Running dynamics concept with mechatronic guidance

The two-axle intermediate waggons of the DLR's Next Generation Train (NGT) have single-axle running gears with independently rotating wheels (IRW) and mechatronic track-guidance. This enables centring in the track and active radial steering for IWR pairs during curve passing. The wheel wear and noise generation can thus be considerably reduced. Multibody simulations are used to verify and optimise the dynamic behaviour. Moreover, detailed approaches for simulating high-frequency wheel-rail dynamics are being introduced in the project.

Dr.-Ing.
Bernhard Kurzeck

Scientist

DLR-Institute of Robotics and Mechatronics, Systems Dynamics and Control, Oberpfaffenhofen

Bernhard.Kurzeck@dlr.de

Dipl.-Ing.
Ingo Kaiser

Scientist

DLR-Institute of Robotics and Mechatronics, Systems Dynamics and Control, Oberpfaffenhofen

Ingo.Kaiser@dlr.de

1 Introduction

The NGT is a double-decker high-speed train enabling continuous passage to passengers on both levels. The train concept provides two-axle intermediate waggons with a length of 20 metres, because both the advantages of lightweight construction for the car body and the axle load of the individual running gears can be optimally exploited at this length. Conventional wheelsets can't be used in the running gears for space reasons, so all of the running gears receive IRW pairs. The end waggons are equipped with two-axle running gears.

The abandoning of the traditional bogie with wheelsets associated with this concept offers considerable scope for critically rethinking certain typical functional and design characteristics. At first glance, the simplest running gear consists of an independently rotating wheel module containing all the necessary components such as brakes, wheel-hub drive, suspension, and track guidance, and four of the identically-constructed modules just mounted on the car body. However, it's questionable whether this approach meets all of a rail vehicle's requirements.

Thus basic questions will first be investigated after summarising the boundary conditions resulting from the train concept for the running gear: How many suspension levels are actually required in the running gear? Can the requirements for ride comfort and safety be met with the usual passive springs and dampers? Based on this, the running gear's structure is developed with mechatronic track-guidance and its dynamic behaviour is verified with multibody system (MBS) simulations. Thanks to novel approaches, dynamic adjustment of the double-axle intermediate waggon can be realised for the most part with energy-saving passive elements.

Furthermore, verification, particularly in terms of wear and acoustics, requires improved methods of analysis that also include the higher-frequency phenomena of the wheel and rail dynamics. The development of appropriate methods and models will thus be presented at the end of the paper as well.

2 NGT running gear requirements from train concept and standards

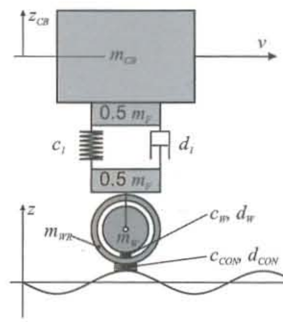
The double-deck construction is an essential component of the train concept (cf. RTR Special – NGT: Systematic derivation of the NGT rail vehicle concept). To offer the passenger overall greater comfort, the NGT will be barrier-free over the entire length of the train on both levels – thus also over the running gears. To prevent the wheelset axle from protruding into the passageway when wheelsets are used, their diameter would be restricted to less than 500 mm because of limited vehicle height. However, the shorter service life due to more frequent overruns and higher pressure peaks, as well as higher rotational speeds make wheelsets with such small wheels appear less suitable. The NGT is therefore equipped with IRW pairs with normal diameter, connected by a cranked beam, which can be placed under the floor.

All wheels are individually driven in the NGT (cf. RTR Special – NGT: The NGT propulsion concept). Each of the intermediate waggons' wheels has its own 260 kW traction motor that provides the necessary torque without gearbox transmission. A diameter of 1250 mm was selected for the wheels taking the traction motor's dimensions into consideration. The traction motors also serve as actuators for active track guidance and for radial steering.

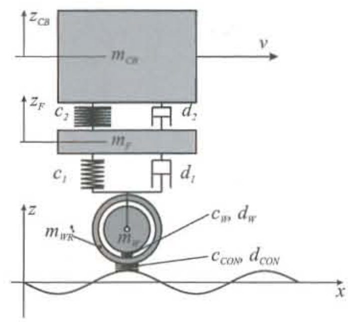
In addition, valid approval regulations (e.g. [1, 2]) must be considered during running gear development. Although the TSI [1] allows a 17 t axle-load, the axle-load for the NGT is set at 16 t, because the limiting value of 160 kN for dynamic wheel force was already nearly reached in simulations during a preliminary investigation with this axle-load.

The use of IRW pairs makes it possible to overcome the usual goal conflicts between low-wear curve running and large stability reserves at high speeds. However, the conventional wheelsets' passive self-centring mechanism must be abandoned for these benefits, and active track-guidance must assume this function in the NGT. An inadequate ability to self-centring in the track holds the risk of increased and, moreover, unevenly distributed wear on the wheel profile. Due to the wheel base of 14 m, which is the very large for a two-axle vehicle, mechatronic track-guidance must also include active steering for the wheel pairs' radial alignment. During curving, the wheel pairs are steered in such a way that any sliding, and thus wear, is minimised. One of the preconditions for curve squeal is eliminated as a consequence of the minimal sliding, thereby achieving considerable noise reduction.

Single level suspension (1)



Two level suspension (2)



			Single level suspension (1)				Two level suspension (2)			
			standard wheel (A)		rubber-sprung wheels (B)		standard wheel (A)		rubber-sprung wheels (B)	
			wheel hub motor (c)	suspend-ed motor (d)	wheel hub motor (c)	suspend-ed motor (d)	wheel hub motor (c)	suspend-ed motor (d)	wheel hub motor (c)	suspend-ed motor (d)
configuration			1Ac	1Ad	1Bc	1Bd	2Ac	2Ad	2Bc	2Bd
carbody mass	m_{CB}	[kg]	6250	6750	6250	6750	6000	6000	6000	6000
frame mass	m_F	[kg]	–	–	–	–	500	1000	500	1000
wheel mass	m_W	[kg]	1750	1250	1550	1050	1500	1000	1300	800
wheel rim mass	m_{WR}	[kg]	–	–	450	450	–	–	450	450
secondary stiffness	c_2	[kN/mm]	∞				0.222	0.176	0.218	0.175
secondary damping	d_2	[kNs/m]	∞				5.500	10.300	6.000	10.300
primary stiffness	c_1	[kN/mm]	0.163	0.175	0.163	0.175	0.505	1.295	0.525	1.330
primary damping	d_1	[kNs/m]	9.750	10.500	9.750	10.500	0.000	1.000	0.000	1.000
wheel stiffness	c_W	[kN/mm]	∞		130.0		∞		130.0	
wheel damping	d_W	[kNs/m]	∞		130.0		∞		130.0	
contact stiffness	c_{con}	[kN/mm]	990.0							
contact damping	d_{con}	[kNs/m]	100.0							

Table 1: Structure and parameters of the vertical quarter model

Various concepts have already been presented in the past [3–5], the implementation in practice of which has, however, failed due among other things to the requirement for a robust sensor concept.

The unusual car-body geometry (tall and short) together with an axle base of 14 m, which is significantly shorter compared to the pivot distance of conventional bogie vehicles, produce modified dynamic properties. In addition, the bogie's effect of levelling track unevenness doesn't apply in single-axle running gears. Nonetheless, the NGT should exhibit 20% better ride comfort (20% lower comfort coefficient) vis-à-vis current vehicles with a 20% greater wheel running distance.

3 Basic running gear design considerations

3.1 Necessary number of suspension levels

In general, high-speed trains have a two-level suspension consisting of primary suspension between the axle bearings and bogie frame, and secondary suspension between the bogie frame and car body. Partially rubber-sprung wheels are employed as a third suspension level in vehicles that travel in areas with permafrost soils, and in street cars.

While the secondary suspension's properties essentially determine ride comfort, primary suspension in the vertical direction mainly ensures sufficient twisting flexibility of the bogies and decoupling of the running gear's mass from the wheel to reduce dynamic wheel forces.

In the two-axle NGT intermediate waggon, the required compliance is achieved with one suspension level. But is a single-level suspension also sufficient for ride comfort? Can a suspension in the wheel compensate for the resulting increase of the unsprung masses due to the wheel-hub motors?

To investigate these questions, a vertical quarter-model of an intermediate waggon is being used. It consists of a wheel, the proportional mass of the running gear frame, and a quarter of the railcar's body mass. Variations of the model with single- and two-level suspension, each constructed with and without suspension in the wheel, as well as with wheel-hub motor or suspended drive, result in eight variations (Table 1). The following similarities are defined to enable a comparison among the eight variations, and thus a generally valid statement:

- ▷ A single-level suspension evenly distributes the frame's 500 kg mass over the car body and wheel.
- ▷ The traction motor's mass is estimated at 500 kg. With a wheel-hub motor, it's considered part of the wheel; otherwise, it's part of the frame.
- ▷ The wheel's mass (incl. support, proportionately axle carrier and guides) increases by 250 kg with rubber-sprung construction. The entire mass is distributed across two bodies in the process.
- ▷ The suspension level's stiffness and damping are set so that the car body's eigenfrequency is 0.8 Hz and the running gear frame's frequency is 6.0 Hz, both with a damping ratio – if possible – of 0.15.

A very stiff linear contact spring, the base point of which experiences a vertical displacement excitation, z(t), corresponding to typical track irregularities, is assumed between the wheel and rail. The calculation is carried out in the frequency domain with the SIMPACK MBS simulation software.

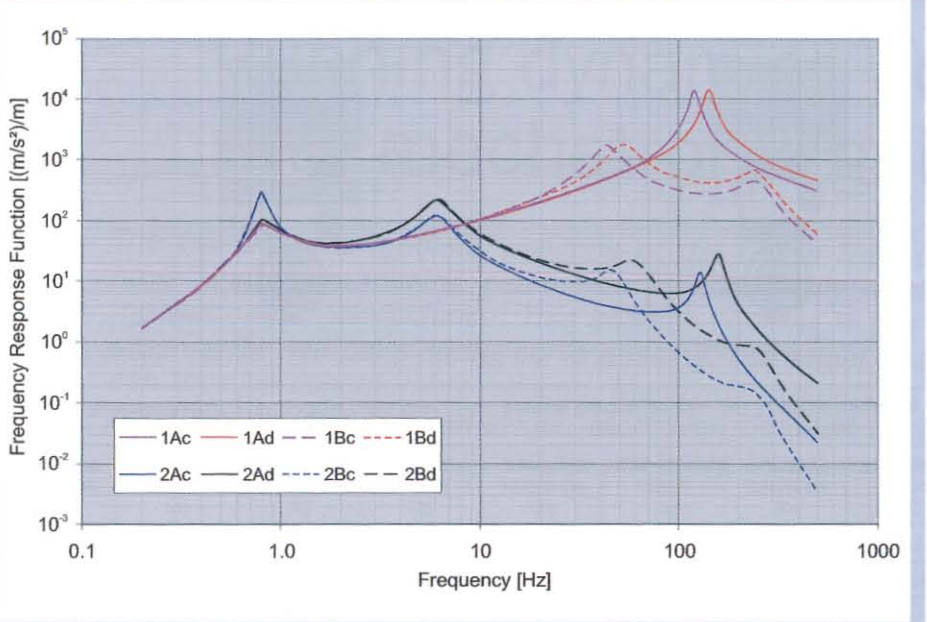


Figure 1: Vertical frequency response function (excitation: vertical displacement contact spring; response: acceleration at car body)

The peaks in the transfer function between vertical excitation from track irregularities and the vertical car body acceleration (Figure 1) are related to the eigenfrequencies. The car body's eigenfrequency is 0.8 Hz. The second eigenfrequency occurring for the variants with secondary suspension is 6 Hz. The eigenfrequencies of the rubber-sprung wheels are 60 Hz. The vehicle vibrates on the contact spring at eigenfrequencies above 100 Hz.

The transfer function increases in the region around an eigenfrequency and declines precipitously at higher frequencies. A two-level suspension only provides advantages over a single level for frequencies greater than 8 Hz. Each additional suspension level leads to improved vibration isolation only in the region above the eigenfrequency.

To be able to compare comfort-related vertical acceleration amplitudes in the car body for the entire frequency range up to 100 Hz, the transfer functions from Figure 1 must be multiplied by the spectrum of typical vertical track irregularities to account for the greater excitation in the low frequency range due to the different amplitude levels of long- and short-wave track irregularities. The resulting acceleration amplitudes are filtered according to ISO 2631 [6] (comfort rating). RMS values are subsequently added. The results are presented in Figure 2 as

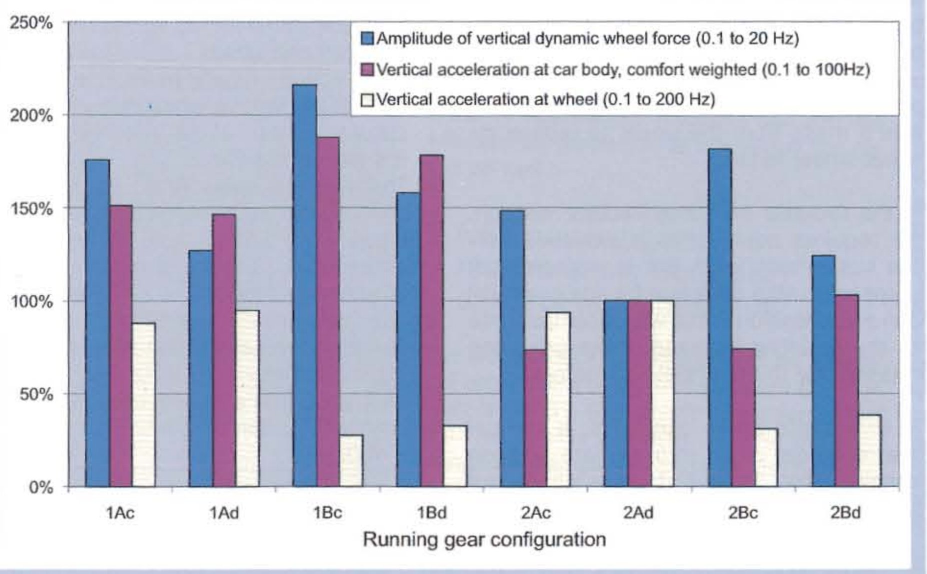


Figure 2: Comparison of the eight configurations relative to configuration 2Ad: dynamic vertical wheel force and acceleration at car body and wheel disc (frequency-range mean of 0.1 Hz up to 20 Hz; 100 Hz or 200 Hz)

the values related to variation 2Ad, which essentially corresponds to the current standard. Analogously, vertical dynamic wheel forces are identified in the frequency range up to 20 Hz, considered approval-relevant according to EN 14363, as are accelerations on the wheel up to 200 Hz.

In direct comparison, variations with rubber-sprung wheels exhibit greater acceleration in the car body resulting in poorer ride comfort. One of the rubber-sprung wheels' positive effects only appears above the eigenfrequency beginning at about 80 Hz and no longer has an effect on comfort-rated acceleration amplitudes up to 100 Hz due to the excitation characteristics and the comfort rating. A lower radial stiffness does increase the rubber-sprung wheels' effect on comfort, but very low wheel stiffnesses on the order of magnitude of the primary suspension (1–2 kN/mm) are required. Such a low stiffness is critical because of the great static deflection, and therefore increased flexing resistance from the damping in the rubber. Wheel designs with nearly damping-free springs are conceivable in principle, but disadvantageous for traction behaviour because of the resulting undamped wheel-rim dynamics.

The variants with hub motors (index c) have the greatest dynamic wheel forces, especially in combination with rubber-sprung wheels (1Bc, 2Bc). This is because the wheel's mass increases and the wheel spring is hardly effective in the approval-relevant frequency range up to 20 Hz. Wheel stiffness would also have to be considerably decreased to reduce the dynamic wheel/rail forces. However, rubber-sprung wheels very severely diminish the wheels' acceleration amplitudes in the frequency range up to 200 Hz, and thus a wheel-hub motor's dynamic load as well. Nonetheless, rubber-sprung wheels are advantageous in high speed trains (HST) if, besides stochastic excitation from track irregularities, excitations in the higher frequency range such as corrugation or wheel polygons occur.

In order not to jeopardise an approval, the unsprung mass per wheel should not exceed 1000 kg. This is because previous simulations for a vehicle model, comparable to the 2Ad variation, exhibit only small reserves at the limit value so that the wheel-hub motor can no longer be traced as long as the motor's mass is not considerably reduced. This also eliminates the need for rubber-sprung wheels. Consequently, the 2Ad variation with two-level suspension, suspended drive, and rigid wheels is the best solution in terms of ride comfort and dynamic wheel loads. Similar considerations were also carried out for the horizontal plane.

Further investigations were conducted to determine whether an additional vertical degree of freedom between both IRW of one pair offered advantages. Ultimately, the dynamic benefits were minimal compared to the additional mechanical complexity of a sufficiently rigid vertical guide, with which the tolerances required for track guidance would also have to be adhered to.

3.2 Mechatronic track-guidance

The enhancement of the "mechatronic wheelset" forms the core of the NGT running gear [4]. Its basic idea consists of an IRW pair, both wheels of it having an independent drive, so that the rotational speed of both wheels can be individually regulated. A sensor continually detects the wheel pair's lateral position in the track. If the pair of wheels shifted so far laterally in the track that flange contact impends, the affected wheel is accelerated slightly and the wheel on the other side simultaneously slowed, creating a torque around the vertical axis. This introduces a steering motion of the pair of wheels – if the wheel pair has the proper degree of freedom in the running gear frame. The wheel pair is thereby directed into the middle of the track. In this relation, a decreasing oscillation with overshoot is even beneficial for uniform wheel wear. In principle, mechatronic track-guidance allows a relatively free design of the wheel profiles. For compatibility reasons, the NGT's wheels receive the standard S1002 wheel profile.

Controller requirements vary greatly for different operating situations. At high speed on straight track, running both wheels of a pair with the same rotational speed is sufficient in practice, so that self-centring occurs via the wheel profiles' conicity, provided the usual wheel profiles are be used. Here, only small control interventions are necessary to prevent instabilities even at maximum speed. In tight curves, however, wheel rotational speed must be more vigorously adjusted for radial steering due to the long wheel base.

A new running gear was built to a scale of 1:5 for the DLR roller rig to develop and to test a suitable sensor concept (Figure 3). The running gear has two mechatronic IRW pairs, which indirectly detect the running gear's position on the tracks using force/moment sensors integrated into the wheel axle. The actuator system (synchronous motor) and the other sensors are also integrated in the wheel.

In addition, an MBS model of the roller rig, with which the design and optimisation process is supported, is being created in SIMPACK. Due to the short wheel base in the roller rig, the running gear on the roller rig corresponds to the double running gear in the end waggon. An expanded model for the intermediate waggon with two IRW running gears is derived from the simulation model validated on the roller rig.

Besides the sensor concept, a global control concept for mechatronic NGT running gears should be developed and tested mid-term. For safety reasons, this should then also include the required locking system or other fail-safe level, because large error angles might occur when the controller fails or malfunctions due to the large steering angle of up to 3° in the intermediate waggon's running gear.

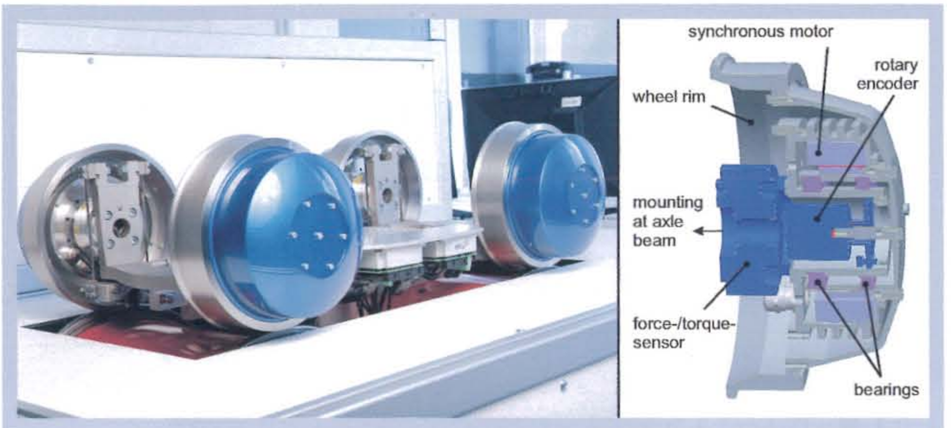


Figure 3: Test running gear with independently rotating wheels at scaled 1:5 DLR roller rig

4 Chassis-frame structure

A running gear with the following characteristics offers the best compromise between the requirements for track guidance, ride comfort, wheel and rail forces, rolling resistance, total weight, and robustness (Figure 4):

- ▷ Two-level suspension
- ▷ Rigid connection of both IRWs via a light-as-possible structure (wheelmount)
- ▷ Stiffer primary suspension between the wheel pair and the running gear frame
- ▷ Rotational degree of freedom of the wheel pair on the vertical axis relative to the frame for a radial alignment
- ▷ Attaching the traction motors to the frame (low unsprung masses)
- ▷ Softer secondary suspension between the frame and the car body

A third suspension level in the wheel (rubber-sprung wheel) offers no advantage with currently realisable stiffnesses. The connecting beam is designed cranked to enable the passageway in the lower level to be placed at the required depth. The rotational degree of freedom for steering within the primary suspension reduces the inertial mass that has to be moved during the steering procedure.

Torque transmission from the suspended motor to the wheel occurs via a drive shaft, which is placed in the motor's hollow shaft. The design presented integrates the essential parts of the electrical machine, already conceived for the wheel-hub design, in a suspended concept. The bearings are dimensioned for a service life of roughly 5 m kilometres.

5 The NGT running gear's dynamic designs

5.1 NGT model

The aim of the running gear's design is to determine the suspension level's optimal mechanical parameters. At first, only the mechanical degrees of freedom between the wheels, motors, car body, and running gear frame are defined in the MBS model without thereby taking into consideration their technical realisation.

For the simulations with SIMPACK, a shortened NGT unit comprising four intermediate

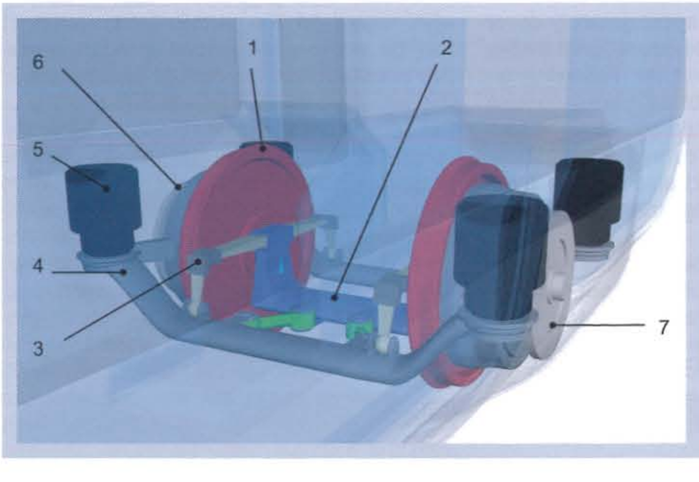


Figure 4: Design study for the running gear

1 wheel
2 axle beam
3 primary suspension – e.g. piezo-actuated composite leaf spring
4 running gear frame
5 secondary suspension
6 traction motor
7 brake disc

		Test track 1	Test track 2	Test track 3
Speed	[km/h]	44	100	400
Curve radius	[m]	150	600	8500
Super elevation	[mm]	0	120	170
Slope	[%]	-	1	2
Track irregularities		ERRI high	ERRI high	conform to EN 14363
Simulated time	[s]	60	120	100
Percentage of total running distance	[%]	3	7	90

Table 2: The three test tracks' parameters

cars and two end cars is being used. Four intermediate waggons form a reasonable compromise, on the one hand to enable investigation of the dynamic interactions between vehicles when inter-car dampers and the like are used, and on the other hand, not to increase the computational cost unnecessarily.

Two wheel pairs are integrated into a bogie frame on the end waggon. The components' masses and inertias come from the initial design sketches and therefore still exhibit some uncertainties. Apart from active lateral centring device in the secondary suspension and the mechatronic track-guidance, all of the suspension's other elements are passive.

5.2 Reference vehicle

A single-decker, four-axle single car with bogies and conventional wheelsets serves as the reference vehicle. This reflects the approximate development status of a current, single-decker HST vehicle. The wheel diameter is 920 mm. The vehicle's weight is defined to be 64 t corresponding to a wheelset load of 16 t. The bogies have a relatively stiff wheelset guidance and a rather soft secondary suspension, which consists essentially of an air suspension. All of the wheelsets are motorised in the model.

5.3 Track scenario

Three test tracks that approximately cover the spectrum of possible operational tracks are used for the simulation. Following an initial straight line, all three tracks contain an S curve with intermediate straight sections and transitional curves. In addition, tracks 2 and 3 (Table 2) also contain a descending- and ascending-slope section.

The vehicle model's speed control guarantees nearly constant running speed during the simulation. The traction motors generate the necessary drive or braking torques, which creates additional wear on the wheels from the traction, although the driving-resistance forces from air resistance are neglected.

5.4 Mechatronic track-guidance

A PID controller is temporarily used for mechatronic track-guidance in the simulations, since the current development of the global control algorithm is not yet finished. A nominal angular speed for all of the wheels is calculated based on the current driving speed and the difference from the nominal speed. The nominal angular speeds for the left and right wheel of a wheel pair are calculated from the current lateral displacement of a pair of wheels in the track and a virtual rolling radius function. A PID controller sets the angular speed via the traction motor's torque. The control parameters were empirically determined previously for the three test scenarios (cf. sec. 5.3). Control parameter adjustment occurred according to the following criteria:

- Minimal wheel wear
- Minimum transfer from longitudinal shocks to the car body
- Low level of lateral acceleration at the running gear frame level
- Minimal energy expenditure (as far as possible)

After excitation by a lateral track irregularity, the controlled wheel pair performs a decaying oscillation between the two rails, which is comparable to the hunting motion of a wheelset and is important for uniform wear distribution on the wheel profile. While the dynamic behaviour of fixed, mechanical parameters is determined for the reference vehicle, decaying oscillation frequency, and thus the running gear's lateral dynamics with regard to the comfort requirements can be actively influenced through operative adaptation of the control parameters in the regulated wheel pair, which is another of the active control system's advantages: the controller's intervention frequency can be properly adjusted in the regulated system in operating situations in which resonance occurs between the running gear and vehicle body dynamics in conventional systems. The applied temporary track-guidance regulation takes control of the traction motors even more vigorously than should be the case with a real vehicle. Short term, very high moment peaks are required, which at a speed of 400 km/h would have to overload the traction motor by 100%. The aim of further development must therefore be a track-guidance controller that restricts regulating interventions so that the traction motors and the power converters don't become overloaded.

5.5 Ride comfort

According to established expectations, the two-axle intermediate cars with the initially selected spring stiffness and damping parameters exhibit unsatisfactory comfort behaviour compared to the reference vehicle, since shorter car bodies are generally more vibration-prone.

However, a global-train dynamics concept is being pursued to achieve better comfort than in the reference vehicle. This increases the individual car bodies' mass inertia through suitable coupling of the cars to reduce the effect of a single excitation. The running gear's suspension continues to support the car bodies' mass, and must ensure that the entire train follows the stretch as a unit and doesn't tip over. On the other hand, the suspension should transfer as little of the higher-frequency excitation from track irregularities as possible to the vehicle. Therefore, a large part of the damping from the secondary suspension is being relocated to dampers between the individual cars. Moreover, great progress is being achieved via inter-car anti-roll elements.

The criterion for average comfort (N_{MV} value) according to EN 12299:2009 [7] is used to assess ride comfort. In this connection, the acceleration signals are initially evaluated with comfort filters and then analysed statistically. Measurement points in the individual NGT cars are placed in the same way on each floor as in the reference vehicle.

N_{MV} values calculated for intermediate waggon 2 at 400 km/h on test track 3 are all considerably below 1.5 (Figure 5). According to the standard, the vehicle should be described as "very comfortable." The relative values related to the reference vehicle (each in the same measurement position) are more interesting. Due to the car body's pitching and yawing motions, ride comfort is fundamentally worse in the area over running gear than in the middle of the car body. However, since comfort values between the individual measurement positions in the NGT scatter significantly less than in the reference vehicle, the NGT provides a more consistently good ride for all seating positions. On average, the required 20% comfort improvement in the intermediate waggons was even exceeded by five percentage points.

5.6 Wear

The area in which the wear occurs, and the volume proportional to the frictional energy dissipated on each wheel is calculated in a postprocessing routine from the simulation results and averaged over all of the wheels. The wear parameters used were validated in a previous investigation. The results from the three test tracks are then weighted by the percentage of total running distance specified in Table 2 and extrapolated to a 10,000 km running distance. Overall, material abrasion on the NGT intermediate waggon's wheel declines by 32% compared to the reference vehicle and by 55% on the end waggons. The higher wear on the end waggons can be traced back to, among other things, to the fact that the end waggons are more heavily motorised.

Besides the absolute amount of material loss, the place on the wheel where the wear occurs is especially relevant. Figure 6 shows profile geometry in the upper portion and the wheel profile's calculated wear depth below. The wear cross section is on the whole lower with the NGT, which is also favoured by the larger wheel diameter (1,250 mm instead of 920 mm). The NGT running gears' active track-guidance in narrow curves leads to much better wheel position in the track and thus to considerable wear reduction, primarily in the flange area, where wear particularly critical to the wheel's service life has an influence. But active track-guidance also leads to an improvement in the running surface area, which is stressed primarily on straight track Wear is initially predicted for a 10,000 km running distance, because the wheel profile's geometric properties can't be assumed to remain unchanged for greater running distances. For a further prediction, the wheel's profile would have to be progressively adjusted as a function of wear condition with increasing running distance. That kind of expensive calculation is provided with the final model. It's currently assumed that reduced material abrasion will accordingly enable an extension of wheel running distance. Wear volume reduction is currently significantly above the required 20%, so that even moderately increased wear can be tolerated for the benefit of reduced engine torques.

6 Enhancements of the modelling methods

All results presented so far were computed by using the MBS simulation software SIMPACK originally developed at DLR. Today, this software is being commercially marketed and further developed as a SIMPACK AG product.

MBS modelling basically idealises reality by describing the real system through a system of discrete elements, mainly rigid bodies having a mass and massless connecting elements, which, besides force elements such as springs and dampers, also include joints and constraints to define the body's movement. A few MBS programs, including SIMPACK, offer the possibility of taking individual bodies' deformations into consideration.

The description of wheelsets and rails as flexible bodies within an MBS is especially difficult because the point where the wheel-rail forces act is constantly moving due to the wheels' rolling motion on the rails. Such modelling is nevertheless desirable for two reasons: First, wheels and rails are substantially responsible for the rolling noise. However, these kinds of investigation require consideration of the body's structural dynamics. Second, small deformations of wheelset and rail have an influence on the stress distribution in the rolling contact, and hence on wear. The wear at the wheel's running-surface in turn has a significant impact on the mileage and thus on the vehicle's cost-effectiveness.

Wear-behaviour investigation also requires an enhancement of the contact modelling. Since abrasive wear can generally only occur where wheel and rail are actually in contact with each other, the contact model must describe the actually occurring contact area and friction-energy distribution in more detail. The study of other phenomena, such as rolling-contact fatigue, requires more exact knowledge of the stresses occurring in the contact.

As part of the NGT project, new modelling methods for solving the problems present-

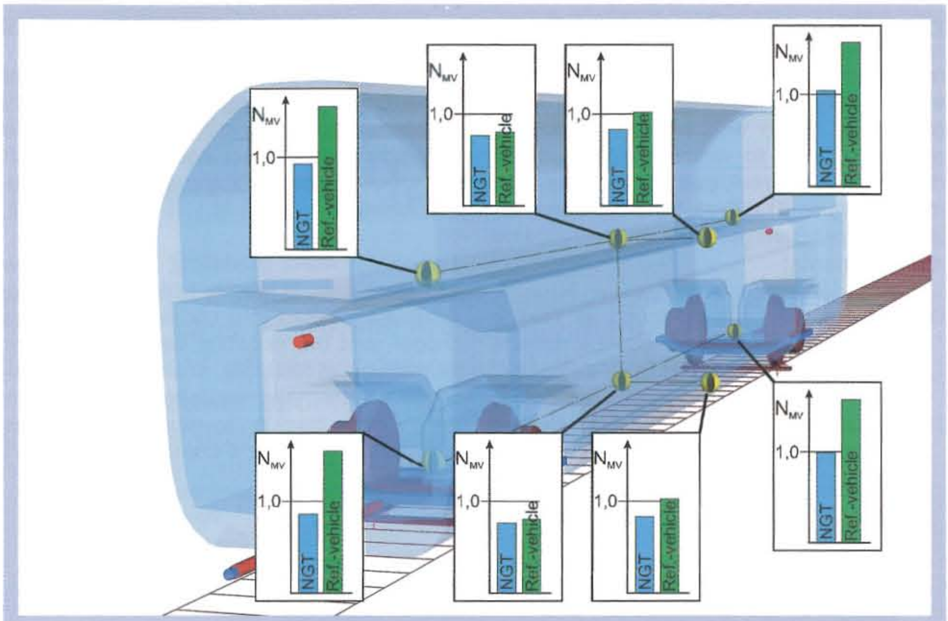


Figure 5: Comparison of the comfort numbers for the first NGT intermediate waggon and the reference vehicle at different seating positions

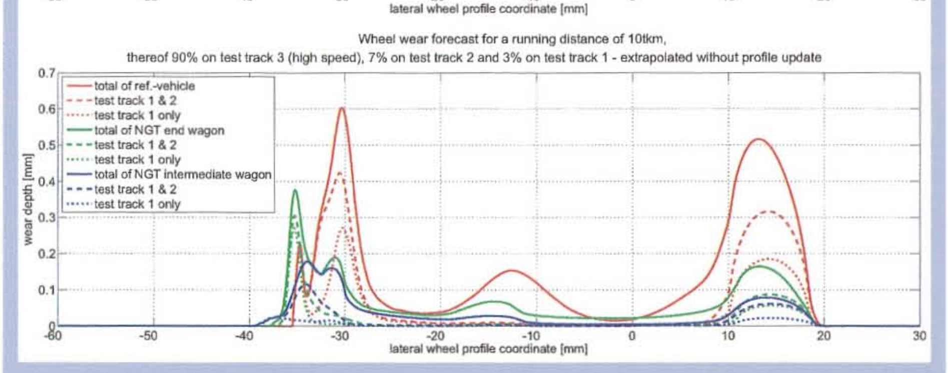


Figure 6: Forecast of the wheel wear for 10 000 km

ed are being developed in a non-commercial programme. In this development environment, the methods will be tested in a model describing the running of a conventional passenger coach with two-axle bogies on a straight track, to validate the results on the behaviour of real, existing vehicles. This model (Figure 7) differs from commercially available programmes through the following enhancements:

- ▷ The wheelsets are modelled as flexible bodies, whereas the gyroscopic effects resulting from rotation are expressed mathematically correct.
- ▷ The rails are also modelled as flexible bodies in which cross-sectional deformations can be described. Discrete railway sleepers provide support.
- ▷ Distribution of the normal and tangential stresses in the rolling contact is accomplished by an iterative solution of the discretised contact equations, in which non-elliptical contact surfaces can be included.

Especially when modelling the track, the structure's great length, which must be appropriately taken into consideration in the model, and the resulting very high number of degrees of freedom, poses a major challenge. Systematic exploitation of wheelsets' and track's symmetry properties provides a significant computational effort reduction without loss of accuracy, which contributes to modelling efficiency [8].

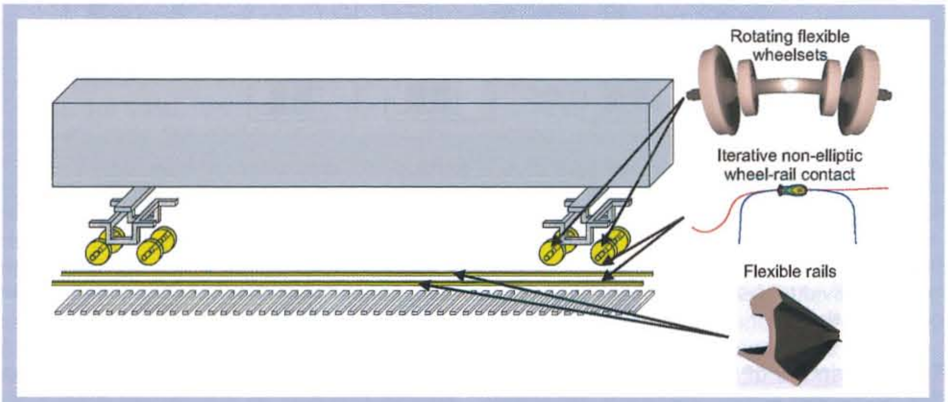


Figure 7: Overview of the vehicle-track model and enhancements compared to standard modelling

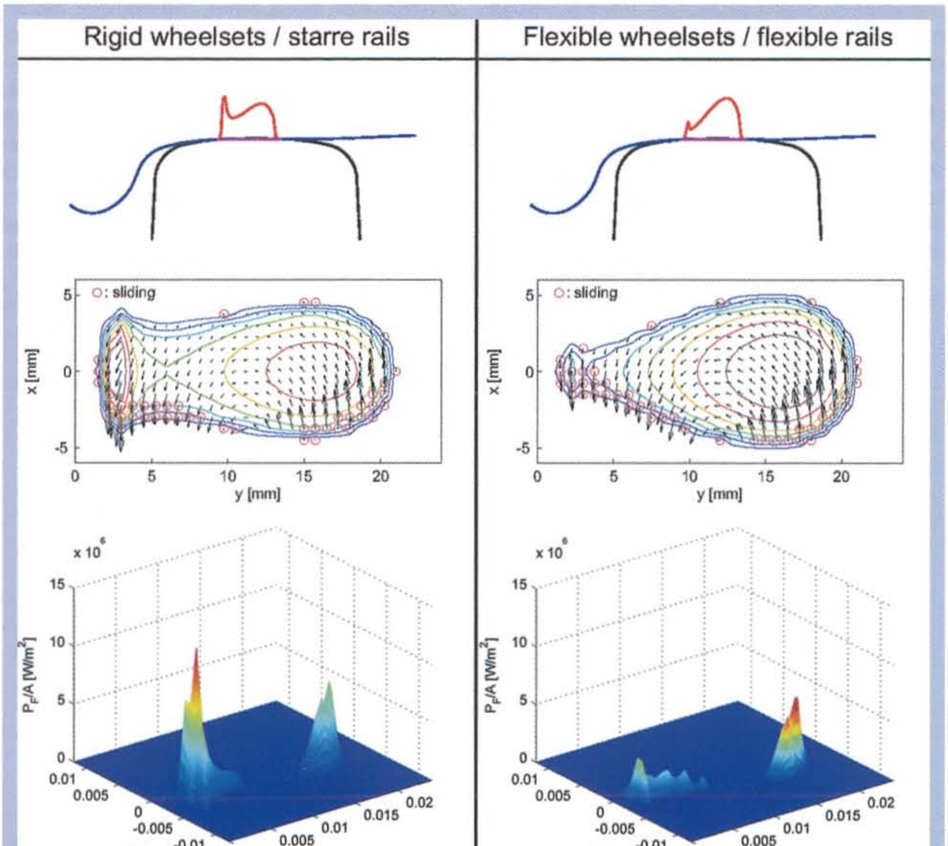


Figure 8: Contact geometry and normal stress (top), distribution of normal and tangential stresses (middle) and distribution of the frictional power density (down) obtained for modelling wheelsets and rails as rigid bodies (left) and as flexible bodies (right) for centred running at 200 km/h

The importance of accurate modelling is already reflected in the relatively "unspectacular" scenario of undisturbed centred running: The wheelset's and rail's structural dynamics has a significant influence on the contact (Figure 8). In particular, the contact angle changes due to the wheelset axle's deflection and thus also to a considerable extent does the contact area and the distribution of the friction energy occurring there. Further calculations for the scenario of hunting also show a clear influence of the wheelset's and rail's deformations at both the contact as well as on the entire vehicle's running behaviour. Altogether, this underlines the importance of consistency for reliable modelling, i.e. the modelling of all components involved. Wheelsets, rails, and contacts must therefore be refined in a comparable manner.

As the project progresses, the upgraded modelling methods should be used to further optimise the NGT concept. Models of the flexible wheelset and the flexible track were already implemented and tested for this in an in-house, DLR-developed version of the SIMPACK program. Improved rolling-contact modelling in SIMPACK will follow.

The use of higher-frequency models significantly improves the realistic prediction of both the vehicle's wear behaviour and its acoustics (cf. RTR Special – NGT: Activities in acoustics). Any problems can be detected early at low cost, appropriate remedial measures selected, and expected life cycle costs (LCC) estimated.

Sources

[1] 2008/232/EC: Commission Decision of 21 February 2008 concerning a technical specification for interoperability relating to the rolling stock sub-system of the trans-European high-speed rail system

[2] DIN EN 14363: Railway applications – Testing for the acceptance of running characteristics of railway vehicles – Testing of running behaviour and stationary tests, German version EN 14363:2005

[3] Bruni, S., Goodall, R., Mei, T. X., and Tsunashima, H.: Control and monitoring for railway vehicle dynamics, Vehicle System Dynamics, 45, 2007, pp. 743–779

[4] Gretzschel, M., Jaschinski, A.: Design of an Active Wheelset on a Scaled Roller Rig, Vehicle System Dynamics 41, 2004, pp. 365–381

[5] Schade, S., Hermanns, M., and Dellmann, T.: Mechatronische Spurführungsregelung zur aktiven Spurführung von angetriebenen Losradfahrwerken, ZEVrail 134, pp. 404–410

[6] ISO 2631-4:2001: Mechanical vibration and shock – Evaluation of human exposure to whole-body vibration – Part 4: Guidelines for the evaluation of the effects of vibration and rotational motion on passenger and crew comfort in fixed-guideway transport systems. International Organisation of Standardisation, Geneva, 2001

[7] Railway applications – Ride comfort for passengers – measurement and analysis; German Edition EN 12299:2009.

[8] Kaiser, I.: The Impact of Structural Deformations of Wheelset and Rail on Wheel-Rail Contact. CST2010: The Tenth International Conference on Computational Structures Technology, 14–17 Sept. 2010, Valencia, Spain