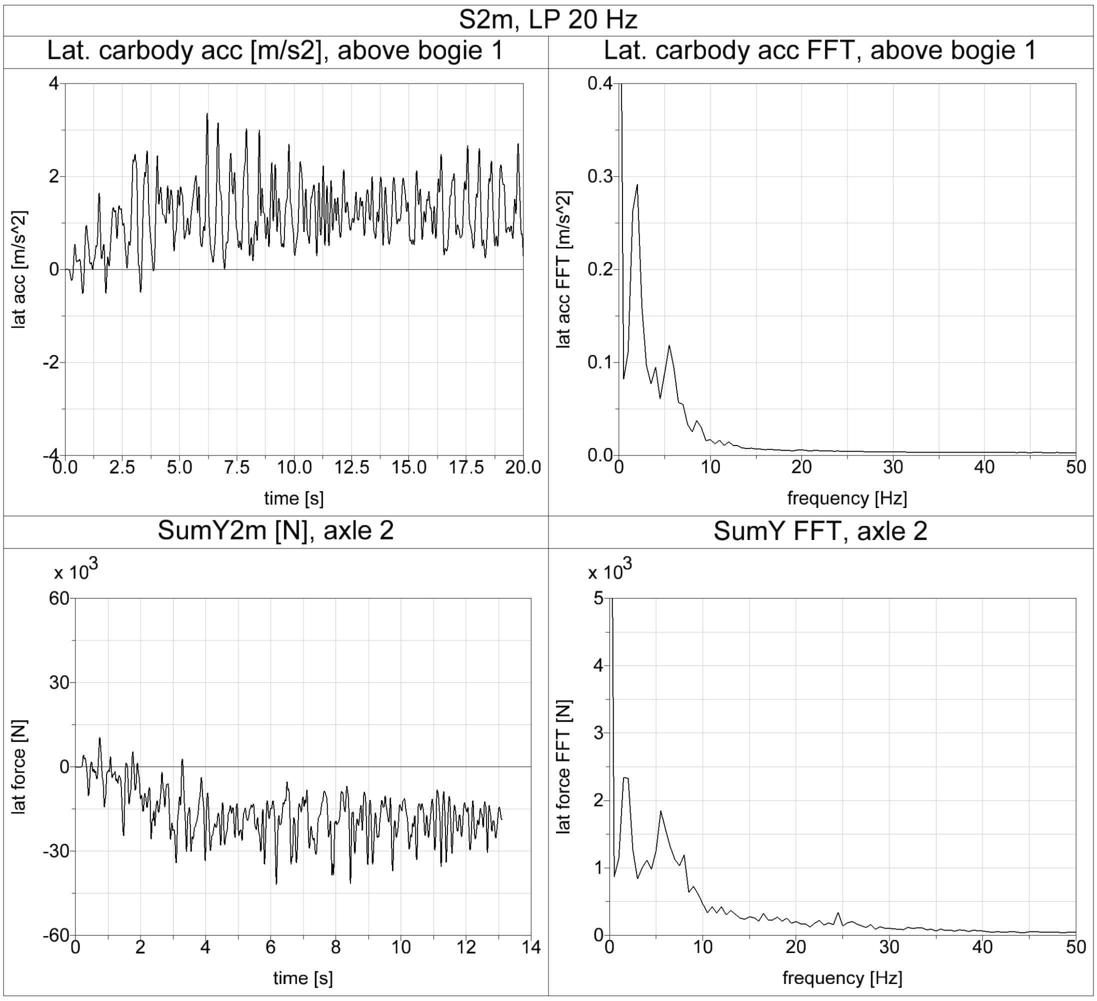


Figure B-4 **Lateral acceleration on carbody floor (above bogie 1) and lateral track shift force ΣY_{2m} (axle 2), with corresponding FFT (Fast Fourier Transform).** *Simulation in curve with radius* R = *3200 m, vehicle speed 275 km/h, track plane acceleration 1.1 m/s*^{2}*, on comfort track and with equivalent conicity 0.067 (Type1).*

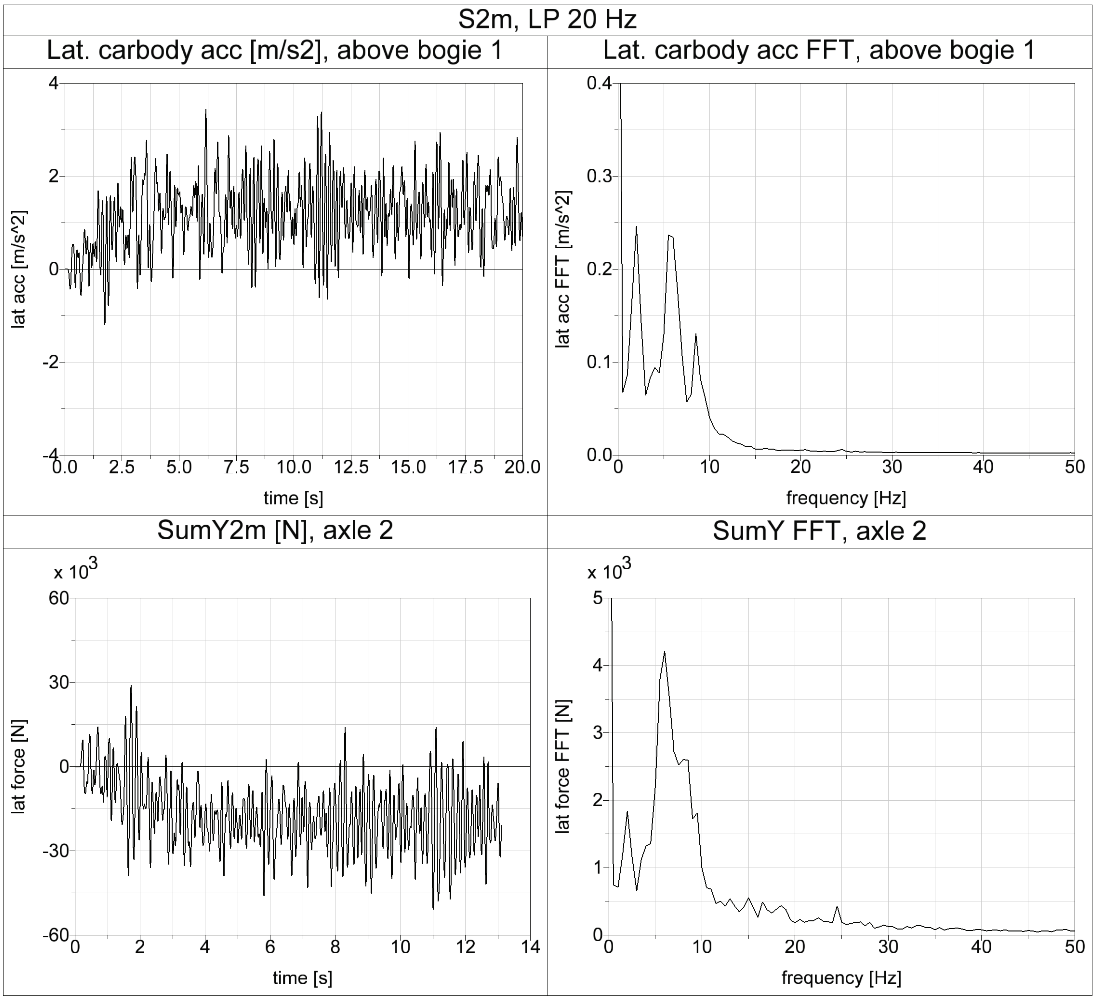
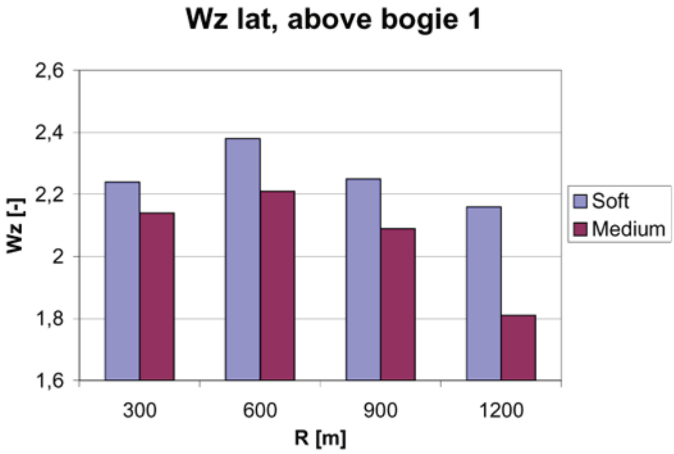


Figure B-5 **Lateral acceleration on carbody floor (above bogie 1) and lateral track shift force ΣY_{2m} (axle 2), with corresponding FFT (Fast Fourier Transform).** *Simulation in curve with radius* R = *3200 m, vehicle speed 275 km/h, track plane acceleration 1.1 m/s^2, on comfort track and with equivalent conicity 0.4 (Type2).*



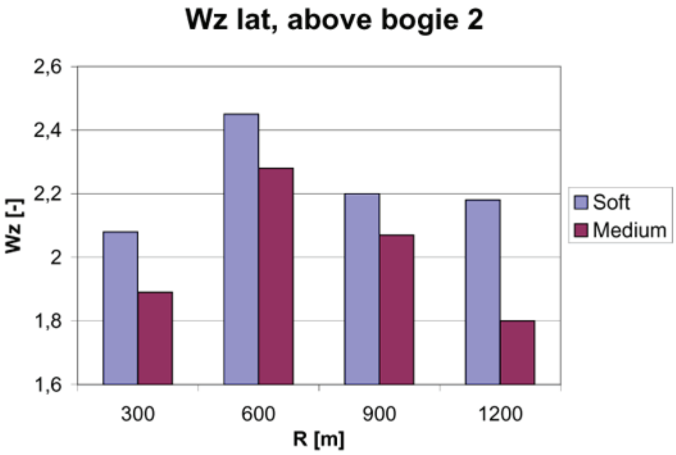


Figure B-6 Lateral Wz on the carbody floor above the two bogies of the soft and medium bogie configurations in small-radius curves. *Simulations performed on comfort track in curves with radii 300, 600, 900 and 1200 metres and track plane acceleration 0.8 m/s*^{2}.