ON THE OPTIMIZATION OF A TRACK-FRIENDLY BOGIE FOR HIGH SPEED

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Abstract

When designing and optimizing a rail vehicle there is a contradiction between, on the one hand, stability on straight track at high speed and, on the other hand, reasonable wheel and rail wear in small- and medium radius curves. Higher speeds require to some extent stiffer wheelset guidance to avoid hunting and ensure stability. However, with stiffer wheelset guidance the risk of increased wheel and rail wear in curves is imminent. In this paper, the process of developing and optimizing a track-friendly bogie is described. A multi-body system (MBS) simulation model was used, taking due consideration to nonlinearities in suspension and wheel-rail contact, as well as realistic flexibilities in the track. Adequate and systematic consideration is taken to a wide range of possible non-linear wheel-rail combinations. Dynamic stability is investigated both on straight track and in wide curves at high speeds. The balance between flange wear and tread wear is studied in order to maximize wheel life between re-profiling operations in the intended average operation. The result is a bogie with relatively soft wheelset guidance allowing passive radial self-steering, which in combination with appropriate yaw damping ensures stability on straight track at higher speeds. The bogie has been subject to both certification testing and long-term service testing in the Gröna Tåget (the Green Train) research and development programme.

1. INTRODUCTION

Track friendliness is an important matter in vehicle-track interaction. Firstly, track friendliness means that the vehicle produces low or moderate forces on the track and/or produces low abrasive wear or rolling contact fatigue on the track. The target is to minimize track deterioration, causing maintenance and renewal with associated cost and interruption of train operations. In the future, the marginal cost for track deterioration should be included in the track access charges on a number of European rail networks. This is necessary for the internalization of the cost caused by the train operator when using the railway infrastructure. Rail vehicle operations causing a high rate of track deterioration should pay more than those causing low or moderate track deterioration. This will sharpen the need for track-friendly vehicles.

Track friendliness in this sense is depending on different properties and features of the vehicles, such as low axle load and unsprung mass, ability of the wheelsets to steer radially in curves, low centre of gravity and others [2]. In addition, operational parameters, such as speed and cant deficiency, affect track deterioration.

Secondly, track friendliness sometimes also means that the vehicle is able to run on *non-perfect track*, i.e. with considerable geometrical irregularities, with favourable response regarding forces on track as well as ride quality. Also, this aspect is important in order to avoid excessive maintenance cost at higher speeds. Track friendliness is a prerequisite for tilting trains, running at extra-ordinarily high speed and cant deficiency.

Track friendliness usually also results in favourable properties regarding wear and deterioration on the vehicles themselves, for example, low *rail wear* will most likely be associated with low *wheel wear*. Independently of the above-mentioned aspects, *safety* requirements must always be met.

Some of these requirements (low forces on track, low wheel and rail wear, smooth ride and safety) may sometimes be contradictory and conflicting. A widespread opinion in Europe is that vehicles intended for high speed need very high stiffness in the wheelset guidance, otherwise there is a risk of unstable hunting oscillations. A high guiding stiffness, however, may produce excessive wear of wheels and rails in small- and medium-radius curves along the lines. In particular, this conflict is obvious if the same vehicles run both on high-speed lines at very high speed and on conventional lines in small- and medium-radius curves. This situation is quite usual on different rail networks in Europe. The intension of this paper is to show how this conflict can be solved, namely the conflict between good curving performance and high-speed performance.

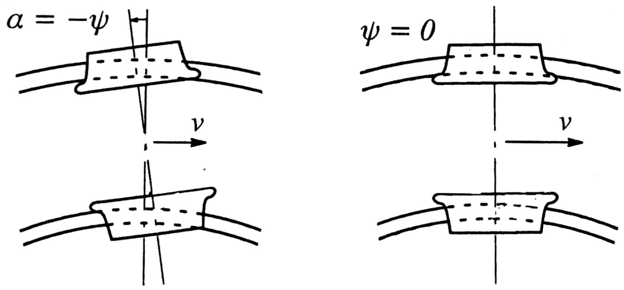
2. RADIAL STEERING RUNNING GEAR

2.1 Wheelset steering ability versus wheel-rail wear

A conventional railway wheelset has two wheels, each having a *conical shape* at the running surface, i.e. the radius increases at the flange side; see Figure 1. If the wheelset is displaced laterally outwards in a curve, the outer wheel will run at a larger rolling radius than the inner wheel. Further, the *two wheels are firmly connected through the axle* of the wheelset, thus the two wheels are always rolling at the same angular speed. A larger rolling radius on the outer wheel will therefore force that wheel to run at a higher linear speed than the inner wheel. This compensates for the longer way the outer wheel has to travel along the outer rail, allowing the wheelset to take up an approximate radial position along the curve [1].

A radial attitude approximately minimizes the *creep* – the degree of sliding – in the contact patches between wheels and rails. Creep is closely related to the friction forces – or *creep forces* – in the same contact patches. Creep and creep forces together produce wear and rolling contact fatigue (*RCF*).

Thus, a free railway wheelset has an *inherent self-steering ability.* Approximate radial steering minimizes creep and creep forces and consequently also abrasive wear and rolling contact fatigue.



v = forward speed
α = -ψ = angle of attack

Figure 1 *A radial attitude (right) of the wheelset in a curve produces normally a low amount of creep, friction forces and wear in the contact patches between wheels and rails. An angle of attack (left) will produce a higher amount of wear.*

However, all wheelsets in a rail vehicle are guided in and connected to a bogie frame or to the main frame of the vehicle. If these connections are stiff, they will prevent the wheelsets to take up radial positions. Cases with "stiff" and "soft" wheelset guidance are shown in the left and right parts of Figure 2, respectively.

In order to allow radial self-steering, the *longitudinal* flexibility is a most important issue. If the radial movement of the wheelset has to overcome a restraining force due to the stiffness, the friction-creep forces must help to steer radially, although with a quite low amount of creep. Thus, in self-steering running gear the radial steering is to some degree dependent on the friction forces. If the friction is low, there is a lower degree of radial steering; however, with a low friction the wear is usually low anyhow.

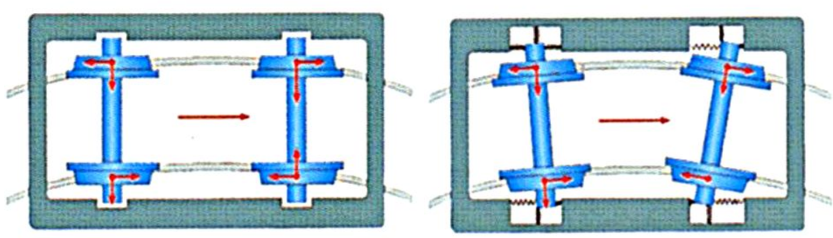


Figure 2 *A stiff wheelset guidance (left) generates a considerable angle of attack between the leading wheels and the rails. More flexible – or soft – wheelset guidance (right) allows the wheelsets to take up approximate radial positions in curves.*

In this paper running gears for fast passenger vehicles are the subjects, i.e. motor coaches and passenger cars for axle loads above 10 tons and permissible speeds of at least 140 km/h. These vehicles have usually two-axle bogies with axle distances of 2.5–3.0 m and axle journal bearings outside the wheels. We will use the terms '*soft*' and '*stiff*' to characterize the guiding stiffness of the wheelsets, in particular the longitudinal stiffness on each side between wheelset and bogie frame, Figure 3.

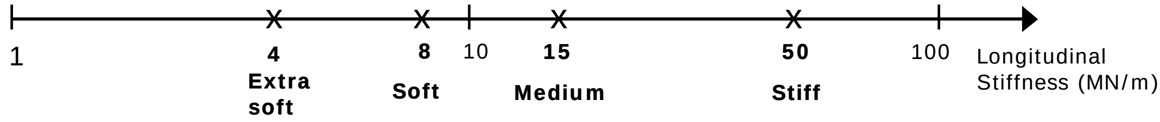


Figure 3 Longitudinal stiffness for a vehicle having about 15 tons of axle load.

The advantages of flexible or “soft” wheelset guidance has been known and utilized for a long time. Very common examples are the flexible wheelset guidance being used in most European freight wagons, two-axle wagons as well as bogie wagons. Freight wagons with flexible wheelset guidance are known to have just a modest wear of the wheel flanges. Thus, also the wear of rails – caused by these freight wagons – is expected to be low.

There are special cases where the self-steering ability is limited, for example on very narrow curves or with high traction forces acting on the wheels. In such cases some steering linkage, or even active mechatronic means, may help to steer the wheelsets radially; see for example [5, 6]. However, in normal railway operations with few very tight curves (radius $R < 300$ m) and moderate traction forces, the simpler self-steering solution may be sufficient.

2.2 Wear models

A decisive measure for wear and RCF – although simplified – is the *energy dissipation* in the contact patch. In this context, the energy dissipation is defined as the energy loss per metre of travelled distance. Somewhat simplified, neglecting the rotational spin creep and its resulting spin moment, the energy dissipation E is equal to the product of creep force vector $\boldsymbol{F}$ and creep vector $\boldsymbol{\upsilon}$ in the contact patch:

$$E = \boldsymbol{F} \cdot \boldsymbol{\upsilon} \qquad \text{(Nm/m)} \qquad (1)$$

Sometimes the simplified energy dissipation according to Equation (1) is called *wear number.* A simple model, verified through comparison with measured field data on rail surface damage, is presented in [4]. This model defines areas of rolling contact fatigue between wear numbers of 15–75 Nm/m (with a maximum at 60 Nm/m) and a progressively increasing wear rate from 60 Nm/m and up. The total damage is a progressively increasing function of the wear number from 15 Nm/m and up. Models that are more comprehensive also consider the contact pressure; see for example [7, 8]. The latter models are verified against wheel wear data.

2.3 The hunting challenge

Freight wagon running gear is known to have quite bad ride qualities, i.e. they are sometimes shaking considerably while running over the track. Partly, this is due to the flexible wheelset guidance combined with lack of adequate damping, allowing the wheelsets to perform lateral, approximately *sinusoidal, oscillations* when they are rolling along the track. The upper parts of the vehicle, i.e. bogies and the carbody, interact with these oscillations and sometimes the wheelset motions are further amplified by this mutual interaction.

The lateral oscillations may be damped or undamped. Above a certain speed, the oscillations usually tend to be sustained and sometimes violent, i.e. there is a so-called limit-cycle oscillation, limited by the wheel flanges hitting the rails. This phenomenon is usually called *instability* or *hunting*, which is a potential problem for most rail vehicles having normal wheelsets with conical wheels, not only for freight wagons. Such hunting motions would produce both a bad ride and possibly excessive and unsafe lateral forces on the track.

Generally, the natural wheelset oscillations need to be restricted by the bogie frame (or the vehicle’s main frame) and/or by appropriate damping means. A certain amount of stiffness in the wheelset guidance is necessary to avoid hunting; the needs for longitudinal and/or lateral stiffness normally increase at higher speeds. Another very efficient mean to avoid hunting is to arrange so-called *yaw damping* of the bogies in relation to the carbody, i.e. longitudinally directed dampers placed between each side of the bogie frame and the carbody; see Figure 4. Such a yaw damping stabilizes the yaw motion of the bogie (i.e. the rotation around a vertical axis) as well as the lateral oscillations. Sometimes also, dampers between wheelsets and the bogie frame may help to stabilize.

A widespread opinion in Europe is that bogies intended for high speed need a very high guiding stiffness for the wheelset connections; otherwise there is a risk of unstable hunting oscillations. However, above a certain amount of guiding stiffness the improvements (in terms of stability) of having an additional guiding stiffness are just marginal or non-existing. Further, a lower guiding stiffness can often be compensated by a higher amount of damping – in particular yaw damping – from a stability point of view. It should be noted that also “stiff” bogies usually need yaw damping to avoid hunting, at least at speeds around 200 km/h and above.

Finally, a vehicle with its running gear should not simply be designed for a maximum margin against hunting; other requirements such as track forces, wear and ride quality should primarily be considered in the optimization process. It is rather an issue of designing for a *sufficient stability margin* while maximizing the other benefits.

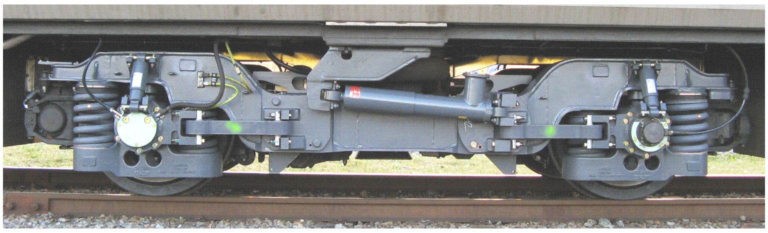


Figure 4 Yaw dampers placed longitudinally between bogie frames and carbody, attenuating the yaw motions of the bogie, are efficient means of stabilizing bogies at higher speeds.

3. TRACK-FRIENDLY TRAIN DEVELOPMENT FOR SCANDINAVIA

In the 1970's the Swedish railways suffered from an epidemic rate of wheel and rail wear, resulting from the introduction of more "dry" vehicles, i.e. with roller bearings instead of earlier "wet" plain bearings. This is the reason why radial steering bogies were required and introduced on passenger cars and multiple units in the 1980's. Evaluations showed most favourable effects of this introduction [11].

Further, the Swedish State Railways (SJ) made a specification and initiated a comprehensive development programme for future high-speed tilting trains, running at enhanced speeds on conventional curvy track. To meet requirements on track forces as well as wheel and rail wear, radial steering bogies were developed. This programme resulted in the tilting train *X 2000*, which entered service successively from 1990. Since 1998, 43 units are in service on most Swedish main lines, Figure 5. Each train has a power unit and 4–6 tilting cars.

A further development is the high-speed trains *BM71* and *BM73* for Norway. *BM73* is a tilting train, Figure 6. Another development is *OTU* (the Öresund Train Unit), linking Denmark and Sweden over the new bridge. Finally, the wide-body *REGINA* trains are used in a number of services in Sweden. All these trains have maximum speeds reaching from 180 to 210 km/h. They are electrical multiple units (EMU) with distributed power. In total about 1300 four-axle passenger vehicles are in service (2009) and further more than 100 are on order. All these vehicles are equipped with radial self-steering bogies.

Figure 5 X 2000 tilting high-speed train for Sweden

Figure 6 BM73 tilting high-speed train for Norway

Gröna Tåget – the Green Train programme

Banverket (Swedish Rail Administration) initiated the research and development programme Gröna Tåget (the Green Train) in 2005. The overall aim with the programme is to develop a train concept and technology for the next generation of high-speed trains for Nordic conditions, in particular for Sweden. The train should be suitable for operation at speeds of up to 250 km/h on conventional lines and around 300 km/h on new dedicated high-speed lines. To make this possible, co-operation within the railway industry is needed, in order to improve vehicle dynamics, carbody tilt, energy consumption, winter climate reliability, aerodynamics and acoustics, among other things. Also market needs, capacity and economics, as well as passenger issues are addressed.

One of the subjects is the development of track-friendly running gear, being capable to run at very high speeds on suitable track, while running with moderate track forces and wear on conventional curvy track with enhanced speed (i.e. high cant deficiency) in curves. Ride qualities should be good also on non-perfect track subject to damage from heavy freight trains and frost.

There are several parts in the development of track-friendly bogies for high or very high speed:

(1) Radial self-steering *(RSS)*
(2) Active radial steering (*ARS*)
(3) Active lateral secondary suspension (*ALS*)

Active radial steering (*ARS*) is described in [6, 9], and active lateral suspension (*ALS*) in [10]. In the present paper the optimization of radial self-steering bogies (*RSS*) is described. General presentations of the *RSS* technology and experience can be found in [11].

Track-friendly bogies for high or very high speed are developed in close co-operation between Bombardier Transportation and KTH. The main part of theoretical analysis and dynamic simulations on *RSS* and *ALS* is made by KTH, while Bombardier have designed and built the hardware. The newly developed bogies have been subject to certification testing according to UIC 518 as well as long-term service testing. A modified *BOMBARDIER REGINA* train is used as test bench for new technologies, Figure 7. Tests have been made in all years 2006–2009, on different tracks and curve radii, at different speeds. In September 2008, a new speed record of 303 km/h was set on an ordinary Swedish track otherwise used for 160–200 km/h. Track geometric quality was close to the limit of what is accepted for the lower speeds.

Figure 7 A modified REGINA train is used for testing of track-friendly bogies and other systems.

4. TARGETS ON TRACK FRIENDLINESS, STABILITY AND RIDE COMFORT

The Swedish rail network is quite curvy. About 8 % of passenger operations (in ton-km) are carried out in the curves radius interval 551–900 m (with a concentration around 600 m), while 2.4 % is carried out in curves with smaller radius [3]. Most curves with radius less than 550 m are lubricated, larger radius are usually not. Considerable wheel and rail wear may be produced if wheels or rails are not lubricated. Rail surface damage is estimated to account for 40 % of total maintenance and renewal cost for tracks on the Swedish rail network [3].

For Gröna Tåget (the Green Train) we have firstly considered two important aspects of track friendliness, according to the developed model [2, 3], namely modest vertical wheel-rail forces and a modest amount of rail surface damage (abrasive wear and rolling contact fatigue):

1. Target axle load is 150 kN (15.3 ton) with average load factor in the trains. With a normal unsprung mass of 800–900 kg per wheel, this should produce dynamic vertical forces in the order of 100–110 kN (with 20 Hz low-pass filtering) which is 60–65 % of the allowed limit value according to UIC 518 [12].
2. Target wear number on wheel-rail patch on outer rail is 100 Nm/m in curve radius R = 600 m and dry friction (μ = 0.5) as average with common wheel-rail combinations. This is to avoid excessive wear on wheels and rails in common curves, but wear is expected to be large enough to produce some wear on wheel flanges to avoid the development of hollow wheels.

Besides axle load and wheel-rail wear other aspects are important as well:

3. Dynamic stability of the bogie (without hunting) shall be achieved at an equivalent conicity in the whole range of 0.01–0.40, at testing speeds up to 275 km/h. This is to be evaluated by *rms* of lateral wheelset force ΣY sliding mean over 100 m, designated ΣY_{100rms}, according to UIC 518 [12]. However, to allow for a wider range of wheel and rail geometry conditions, the range of equivalent conicity is wider than anticipated in UIC 518.
4. Good ride in the carbody, as expressed by the Ride Index *Wz* or the frequency-weighted *rms* acceleration according to EN 12299. No fixed targets are set, but *Wz* in the range of 2.2 – 2.4 should be aimed for, at the highest operational speeds with track irregularities on the Swedish track maintenance limit [17] related mainly to “comfort”. Particular attention should be paid to low-frequency lateral motions (1–3 Hz) when running at high cant deficiency in large-radius curves, dimensioned for top speeds of about 250 km/h. For a non-tilting version of the train, an admissible cant deficiency of 150–180 mm (lateral quasi-static acceleration ca 1.0–1.2 m/s^2) is aimed for.

Finally, an additional target was set up, having in mind that future versions should be suitable to run at very high cant deficiency (275–300 mm) in future high-performance tilting trains:

5. Lateral ΣY_{2m} wheelset forces (also called track-shift forces) should be limited to 60–65 % of limit values according to UIC 518 at 165 mm of cant deficiency. At an axle load of 150 kN the limit value is 60 kN.

5. METHODOLOGY

5.1 Optimization process

The optimization process is an iteration procedure, where firstly the major targets – low wheel-rail forces and low wheel-rail wear – have to be achieved. Then stability and ride quality must be assured to be satisfactory.

(1) The first step is to check that the axle load target is realistic for the train in question; this is based on estimations of the industrial partner in the Gröna Tåget programme.

(2) The second step is to investigate, by computer simulation, the necessary requirements on wheelset steering ability – in particular the longitudinal guiding stiffness between bogie frame and axle journals. This should be made by using realistic friction levels and wheel-rail combinations.

(3) The third step is to investigate, by simulation, the degree of stability at different parameter settings. In particular, the trade-off between self-steering ability in curves (the wear number being the characteristic quantity) and stability is determined. Different measures to achieve stability at high levels of steering ability are tested.

(4) Step number four is to investigate the possibilities of achieving a good ride and lowest possible lateral ΣY forces without compromising track friendliness and stability.

5.2 Simulation software

Any verified and validated multi-body systems (MBS) simulations software can be used, for example *VAMPIRE*, *GENSYS* or *SIMPACK*. In this case *SIMPACK* is used [14].

In this case, all bodies are assumed to be stiff and interconnected by linear or non-linear spring and damper elements. Simulation is performed by step-wise time integration of the differential equations generated by the software. The wheel-rail contact forces and conditions are calculated according to Hertz' contact theory (with elliptic contact patches) and Kalker's simplified non-linear theory using the FASTSIM algorithm [15].

5.3 Vehicle model and data

An existing *SIMPACK* model of a *REGINA* train was used as a starting point. This model was changed according to modifications of the test train. Only one car is simulated, which is a simplification and approximation as the two cars in the test train have couplers with lateral dampers. This is judged to have negligible effects on wear, track forces and stability, but may have some effect on the ride quality in carbodies.

The carbody is in this case assumed to be rigid, which is also an approximation judged to be satisfactory in this specific case, although the ride could be affected. Two motorized two-axle bogies are connected to the carbody via a drawbar with flexible elements, non-linear secondary air springs, and lateral non-linear bumpstops as well as lateral and vertical non-linear viscous dampers. In addition, there is one non-linear viscous yaw damper at each side of the bogie as well as an anti-roll bar with linear stiffness. Each damper has a linear spring in series with the viscous element. Each yaw damper has a so-called blow-off level, which means that there is a specific damping rate up to a certain force level (the "blow-off" level), above which the damping rate is considerably decreased.

The primary suspension and wheelset guidance consists of non-linear springs with some internal damping. At each journal box, there is also one non-linear damper with series stiffness, connected to the bogie frame.

Some important main data are:

- Carbody mass: 44.6 tons
- Bogie mass: 8.6 and 7.9 tons (with and without magnetic rail brakes, respectively)
- Longitudinal guiding stiffness per journal box is varied in the range 4–50 MN/m

5.4 Dynamic model of the track

Earlier experience has shown that the track model and the track stiffness may influence both dynamic track forces and dynamic stability of the bogie. Therefore, a simple track model is designed and used. This model is built up by discrete masses connected to ground and wheelset by linear springs and viscous dampers. The flexible connections are intended to approximate different flexibilities, i.e. flexibility sleeper-to-ground and rails in relation to sleepers, wheelset flexibility and contact stiffness between wheels and rails. This is a quite rough and simple model of the real matters, but is nevertheless an improvement compared with a completely rigid track. The model is shown in more detail in Appendix A1.

Stiffnesses in the track model are tuned at the initial stage, in order to generate realistic lateral stability of bogies as well as realistic impact forces that coincide with previous test results from the original *REGINA* train.

5.5 Equivalent conicity

A most critical issue is the equivalent conicity λ_{eq} in the wheel-rail interface, influencing in particular the dynamic stability. Stability studies should be made under a thorough variation of this parameter.

In EN 15302 [13] a methodology for determining equivalent conicity is defined. The evaluated conicity is dependent on the actual lateral displacement Δy between wheelset and track with typically non-linear characteristics; see Figure 8. In order to simplify the communication of a numerical value of equivalent conicity usually the value for $\Delta y = 3$ mm is used. This is also the recommendation in UIC 518. However, the real characteristic may vary considerably and so the performance of the vehicle. In the present work, therefore two types of equivalent conicity are created and used, called Type 1 and Type 2. Type 2 is distinguished by high values of conicity at small displacements Δy. This latter type usually emerges at larger conicity values.

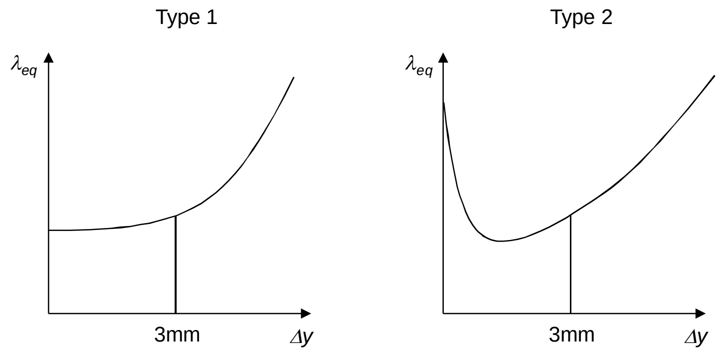


Figure 8 Two types of equivalent conicity, having the same value at 3 mm wheel-rail displacement, but with significant differences at other displacements.

It is essential that real wheel-rail combinations are used, not only with respect to equivalent conicity, but also with respect to other quantities characterizing the wheel-rail contact. In Appendix A2, a list of actual wheel-rail combinations is presented, producing equivalent conicities λ_{eq} of 0.01–0.40 for Type 1 and 0.20–0.40 for Type 2. The choice of wheel-rail combinations is arbitrary, except requirements on equivalent conicity. This methodology, including a systematic variation of equivalent conicity, is a further development of the principles earlier outlined in [16].

5.6 Speed and track geometry

Simulations regarding track forces and wheel-rail wear are made on straight track and in curves with different radii, ranging from R = 300 m up to 2400 m. Dynamic stability is made on straight track and on large-radius curves ($R \geq 2400$ m). In curves the cant deficiency I is varied from zero up to 183 mm, equivalent to a lateral track-plane acceleration of up to 1.2 m/s^2. Speeds are varied accordingly from 83 km/h up to 275 km/h, in some cases up to 330 km/h.

To achieve a realistic evaluation of vehicle-track interaction the built (ideal) track geometry should be superposed by geometrical irregularities. A thorough classification of track irregularities, including their limit values, has been performed; see further [19]. Two types of track are defined:

(1) *Safety track*, where limit values are set according to track quality QN3 in UIC 518 [12].

(2) *Comfort track*, according to the Swedish track maintenance standard [17] related to ride quality.

Track forces and dynamic stability is mainly simulated on *safety track*, while ride quality is preferably simulated on *comfort track*. The magnitude of track irregularities, including limit values, is dependent on intended speed. The simulated tracks are designed from a real track as a starting point. Firstly, irregularities are band-pass filtered within the wavelength interval 3–25 m, according to the UIC standard [12]. Thereafter, irregularities are scaled to suit limit values in the above-mentioned standards [12, 17] for the different speeds.

In the investigations of dynamic stability, it is important that the wheel-rail lateral deviation is large enough to excite a possible instability in the non-linear dynamic system. Of this reason, an extremely large lateral sinusoidal irregularity is included in the simulation track, having an amplitude of ±9.6 mm and a wavelength of 20 m. This track perturbation has 22 % larger amplitude than the QN3 limit for speeds above 200 km/h.

6. RESULTS FROM SIMULATION AND TESTING

6.1 Wear and wear numbers

Wear and rolling contact fatigue is a major consideration in track-friendliness. Figure 9 shows simulation results for different stiffnesses of the longitudinal wheelset guidance. The example shown is for nominal wheel and rail profiles, although also other cases are investigated. For definition of stiffness, see Section 2.1.

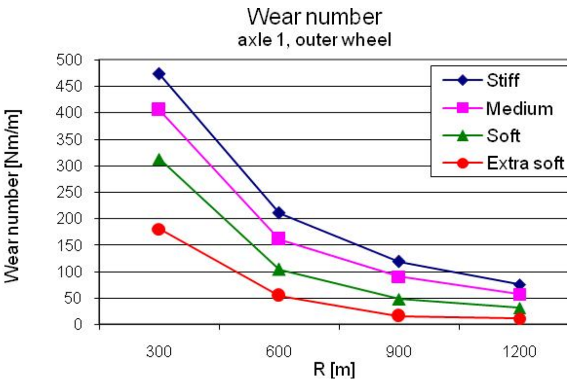


Figure 9 Wear number of the leading outer wheel for bogies with different wheelset guidance. Simulations are performed on dry track with wheel and rails S1002 + UIC 60. Cant deficiency 122 mm (= 0.8 m/s²).

It is concluded that the wear target (Section 4) is met with "soft" wheelset guidance, for common curve radii on Swedish lines. A certain amount of wear on wheel flanges is expected, balancing the tread wear of the wheels, thus avoiding development of hollow wheels, which would cause fast growth of equivalent conicity. For very small curve radii, wear on wheel flanges and rails would occur on dry track. However, at curve radii less than 550 m rails are usually lubricated during the "dry" seasons, thus avoiding excessive wear anyhow.

6.2 Achieving running stability

Dynamic stability is a necessary requirement for rail vehicles at higher speeds. A number of parameter variations are made in order to systematically explore the possibilities to achieve stability with a "soft" setting of the wheelset guidance. A lot of wheelset guidance alternatives were investigated, although longitudinally stiffer alternatives than "soft" did not improve stability margins if other parameters were given optimum values. One of the most decisive parameters is the "blow-off" force level in yaw dampers. Figure 10 shows an example result from simulations. It is concluded that the blow-off force level should be high in this case.

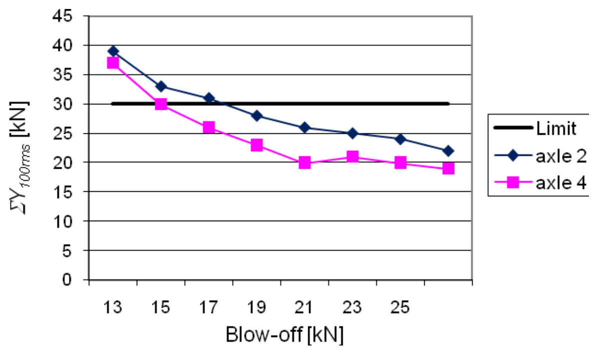


Figure 10 Maximum lateral wheelset forces (ΣY_{100rms}) with varying blow-off forces in yaw dampers. Simulations are performed at 275 km/h with equivalent conicity 0.3, Type 2, on safety track including a large lateral track perturbation.

With optimum suspension parameters, running stability can be achieved with "soft" setting of the wheelset guidance. Figure 11 presents a simulation example where *rms* wheelset lateral forces are shown. The presented case – equivalent conicity 0.3, Type 2 – is found to be the worst case from a stability point of view. It is concluded that the modified bogies have considerable stability margins at 275 km/h, while the "original *REGINA*" bogies have not (the latter are designed for an admissible speed of 200 km/h).

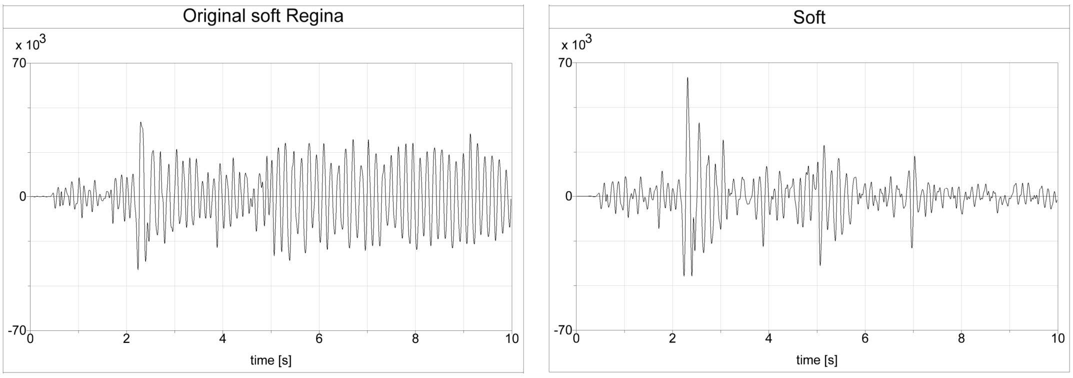


Figure 11 *Simulation of wheelset forces (ΣY sliding mean over 2 m) on axle 4 for the original "soft Regina" bogie (intended for max 200 km/h) and the modified "soft" bogie. Simulations on straight track at 275 km/h, with a large lateral track perturbation at 2.2 s. Equivalent conicity 0.3, Type 2.*

Systematic variations of wheel-rail combinations

In the optimization process, it is necessary to consider and investigate the whole range of possible operational conditions, including track geometry, speed, cant deficiency and equivalent conicity. The latter may exhibit large variations as shown in Section 5.5 and Appendix A2, caused by normal variations in wheel and rail geometry, track gauge etc. In the UIC 518 standard [12] the upper limit value for equivalent conicity is 0.3 in acceptance testing for an admissible speed of 250 km/h. No lower limit of conicity is defined in this standard, although the lower conicity values may cause dynamic sustained motions at low frequencies. In the present work, the low limit of conicity is set to 0.01 and the upper limit to 0.4, with Type 1 and Type 2. Most variations of suspension parameters in the optimization process are investigated with a full or limited set of conicity variations.

Figure 12 and 13 show simulated *rms* wheelset lateral forces ΣY_{100rms} with a complete set of conicity variations. In Figure 12 the difference between the two trailing axles of each bogie is shown, while Figure 13 shows that ΣY is usually largest for conicity Type 2. It should be noted that the levels of ΣY_{100rms} are partly due the very large track perturbation, used for the excitation of bogie instability. However, the optimized bogies run stably; c.f. Figure 11 (right).

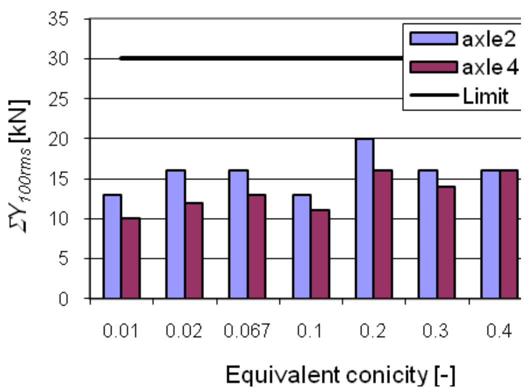


Figure 12 *Simulation results (ΣY_{100rms}) of axle 2 and 4 for equivalent conicity between 0.01 and 0.4, Type 1.*

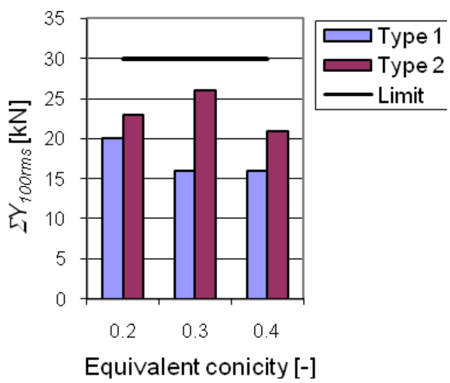


Figure 13 *Simulation results (ΣY_{100rms}) of axle 2 for equivalent conicity 0.2, 0.3 and 0.4, Type 1 and Type 2.*

Testing

High-speed testing with Gröna Tåget (Green Train) is made on different Swedish tracks during the years 2006–2009. Stability testing on straight track is mainly performed on two tracks on the 38 km section Skövde–Töreboda on the main line Stockholm-Gothenburg. There are two types of track, one of them usually producing an equivalent conicity of 0.25–0.3 with the usual S1002 wheel profiles, with limited sections of conicities between 0.5 and 0.8, i.e. extremely high. The high conicity is due to a tight track gauge (down to 1429 mm) and a flat rail top. The other track produces more normal conicities between 0.1 and 0.2. Both tracks have local lateral track irregularities close to the *safety track* characteristics of track standard QN3. These tracks are very useful for "worst case" testing over a wide range of conditions. According to the test report [18], the maximum evaluated ΣY_{100rms} is 10.2 kN at 269 km/h. This is about half of the simulation results and a third of the limit value. This is likely due to the track irregularities, which are smaller on the test track than with the very large perturbation on the simulation track (c.f. Section 5.6). However, generally in this work there is no attempt to compare simulation and test on exactly the same track conditions. The highest evaluated peak value of wheelset forces ΣY_{2m} (sliding mean over 2m) is 27 kN, which is 45 % of the limit value. The conclusion is that the soft bogies run stably, as predicted in simulations, with a large margin.

6.3 Ride quality

The ride quality is simulated on *comfort track*, both on straight track and in large-radius curves. Figure 14 and 15 show lateral *Wz* as examples. It is seen that the ride on straight track is good, while it is "just tolerable" in large-radius curves. These results are almost independent of the actual wheel-rail combination (i.e. equivalent conicity).

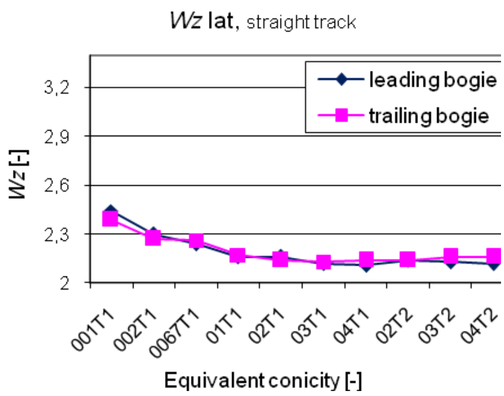


Figure 14 Simulated ride quality (Wz lateral) on the carbody floor above the bogies on straight track.

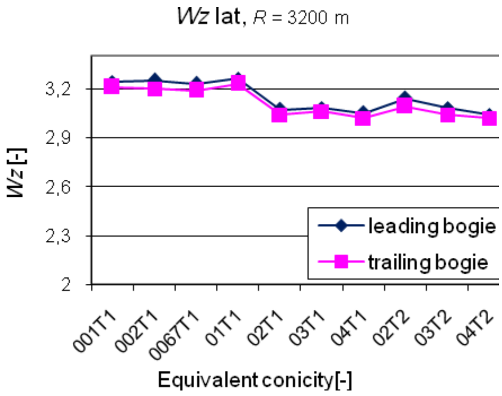


Figure 15 Simulated ride quality (Wz lateral) on the carbody floor above the bogies in a curve with R = 3200 m.

In large-radius curves, quite large lateral low-frequency periodic motions are excited in the carbody. This is due to the hard contact between the lateral buffer stop between carbody and bogie, at the relatively high cant deficiency. The resulting suspension stiffness is much higher than the stiffness in the air springs, which are acting alone at lower cant deficiencies. There are essentially two possible solutions to this problem: (1) to widen the lateral suspension travel, thus avoiding bumpstop contact at desired cant deficiency; (2) to introduce an active lateral suspension (ALS), which allows a soft suspension while still limiting the suspension travel.

Solution (1) is not attractive as it will result in reduced carbody width and will shift the centre of gravity laterally. Instead, solution (2) was selected and an ALS was developed and tested, described in [10]. However, an interesting result from the testing is shown in Figure 16. It shows that the introduction of an active lateral suspension will substantially improve the ride quality at high speed and high cant deficiencies.

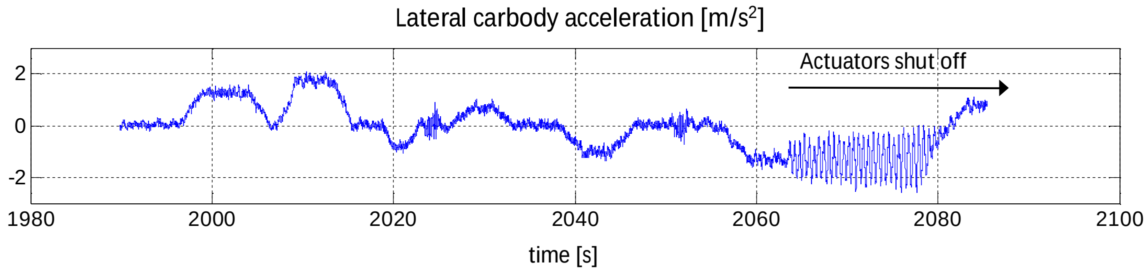


Figure 16 Measured lateral carbody acceleration above the trailing bogie in the leading car. Low-frequency periodic motions in large-radius curves at high speed and cant deficiency occur when the active lateral suspension (ALS) is shut off.

In terms of *Wz* the resulting improvement is very clear. Test results from a number of large-radius curves show that periodic low-frequency motions do not occur in all curves, likely due to the very good track quality. When periodic motions do occur the resulting lateral *Wz* is usually in the order of 3.0–3.2, i.e. close to what is predicted in simulations. With active lateral suspension (*ALS*), the worst *Wz* values are in the order of 2.4–2.5, with an average of 2.3. Thus, the optimization process showed a possible problem, which was solved by appropriate means.

6.4 Lateral track-shift forces

Simulations and tests are also made regarding lateral wheelset-to-track forces ΣY_{2m} (sometimes called track-shift forces); see Figure 17. Simulation and test results show good agreement, although track irregularities are not exactly the same. Due to the different tracks with varying geometrical quality, there is a spread in the measured results. However, it is concluded that the maximum forces in normal curves are less than half of the limit value. In the tightest curves however, the maximum evaluated forces are about 65 % of the limit value.

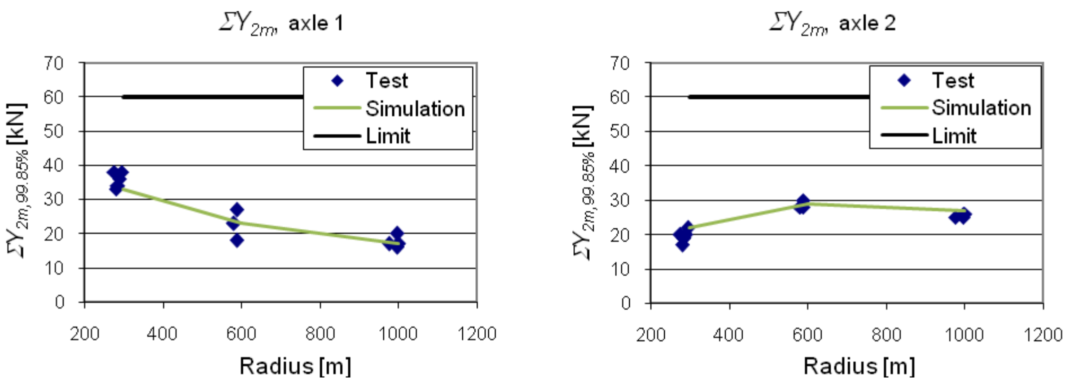


Figure 17 Simulated and tested ΣY_{2m} (99.85 percentiles) on the leading bogie wheelset 1 and 2, in small- and medium-radius curves.

6.5 Lateral wheel-rail guiding forces

No particular requirements have been called for, except that the guiding forces Y_{qst} must meet the limit value of 60 kN, according to UIC 518. This criterion is met with margins of 25–40 % in the tightest curves with radius around 300 m [18]. The lower values ($Y_{qst} \approx 35$ kN) are in accordance with simulation predictions on dry track. The higher values ($Y_{qst} \approx 40$–45 kN) were achieved on a track with rail lubrication on the high rail, thus limiting the self-steering ability of the leading wheelset. However, on lubricated track the wear should be low anyhow. Further, UIC 518 requires that acceptance testing shall be performed on dry track. Guiding forces in larger curve radii (600 m and above) are 18–23 kN, which is also in accordance with simulation predictions.

There is a strong linear relationship between guiding force Y_{qst} and the wear number, if the contact conditions are the same. As the predictions on Y_{qst} are close to reality, we draw the conclusion that also the predicted levels of wear (Section 6.1) should be approximately right.

7. CONCLUSIONS

The described optimization process has addressed a number of issues:

(1) The importance of an overall optimization, taking account to all essential requirements of track friendliness, stability and ride quality;

(2) The importance of thorough consideration to different operational conditions, such as speed, cant deficiency, track quality and wheel-rail contact conditions;

(3) The possibility of developing bogies with “soft” wheelset guidance also for high speed operation;

(4) The possibility of taking due consideration to specific requirements on local rail networks, such as the Swedish network.

ACKNOWLEDGEMENTS

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APPENDIX

A1. Dynamic model of track

The lateral stiffness and viscous damping of the track with reference to a single rail is k_{yt} = 30 MN/m and c_{yt} = 142 kNs/m (for the sum of left '*l*' and right '*r*' side), respectively. The corresponding vertical stiffness and damping is k_{zt} = 75 MN/m and c_{zt} = 94 kNs/m, respectively. The "contact stiffness" is given by k_w = 22 MN/m and c_w = 100 kNs/m, which is perpendicular to the contact patch. Thus, the smallest resulting lateral stiffness between wheel and ground is 12.7 MN/m. The "contact stiffness" is intended to include also the wheelset flexibility as an approximation.

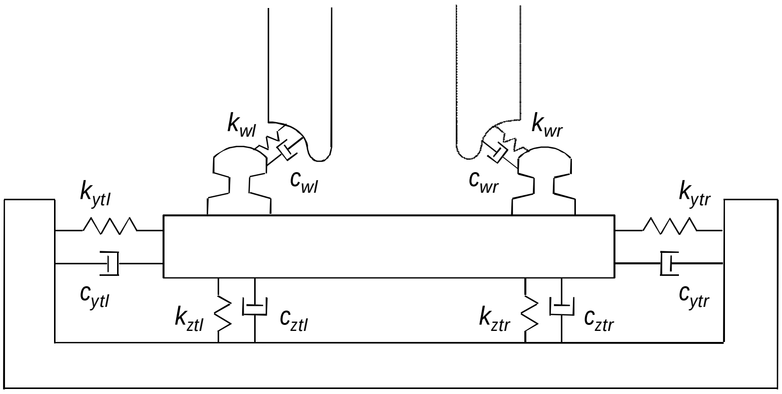

Figure A1 Track model in SIMPACK

A2. Contact geometry combinations

Ten different contact geometry combinations (with various wheel and rail profiles, track gauge and rail inclination) have been regarded in the study, resulting in different values of equivalent conicity λ_{eq}. Two types of equivalent conicity have been taken into consideration, Type 1 and Type 2.

Table A1 Contact geometry combinations

Wheel profile	Rail profile	Track gauge (mm)	Rail inclination	Type	λ_{eq}	Designation
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