Chapter 2
Modeling of Vehicle, Track, and Excitation

2.1 Prior Considerations and Coordinate Systems

In order to be able to examine the dynamics of a vehicle, the vehicle must first be transformed into a mechanical model. For comfort calculations and stability investigations (sinusoidal motion), it is usually enough to create models that can reproduce the eigenfrequency of the system in the low-frequency range accurately up to about 25 Hz. In those cases, it is enough to model wheel axles and bogies as *rigid bodies* that are connected to the carbody and each other by springs and dampers. The carbody is usually modeled with six rigid-body degrees of freedom and the first elastic eigenmodes. One often tries to use the symmetries of the vehicle during these calculations, although railcars and locomotives rarely are built to be truly symmetric. Ignoring small imperfections in the symmetry gives the advantage that the *vertical model*, that is, the vertical and longitudinal movement, and the *lateral model* can each be examined separately.

Software systems such as ADAMS RAIL, GENSYS, MEDYNA, SIMPACK, VAMPIRE, and VOCO not only facilitate the work of the engineer to formulate and solve the equations of motions, but also provide postprocessing tools to calculate the measures needed to judge the amount of wheel/rail forces and ride comfort. These software systems all compute three-dimensional coupled vertical and lateral models. In this monograph, which is meant as an introduction, we will (almost) strictly distinguish between vertical and lateral dynamic phenomena.

Investigations of fatigue strength, the calculation of the forces between wheel and rail in the medium-frequency (40–400 Hz) range, the examination of vehicles running on corrugated rails (500–1500 Hz), and acoustic investigations require models that have the range of several thousand hertz.

Figure 2.1 shows a vehicle consisting of

- one carbody,
- two bogies, and
- four wheel axles,

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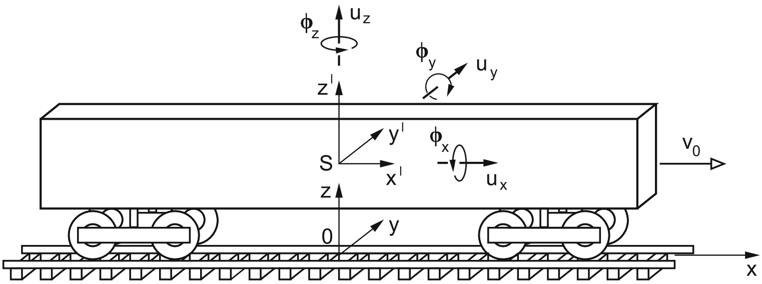


Fig. 2.1 Coordinate systems; nomenclature of motions

which we consider rigid bodies for simplicity. We differentiate between a rail-bound (fixed) (x,y,z)-coordinate system and an (x′, y′, z′)-coordinate system with velocity v_0 that moves with the vehicle. In this reference coordinate system, displacements and torsion are measured as well. These are considered to be *parasitic movements*:

u_z	vertical motion,
u_y	lateral motion,
u_x	longitudinal motion,
φ_z	yawing,
φ_y	pitching,
φ_x	rolling (often in combination with u_y lateral motion).

Rolling in combination with lateral motion is called sway or body rolling.

2.2 Vehicle Modeling

2.2.1 Bogie Frame Design, Primary Suspension, and Bogie Guidance

Figure 2.2 shows one type of Minden–Deutz bogie, which the the Deutsche Bahn has used in many of its advanced designs. The horizontal bogie linkage to the wheelsets is realized through maintenance-free elastic leaf spring guidance. The primary vertical suspension consists of coil springs. There are bogies in which rubber blocks are used as suspension in all three dimensions. In the figure of the cross section of the bolster (Fig. 2.3), the bolster beam as well as the central carbody support in a pivot bearing are shown. The lateral auxiliary or slide supports can be designed in a way that does not make them load-bearing but only roll or sway preventing in function.

They can also be load-bearing, in which case they are used as friction dampers. The bolster beam is suspended by cylindrical coil springs against the hangers. The

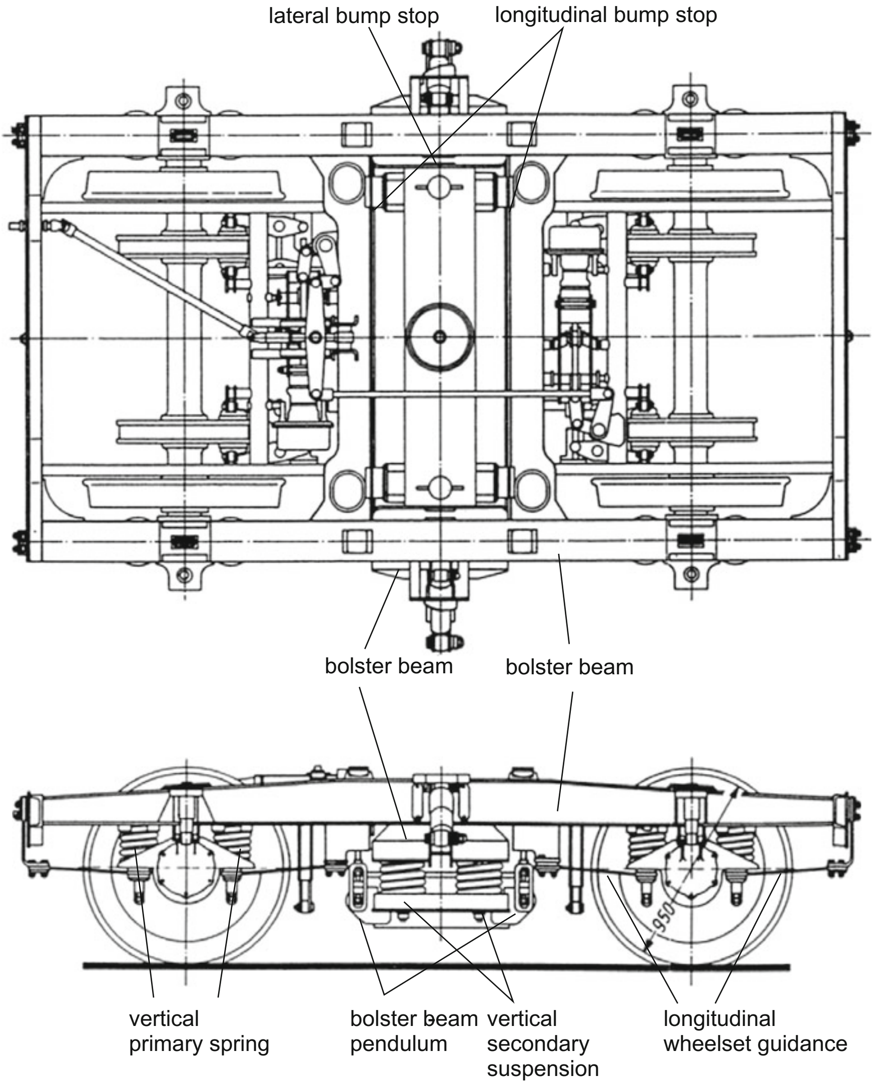


Fig. 2.2 Passenger coach bogie with two wheelsets, type Minden–Deutz with disk brake (MD 36), from [1]

dampers installed with an inclination result in effective damping in the vertical as well as in the horizontal plane transverse to the moving direction. In order to prevent the bolster beam from swinging too far, there are elastic buffers installed in the longitudinal and transverse directions. The clearance of the bolster in the longitudinal direction is usually 5 mm, and the transverse clearance can be up to 70 mm.

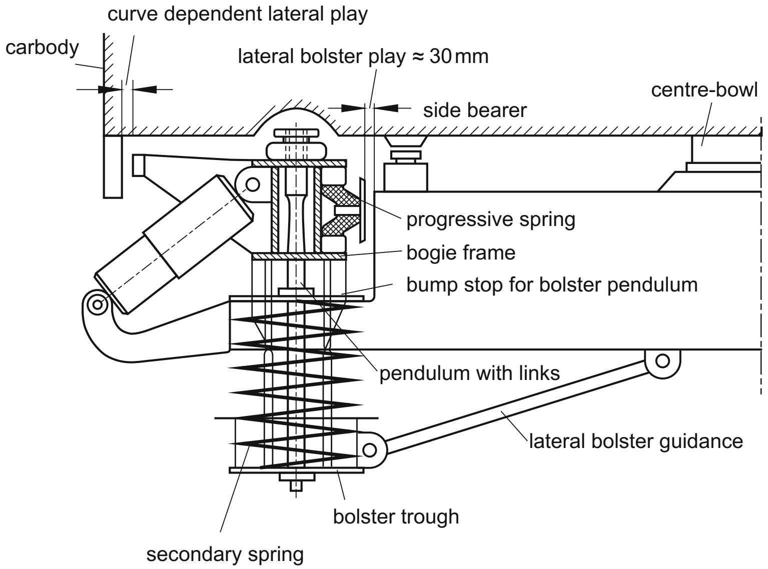
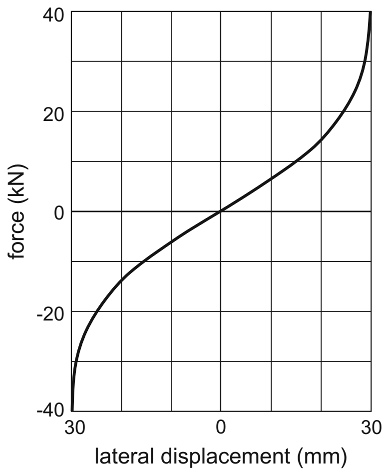


Fig. 2.3 Details of bolster pendulum arrangement of MD-bogie, from [2]

Fig. 2.4 Characteristics of lateral secondary spring



For the guidance of the bolster beam in the hangers, so-called transverse control arms—sometimes also nonwearing blade springs—are used, which do not decrease the freedom of movement of the bolster beam. Due to the limiting buffer and therefore the additional progressive springs used, the characteristic of the lateral secondary suspension becomes highly nonlinear (Fig. 2.4).

In bogies of Minden–Deutz type, there is in most cases a friction damper installed between the carbody and the bolster. On straight track, the friction damper does

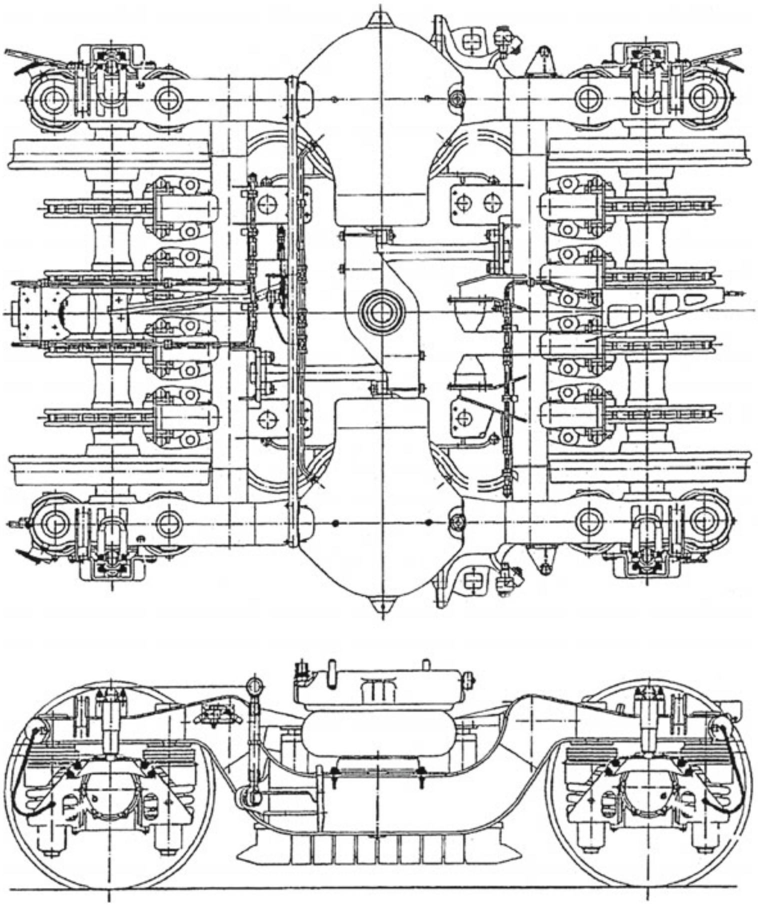


Fig. 2.5 Passenger car bogie SGP 400 for second-generation ICE trains. Four brake disks and eddy current brake

not usually break out. This occurs first on entering a curve. In case of sinusoidal movements strong enough to cause the friction damper to break out on straight track, it will act in a damping manner. The friction damper will have a stabilizing influence on the instability caused by hunting. For the second generation of the ICE, Siemens Verkehrstechnik GmbH developed the high-performance bogie SGP 400; cf. Fig. 2.5. The SGP 400 bogie does not contain any friction elements, removing one strong nonlinearity. Naturally, there are a number of other connecting elements (cf. Fig. 2.4) that contain nonlinearities. These, however, are unproblematic, since they can be linearized for small displacements.

The most commonly used bogie type in freight traffic is the Y25 bogie, which is shown in Fig. 2.6. The Y25 bogie contains a large number of friction elements and behaves therefore highly nonlinearly. A linear treatment is therefore not justifiable.

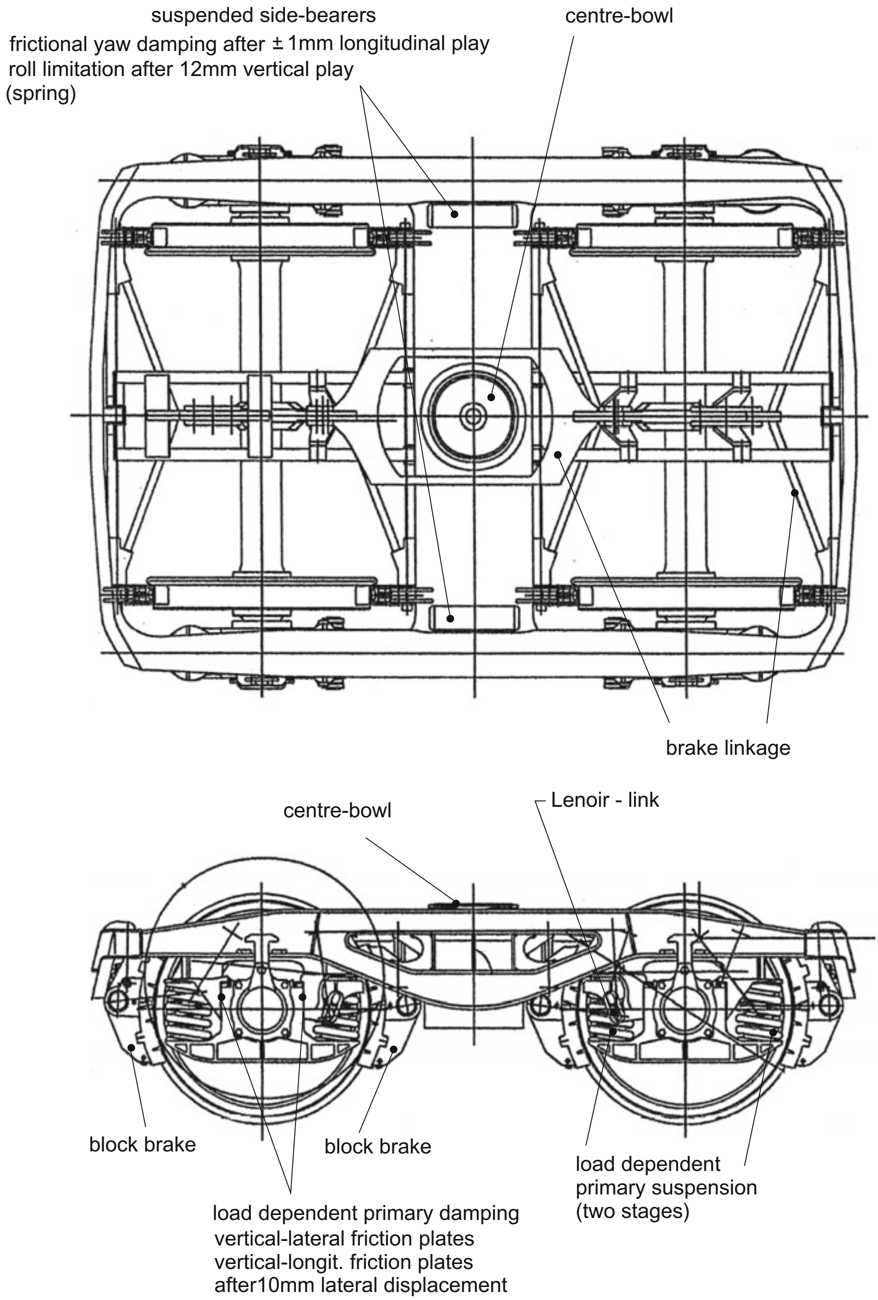


Fig. 2.6 Y25 freight wagon bogie with two wheelsets [1]

An overview of other bogies that is still worthwhile is given by a book from the former GDR [3].

2.2.2 Mechanical Model of the Vehicle. Connecting Elements

The basic model of a passenger coach consists of carbody, two bogie frames, and four rigid wheelsets. Between carbody and bogie frame, the secondary suspension is installed, and between the bogie frame and the wheelsets is the primary suspension. Many design variants of bogie vehicles are traceable to the basic model shown in Fig. 2.7. Figure 2.8 shows typical suspension elements that are available for modeling in multibody simulation tools such as SIMPACK, MEDYNA, and GENSYS.

For high-speed vehicles with four axles, however, there are some additional elastic eigenmodes to consider in order to describe the comfort level correctly up to 25 Hz; cf. Fig. 2.9.

2.2.3 Elastic Carbodies

As already mentioned, it is in general not sufficient for carbodies to consider only the six degrees of freedom of the rigid body.

That is especially true for lightweight carbody designs. Finite-element models of this kind of carbody are relatively easy to realize with today's technology. The inclusion of an entire FE model in a model for studying vehicle dynamics is hardy possible due to the enormous demands on calculation time and memory access. According to ISO [4], however, the frequency range up to 80 Hz should be included. There are two possibilities:

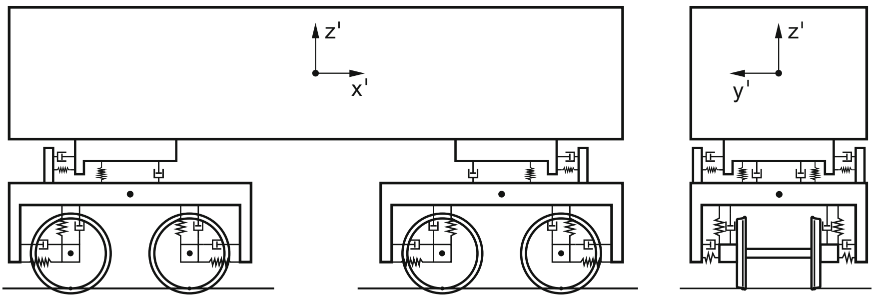


Fig. 2.7 Simple mechanical model of four-axle vehicle. Six carbody degrees of freedom, six degrees of freedom each for the bogie frames (The origin of the carbody and bogie coordinate systems are at the height of the *top* of the rails.)

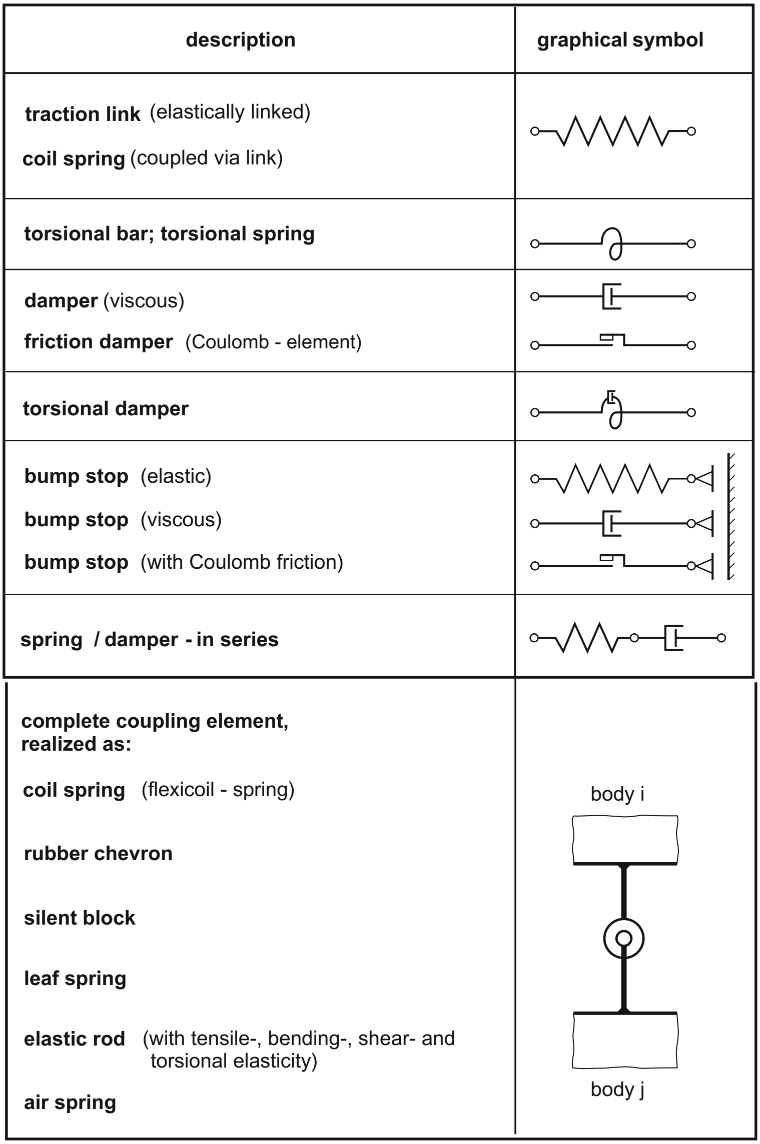


Fig. 2.8 Simple coupling elements for multibody algorithms (*top*) and complex element (*bottom*)

1. The carbody can be approximated in a model as elastic beam with bending and shear resistance as well as torsional stiffness, as done by [5]. The generalized masses and stiffnesses [6] have to be chosen in such a way that the lower vertical and lateral bending eigenfrequencies as well as the torsional eigenfrequency of

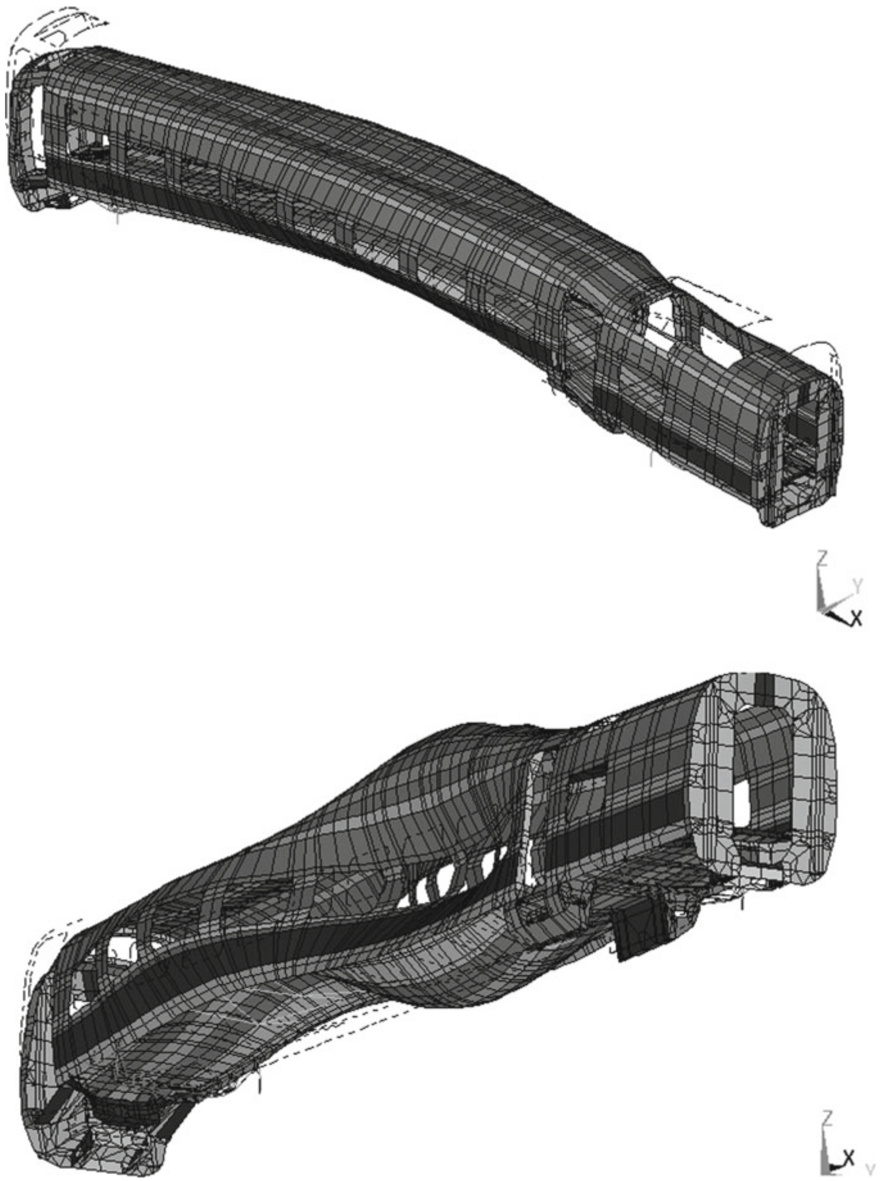


Fig. 2.9 Eignenmodes of a lightweight carbody for ICE (courtesy of DUEWAG, Krefeld Uerdingen). Bending coupled with cross-section deformation at 10 Hz (*top*), dominant cross section deformation at 13 Hz (*bottom*)

the free beam are in accordance with values that can be obtained from a standing sinusoidal test during which the bogie frame is softly supported.

2. In dealing with a lightweight carbody, this is not a viable approach, since the lower eigenmodes no longer consist of pure bending eigenmodes but already contain heavy local deformations, as can be seen in Fig. 2.9. In these cases, the eigenvalues and eigenmodes must be calculated from an FE model. For dynamic calculations, only the lowest eigenvalues and eigenmodes are considered (modal reduction [6]). In practice, this is a bit more complex, since the carbody is connected to the bogie through the secondary suspension, and the forces that are introduced through the secondary suspension into the carbody influence the vibrations. Details about this can be found, e.g., in [7]. In large software systems this is already included.

2.3 Modeling of Track and Excitation

2.3.1 Track Modeling

In vehicle dynamics investigations, most of the time the track is regarded as rigid and fixed. If it is desired to include in the simulations the forces between wheel and rail up to frequencies of 200 Hz, that is, forces due to, for example, wheel flats, then simple track models exist. In Fig. 2.10, such a model is inserted under the wheelset.

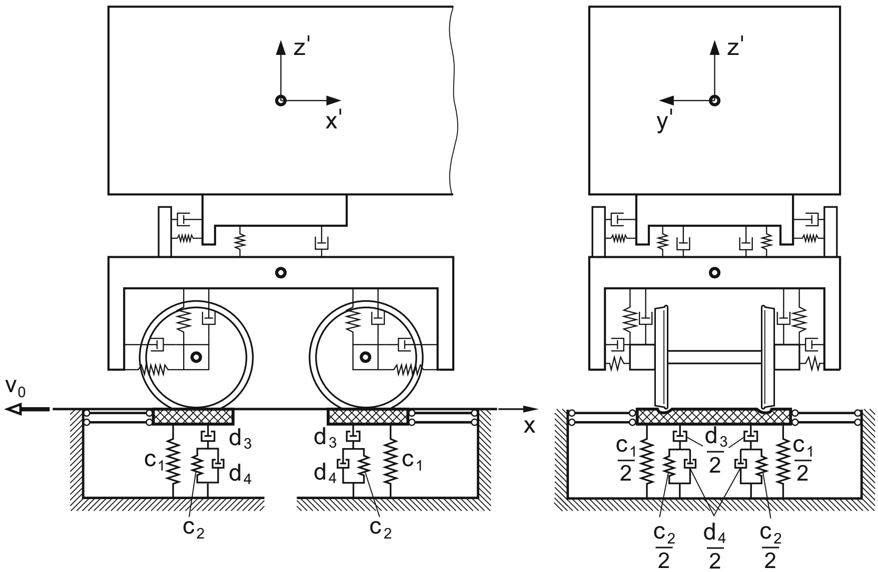


Fig. 2.10 Track modeling for frequencies up to 100 Hz

The basic elements that are inserted in the vertical direction consist of a spring, and in parallel, a damper and a Voigt–Kelvin element (parallel connection of spring and damper).

In order to capture frequencies up to 300 Hz accurately, several of the basic elements have to be connected in consecutive order [8]. The problem in this context is to determine the parameters, which is possible only with track dynamic methods [8–10]. In Table 2.1, the parameters for the vertical model for three different types of track from the DB-network ((1) track with B70 sleepers in an overhauled line, (2) newly built track with B70 sleepers, and (3) test track on the southern bypass of Stendal with B75 sleepers) are compiled. Negative values for the stiffnesses are no problem, since all models have only eigenvalues with negative real parts. In principle, it is possible to formulate analogous models for the lateral and sway motions.

Table 2.1 Vertical rail model consisting of three, respectively four, basic elements arranged serially. If the basic models are to be placed beneath every wheel contact point, the values have to be divided in half. Figure 2.10 shows only one of these basic models

		Overhauled line medium subsoil	New line hard subsoil	Southern bypass Stendal hard subsoil
		Rail UIC 60 Sleeper pad Zw700 Sleeper B70 Distance between sleepers 0.6 m Ballast thickness 0.3 m Subsoil $c_s = 150\ \frac{m}{s}$	Rail UIC 60 Sleeper pad Zw700 Sleeper B70 Distance between sleepers 0.6 m Ballast thickness 0.3 m Subsoil $c_s = 300\ \frac{m}{s}$	Rail UIC 60 Sleeper pad Zwp 104 Sleeper B75 Distance between sleepers 0.63 m Ballast thickness 0.4 m Subsoil $c_s = 300\ \frac{m}{s}$
$i = 0$	c_0	+0.9968E+08	+0.2434E+09	+0.1888E+09
	d_0	+0.4046E+06	+0.3824E+06	+0.2359E+06
$i = 1$	c_1	+0.4487E+09	+0.6395E+09	+0.1756E+09
	c_2	−0.4314E+07	−0.5351E+07	−0.1697E+08
	d_3	+0.1962E+05	−0.2655E+05	+0.4667E+05
	d_4	−0.2000E+05	+0.2594E+05	−0.4680E+05
$i = 2$	c_1	−0.1165E+11	−0.8016E+10	+0.5296E+09
	c_2	+0.2705E+09	+0.5398E+10	−0.1032E+09
	d_3	+0.8651E+06	+0.9640E+07	+0.3032E+06
	d_4	−0.8387E+06	−0.2222E+07	−0.2738E+06
$i = 3$	c_1	+0.2400E+10	+0.5959E+09	
	c_2	−0.9275E+09	−0.1864E+07	
	d_3	+0.1897E+07	+0.3582E+05	
	d_4	−0.1309E+07	−0.3643E+05	

2.3.2 Modeling of the Excitation

The excitation of rail vehicles is mainly the result of track irregularities and out-of-round wheels, which act as excitation in the contact of wheel and rail. Furthermore, there are excitations from out-of-balance wheels or from aerodynamic forces. These excitations can be divided into

- periodical excitations,
- general, deterministic excitations, and
- stochastic excitations.

The excitations resulting from *out-of-round wheelsets* are always *periodic*, since they will be repeated after one revolution. The basic harmonic frequency is

$$f_1 = \frac{v_0}{2\pi r}\,, \tag{2.1}$$

which with a wheel circumference of $2\pi r \approx 3$ m and a velocity of v_0= 60 m/s still falls within the frequency range that is relevant for the vertical dynamics. The higher harmonic frequency, e.g., by polygonization, which occurs in the ICE, results in very uncomfortable medium frequency noise and vibrations ("buzzing" at 100 Hz), which can severely damage track and wheelset; see, e.g., [10–13].

Tracks are never installed perfectly; there are always track irregularities that will become worse during service. Tracks can be described by four types of *track irregularities* (cf. Fig. 2.11). The track gauge fault is relevant for vehicle dynamics only in nonlinear considerations. Therefore, we can concentrate on the first three irregularities, which are all shown in Fig. 2.12.

Vertical errors in the position and level of the track can, like out-of-round wheel defects, be periodic, for example if the rail is not continuously welded. The track irregularity has to be determined as the lowering of the rail under static load, since that is exactly the disturbance that the wheel will experience. At the rail joints, a kink or even a jump will occur. Hanging sleepers also often occur periodically, which results in near-periodic track irregularities. Also, production processes (e.g., production of slab track) can yield periodic track irregularities.

Much more common are *deterministic singular irregularities*, which in most cases result from irregularities in the track, such as passing over switches and crossings, difference in stiffness when driving onto a bridge, passing over level crossings, or several hanging sleepers, which act upon the wheelset as a height error even if no height difference can be measured in the track (e.g., with laser measurement).

The most common, the *random (stochastic) track irregularities*, will be described in Chap. 6 as spectral power densities (Fig. 2.13).

In connection with the track model introduced in Fig. 2.10, a modified excitation model is usually introduced (Fig. 2.14). In doing so, the wheelset will not pass over a rail with a profile irregularity; rather, a band of disturbances without mass, with velocity v_0, is pulled through between wheelset and rail. The carbody (for bogie vehicles) or bogie frame (for a single bogie) is fixed in the rolling direction. The

- lateral alignment

$$y_{\mathrm{G}} = \frac{1}{2}(y_1 + y_2)\,,$$

- vertical alignment

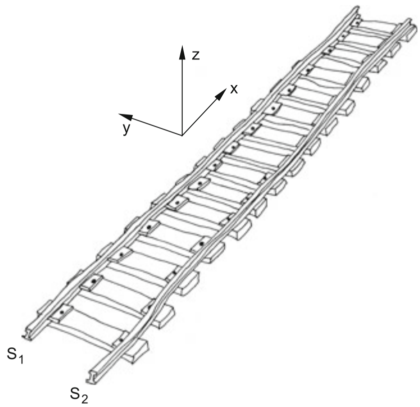
$$z_{\mathrm{G}} = \frac{1}{2}(z_1 + z_2)\,,$$

- cross level

$$\varphi_{\mathrm{xG}} = \frac{1}{2b}(z_1 - z_2)\,,$$

- and gauge

$$\Delta y_{\mathrm{G}} = \frac{1}{2}(y_1 - y_2)\,.$$



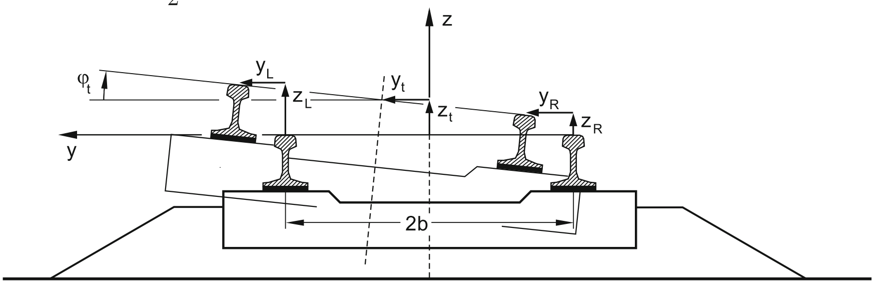


Fig. 2.11 Track irregularities (as in Renger [14])

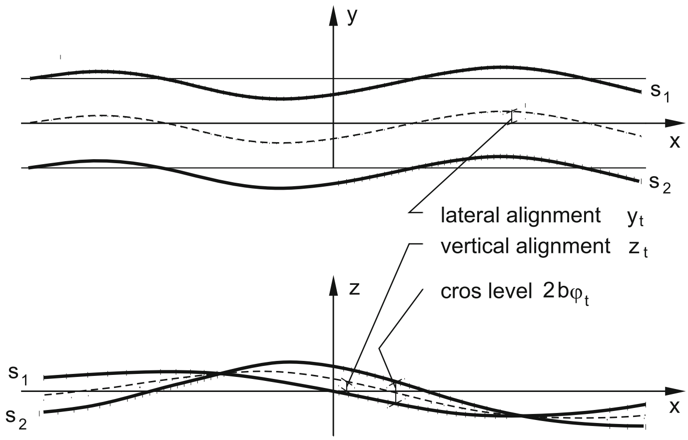


Fig. 2.12 Sketch of track irregularities

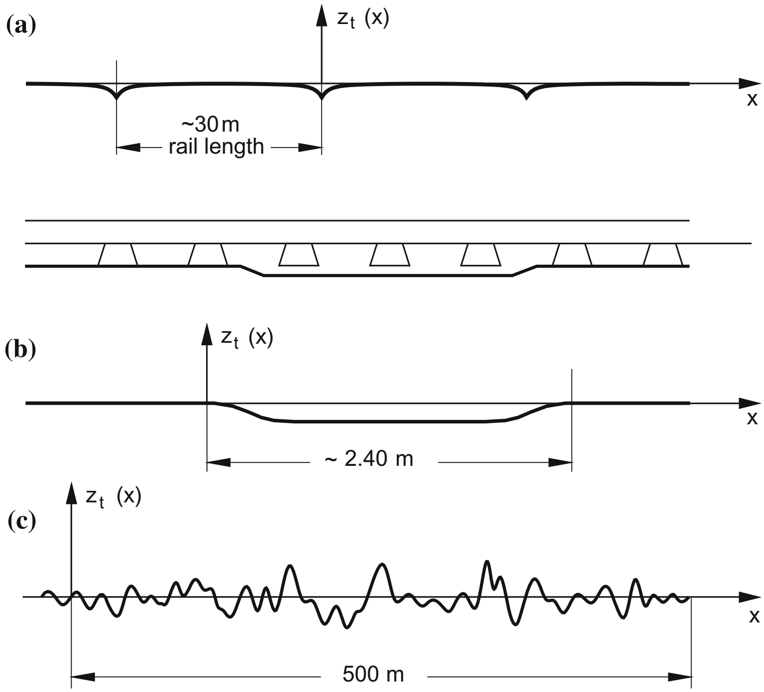


Fig. 2.13 Schematic examples of **a** periodic track irregularity (rail joints), **b** deterministic single irregularity (hanging sleepers), and **c** stochastic vertical alignment

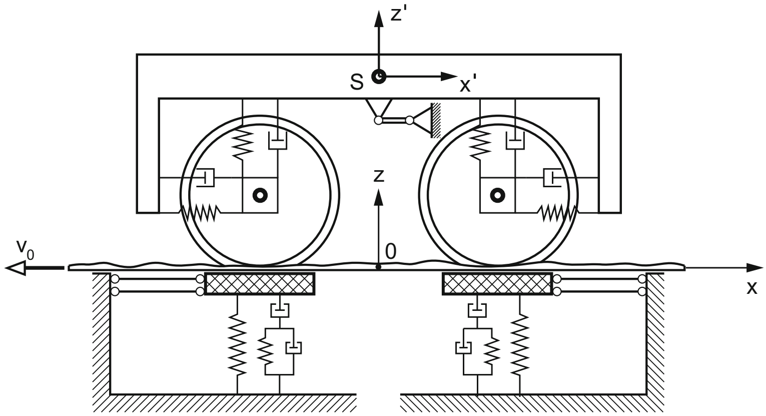


Fig. 2.14 Model of a moving irregularity (For the sake of clarity, the origin of the track following an (x', z')-coordinate system is situated in the bogie frame and not, as is usual, at the height of *top* of the rail.)

results from models with moving wheelset as compared to moving profile irregularity differ only when one is working with continuous track models for high vehicle speed or high excitation frequencies [10].

For the excitation model of a moving profile irregularity, the $(0; x, y, z)$- coordinate system moves with constant velocity v_0 against the rolling direction, while the coordinate system (S, x', y', z') is fixed. Special attention is required when acceleration comes into play (acceleration, braking, curving). In those cases, additional forces of inertia have to be considered.

The excitation of a band of disturbance values is sufficient for most cases of vehicle dynamics investigations. Only when one is simulating passing over bridges does the model reach its limit [15, 16].

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