3 The Anatomy of Railway Vehicle Running Gear

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CONTENTS

I. MAIN FUNCTIONS OF THE RUNNING GEAR AND TERMINOLOGY

The principal difference between a railway vehicle and other types of wheeled transport is the guidance provided by the track. The surface of the rails not only supports the wheels, but also guides them in a lateral direction. The rails and the switches change the rolling direction of wheels and thus determine the travelling direction of the railway vehicle.

The running gear is the system that provides safe motion of the vehicle along railway track. The running gear includes such components as wheelsets with axleboxes, the elastic suspension, the brakes, the traction drive, and the device to transmit traction and braking forces to the car body. Its main functions are:

- Transmission and equalization of the vertical load from the wheels of the vehicle to the rails
- Guidance of vehicle along the track
- Control of the dynamic forces due to motion over track irregularities, in curves, switches and after impacts between the cars
- Efficient damping of excited oscillations
- Application of traction and braking forces

Depending on the running gear, the vehicles may be described as bogied or bogie-less.

In vehicles without bogies the suspension, brakes, and traction equipment are mounted on the car body frame. The traction and braking forces are transmitted through traction rods or axlebox guides (sometimes known as “horn guides”). Conventional two-axle vehicles will generate larger forces in tight curves than the equivalent bogie vehicle; therefore their length is limited.

Running gear mounted on a separate frame that can turn relative to the vehicle body is known as a bogie (or truck). The number of wheelsets that they unite classifies the bogies. The most common type is the two-axle bogie, but three- and four-axle bogies are also encountered, often on locomotives.

Previously, the bogies simply allowed the running gear to turn in a horizontal plane relative to the car body thus making it possible for the wheelsets to have smaller angles of attack in curves. In modern bogies, the bogie frame transmits all the longitudinal, lateral, and vertical forces between the car body and the wheelsets. The frame also carries braking equipment, traction drive, suspension, and dampers. It may also house tilting devices, lubrication devices for wheel-rail contact and mechanisms to provide radial positioning of wheelsets in curves. Bogied vehicles are normally heavier than two-axle vehicles. However, the design of railway vehicles with bogies is often simpler than for two-axle vehicles and this may provide reliability and maintenance benefits.

II. BOGIE COMPONENTS

A. Wheelsets

A wheelset comprises two wheels rigidly connected by a common axle. The wheelset is supported on bearings mounted on the axle journals.

The wheelset provides:

- The necessary distance between the vehicle and the track
- The guidance that determines the motion within the rail gauge, including at curves and switches
- The means of transmitting traction and braking forces to the rails to accelerate and decelerate the vehicle

The design of the wheelset depends on:

- The type of the vehicle (traction or trailing)
- The type of braking system used (shoe brake, brake disc on the axle, or brake disc on the wheel)
- The construction of the wheel centre and the position of bearings on the axle (inside or outside)
- The desire to limit higher frequency forces by using resilient elements between the wheel centre and the tyre

The main types of wheelset design are shown in Figure 3.1. Despite the variety of designs, all these wheelsets have two common features: the rigid connection between the wheels through the axle and the cross-sectional profile of the wheel rolling surface, named wheel profile.

In curves, the outer rail will be a larger radius than the inner rail. This means that a cylindrical wheel has to travel further on the outer rail than on the inner rail. As the wheels moving on the inner and outer rails must have the same number of rotations per time unit such motion cannot occur by pure rolling. To make the distances travelled by two wheels equal, one or both of them will therefore "slip" thus increasing the rolling resistance and causing wear of wheels and rails. The solution is to machine the rolling surface of wheels to a conical profile with variable inclination angle γ to the axis of the wheelset (Figure 3.2). The position of the contact point when the wheelset

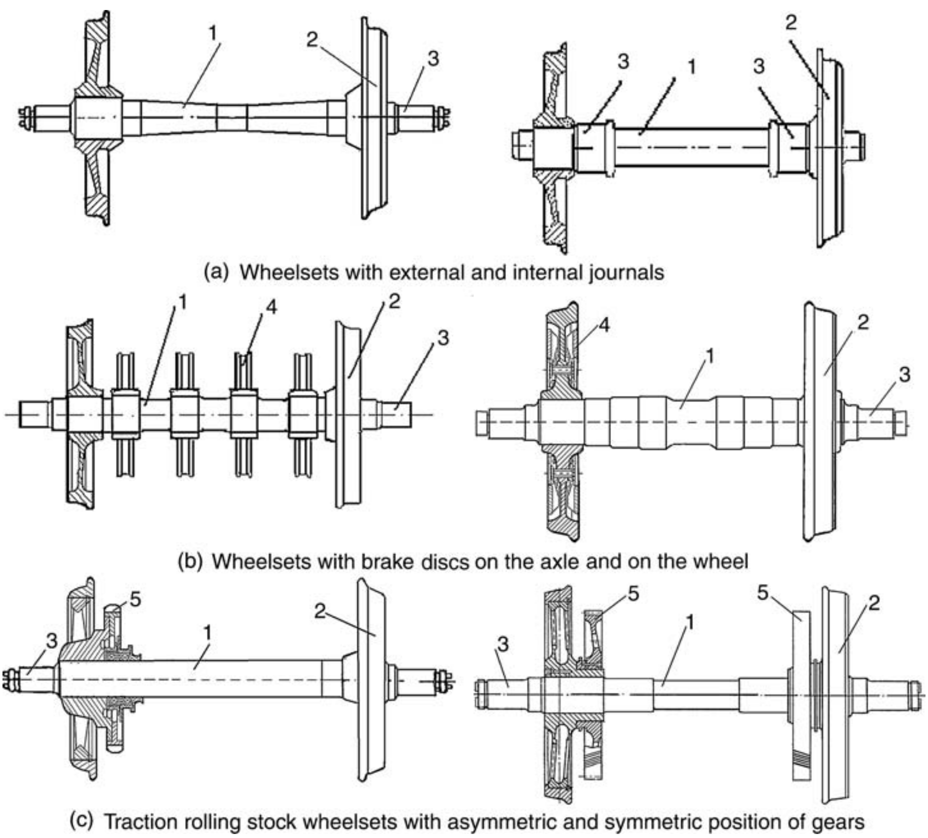


FIGURE 3.1 Main types of wheelset design: (a) with external and internal journals; (b) with brake discs on the axle and on the wheel; (c) with asymmetric and symmetric position of gears (1, axle; 2, wheel; 3, journal; 4, brake disc; 5, tooth gear).

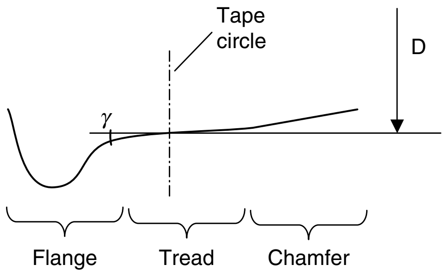


FIGURE 3.2 Main elements of a wheel profile.

is at a central position on the rails determines the so-called "tape circle," where the diameter of the wheel is measured. On the inner side of the wheel, the conical profile has a flange which prevents derailment and guides the vehicle once the available creep forces have been exhausted.

An unrestrained wheelset with conical profiles will move laterally in a curve such that the outer wheel is rolling on a larger radius (due to the cone angle) than the inner one. It can be seen that for each curve radius only one value of conicity exists that eliminates slip. As different railways have varying populations of curve radii the shape of wheel profile that provides minimum slip depends on the features of track. Railway administrations normally specify allowable wheel profiles for their infrastructure and the degree of wear permitted before reprofiling is required.

Figure 3.3 shows several examples of new wheel profiles. For understanding the dynamic behaviour of a railway vehicle the conicity of interface is critical. Conicity is defined as the difference in rolling radii between the wheels for a given lateral shift of the wheelset.

Despite the variety of wheel profiles, they have a number of common features. The width of the profile is typically 125–135 mm and flange height for vehicles is typically 28–30 mm. The flange inclination angle is normally between 65 and 70°. In vicinity of the tape circle the conicity is 1:10 or 1:20 for common rolling stock. For high speed rolling stock, the conicity is reduced to around 1:40 or 1:50 to prevent hunting. It can be seen from Figure 3.3 that the wheel profile has a relief toward the outer side of the wheel. This is intended to lift the outer side of the wheel off the rail and thus ease the motion on switches. Some modern wheel profiles, particularly for passenger rolling stock are not conical but designed instead from a series of radii that approximate a part-worn shape. This is intended to give a more stable shape and prevent the significant changes in conicity that may occur as a conical wheel profile wears. An example of such a profile is the UK P8 wheel profile.

For profiles whose shape is not purely conical (either by design or through wear in service), the term equivalent conicity is applied. This is the ratio of the rolling radius difference to twice the lateral displacement of the wheelset:

$$\gamma_{\mathrm{eq}} = \frac{\Delta R}{2y} \tag{3.1}$$

It is important to note that the rolling radius difference is a function of both the wheel and rail shape and hence a wheel profile on its own cannot be described as having an equivalent conicity.

As the wheel wears, the shape of the profile may alter significantly depending upon a large number of factors. These may include the curvature profile of the route, the suspension design, the level of traction and braking forces applied, the average rail profile shape encountered and the lubrication regime. Tread wear (Figure 3.4) will increase the height of the flange and eventually cause it to strike fishplate bolts, etc. If the tread wear causes the profile to become excessively concave damaging stresses may arise at the outer side of the wheel and rail known as false flange damage. Flange wear may lead to increase of the flange angle and reduction of the flange thickness.

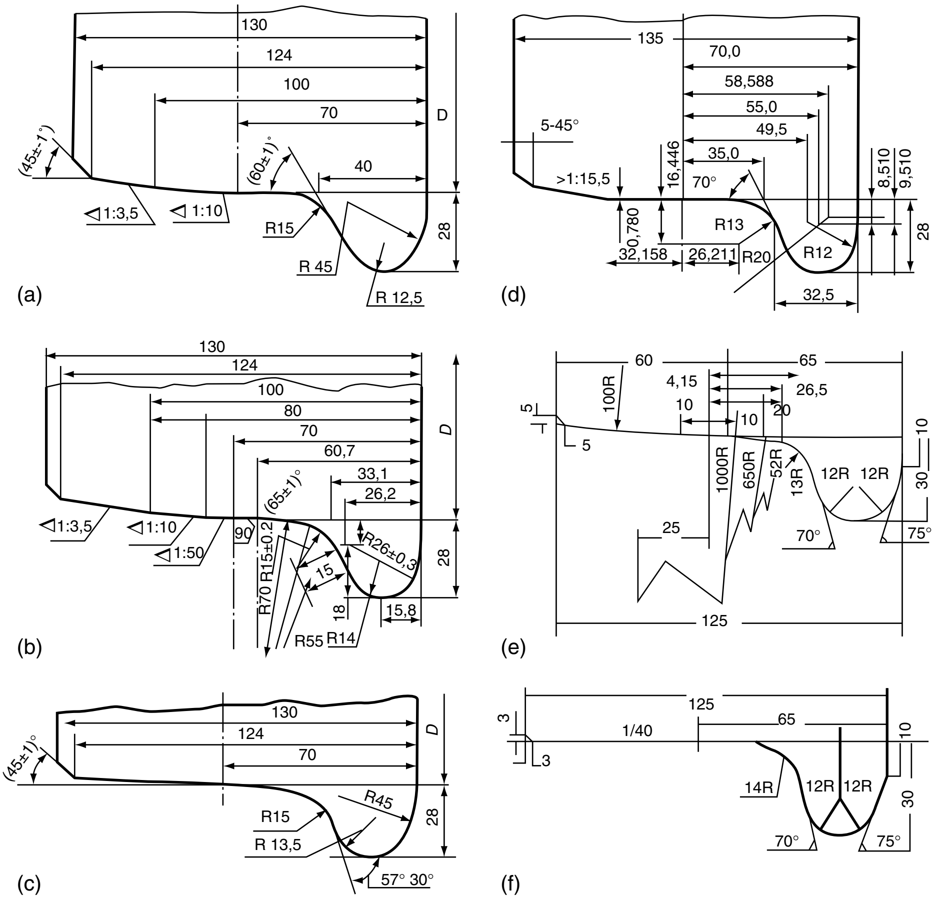


FIGURE 3.3 Common wheel profiles: (a) for freight and passenger railcars (Russia); (b) for high-speed railcars (Russia); (c) for industrial rolling stock (Russia); (d) for European freight and passenger railcars; (e,f) for high-speed trains (Japan).

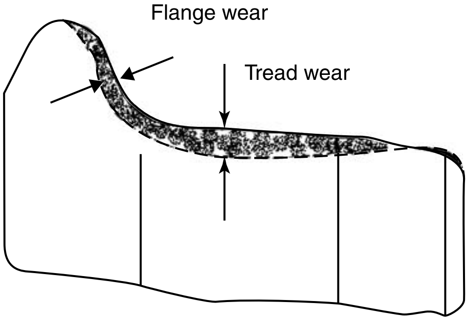


FIGURE 3.4 Tread and flange wear.

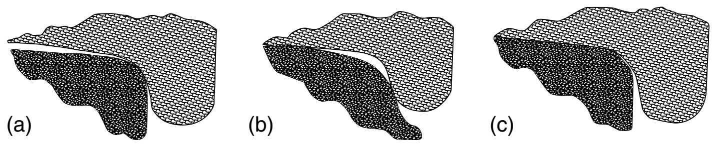


FIGURE 3.5 Possible contact situations between the wheel and the rail: (a) single-point contact; (b) two-point contact; (c) conformal contact.

In extreme conditions, this could increase the risk of switch-splitting derailments. Wheel profiles are generally restored to their design shape by periodic turning on a wheel lathe. This can normally be carried out without the necessity to remove the wheelset from the vehicle.

It is clear that contact conditions will vary considerably depending upon the shape of the wheel and rail profiles. This may take the form of single-point, two-point, or conformal contact as shown in Figure 3.5. One-point contact (a) develops between the conical or tread worn wheel profiles and rounded rail profile. Wheels wear quickly towards the local rail shape. With two-point contact (b) the wheel additionally touches the rail with its flange. In this case, the rolling contact has two different radii which causes intensive slip and fast flange wear. Conformal contact (c) appears when the wheel profile and the gauge side of the railhead wear to the extent that their radii in vicinity of the contact patch become very similar.

B. Axleboxes

The axlebox is the device that allows the wheelset to rotate by providing the bearing housing and also the mountings for the primary suspension to attach the wheelset to the bogie or vehicle frame. The axlebox transmits longitudinal, lateral, and vertical forces from the wheelset on to the other bogie elements. Axleboxes are classified according to:

- Their position on the axle depending on whether the journals are outside or inside
- The bearing type used, either roller or plain bearings

The external shape of the axlebox is determined by the method of connection between the axlebox and the bogie frame and aims to achieve uniform distribution of forces on the bearing. Internal construction of the axlebox is determined by the bearing and its sealing method.

Axleboxes with plain bearing (Figure 3.6) consist of the housing (1), the bearing itself (2) which is usually made of alloy with low friction coefficient (e.g., bronze or white metal), the bearing shell

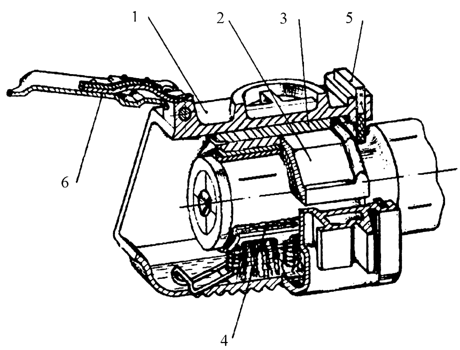


FIGURE 3.6 Construction of an axlebox with friction bearing.

(3) which transmits the forces from the axlebox housing to the bearing, a lubrication device (4) which lubricates the axle journal. Front and rear seals (5 and 6) prevent dirt and foreign bodies entering the axlebox, while the front seal (6) can be removed to monitor the condition of the bearing and add lubricant.

Vertical and longitudinal forces are transmitted through the internal surface of the bearing and lateral forces by its faces.

Plain bearing axleboxes are now largely obsolete as they have several serious disadvantages:

- High friction coefficient when starting from rest
- Poor reliability
- Labour-intensive maintenance
- Environmental pollution

However, from a vehicle dynamic behaviour point of view, axleboxes with plain bearings had certain positive features. In recent years, plain bearing axleboxes that do not require lubrication have been reintroduced on certain types of rolling stock though their use is still rare.

Axleboxes with roller type bearings (Figure 3.7) are classified according to:

- The bearing type (cylindrical, conical, spherical)
- The fitting method (press-fit, shrink-fit, bushing-fit)

The main factor that determines the construction of the axlebox is the way it experiences the axial forces and distributes the load between the rollers.

Cylindrical roller bearings have high dynamic capacity in the radial direction, but do not transmit axial forces (Figure 3.7a). Experience in operation of railway rolling stock showed that the faces of rollers can resist lateral forces. However, to do this successfully it is necessary to regulate not only the diameter, but also the length of rollers, and the radial, and axial clearances.

Conical bearings (Figure 3.7b and c) transmit axial forces through the cylindrical surface due to its inclination to the rotation axis. This makes it necessary to keep the tolerances on roller diameters and clearances almost an order of magnitude tighter than for cylindrical bearings. In addition, conical bearings have higher friction coefficients compared to the radial roller bearings and therefore generate more heat. This not only increases traction consumption, but also creates difficulties for diagnostics of axlebox units during motion.

Recently cartridge-type bearings have been widely used. Their special feature is that the bearing is not disassembled for fitting, but is installed as one piece.

Spherical bearings have not been widely applied due to their high cost and lower weight capacity, although they have a significant advantage providing better distribution of load between

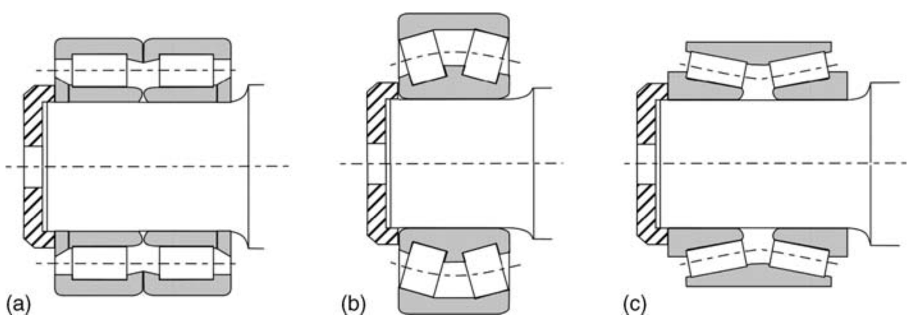


FIGURE 3.7 Constructions of roller bearings: (a) cylindrical double-row; (b) one-row self-alignment; (c) two-row conical.

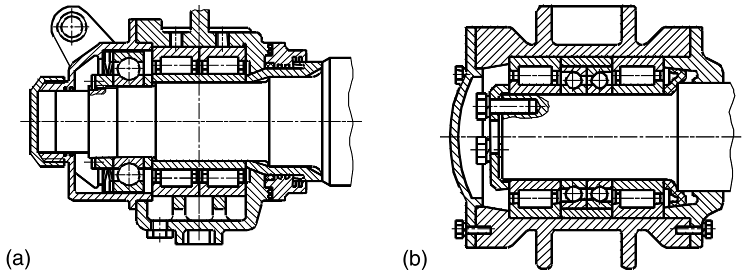


FIGURE 3.8 Use of spherical bearings: (a) triple bearing of Japanese high-speed trains; (b) triple bearing of French high-speed trains.

the front and rear rows in case of axle bending. Ball bearings are, however, often combined with cylindrical bearings in railway applications to transmit axial forces.

High speed rolling stock often has three bearings in the axlebox: two transmitting radial forces and one (often a ball bearing) working axially (Figure 3.8).

C. Wheels

Wheels and axles are the most critical parts of the railway rolling stock. Mechanical failure or exceedance of design dimensions can cause derailment. Wheels are classified into solid, tyre, and assembly types as shown in Figure 3.9.

Solid wheels (Figure 3.9a) have three major elements: the tyre, the disc, and the hub, and mainly differ in the shape of the disc.

Tyred wheels (Figure 3.9b) have a tyre fitted to the wheel disc that can be removed and replaced when it reaches its maximum turning limit.

Wheels may have straight, conical, S-shaped, spoked, or corrugated type discs when viewed in cross-section. A straight disc reduces the weight of the construction and can be shaped such that the metal thickness corresponds to the level of local stress. The conical and S-shape discs serve to increase the flexibility of the wheel, therefore reducing the interaction forces between the wheels and the rails. Corrugated discs have better resistance to lateral bending.

The desire of reducing wheel-rail interaction forces by reducing the unsprung mass has led to development of resilient wheels (Figure 3.9c) that incorporate a layer of material with low elasticity modulus (rubber, polyurethane). These help to attenuate the higher frequency forces acting at the wheel-rail interface.

Improved bearing reliability aroused interest in independently rotating wheels which provide significant reductions in unsprung mass due to the elimination of the axle. By decoupling the wheels, the independently rotating wheelset inevitably eliminates the majority of wheelset guidance forces. Such wheelsets have found application either on variable gauge rolling stock providing fast transition from one gauge width to another or on urban rail transport where low floor level is necessary.

D. Suspension

The suspension is the set of elastic elements, dampers and associated components which connect wheelsets to the car body.

If the bogie has a rigid frame, the suspension usually consists of two stages: primary suspension connecting the wheelsets to the bogie frame and secondary suspension between the bogie frame and the bolster or car body. Such bogies are termed double suspended. Sometimes, typically in freight

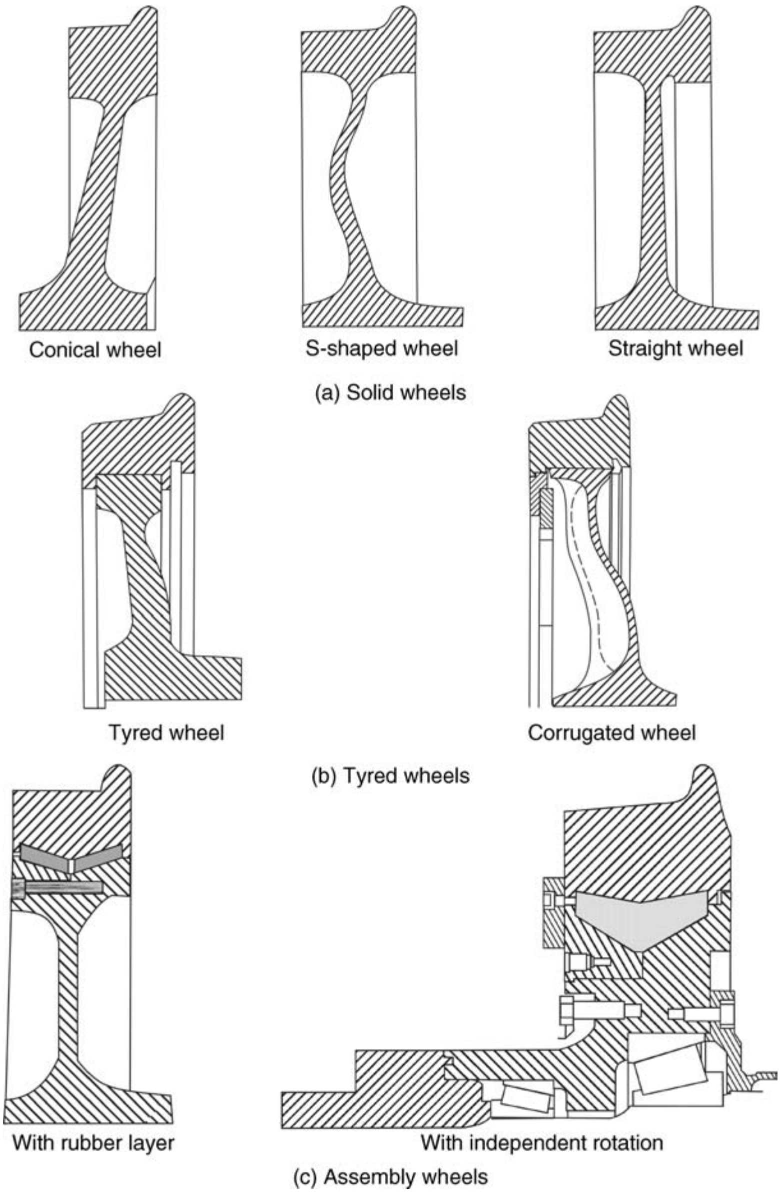


FIGURE 3.9 Major types of railway wheels.

bogies, only a single-stage suspension is used. Where this occupies the primary suspension position it is often termed "axlebox suspension." In the secondary suspension position it may be termed "central suspension."

E. Elastic Elements (Springs)

Elastic elements (springs) are components which return to their original dimensions when forces causing them to deflect are removed. Elastic elements are used to:

- Equalise the vertical loads between wheels (unloading of any wheel is dangerous because it causes a reduction/loss of guidance forces)

- Stabilise the motion of vehicles on track (self-excited lateral oscillations, i.e., hunting of wheelsets is dangerous)
- Reduce the dynamic forces and accelerations due to track irregularities

The capability of elastic elements to provide the above functions is determined by their force characteristic, which is the dependence between the force acting on the elastic element, P, and its deflection z: $P = P(z)$. Force characteristics can be linear or non-linear. In linear characteristics, the deflection is proportional to the force. For non-linear characteristics the deflection rate increases (or less often for railway applications, decreases) with increase of the load. The principal types of elastic elements are shown below in Table 3.1.

A leaf spring (picture A in Table 3.1) is an elastic element comprising a number of steel leafs. Leafs work in bending and the "fish-bellied" shape of the beam provides smaller spring stiffness. Depending on their design, leaf springs can be closed (picture A in Table 3.1), elliptical, or open. They consist of layered leafs 1 and 2 having different length and held together by a buckle 3. The largest leaf (1) is named the master and the other leafs (2) the slaves. Leaf springs also provide damping due to the inter-leaf friction. However, it is difficult to obtain the specific desired damping values and the damping can change considerably due to lubrication or contamination of the rubbing surfaces.

A plate spring (or washer) (picture B in Table 3.1) consists of a set of elastic steel plates having the conical shape with inclination angle β. Under the load P the plates flatten and decrease the angle β, thus providing the spring's deflection. The stiffness of the plate spring depends on the number of plates and their relative arrangement (in series or in parallel).

A ring spring (picture C in Table 3.1) consists of external and internal steel rings that rest on each other with conical surfaces. Under the load P the external rings stretch and the internal rings shrink in radial direction, thus providing the vertical deflection of the spring. Deformation causes significant friction forces between the rings.

Coil springs are the most commonly used elastic elements which can either be cylindrical (picture D in Table 3.1) or conical. Usually they are produced of steel spring wire typically of circular cross-section. Coil springs are cheap and robust, but provide very little damping in suspension applications.

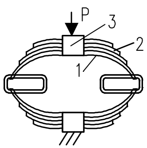
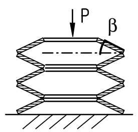
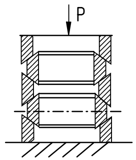
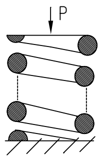
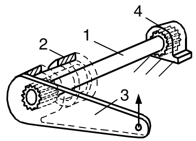
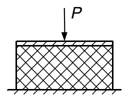
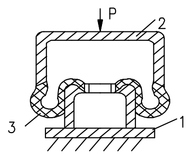
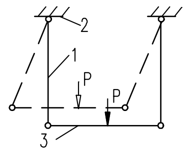
Torsion springs (picture E in Table 3.1) consist of the torsion bar 1, having its first end in the bearing 2 connected to the arm 3, the second end being fixed in the mounting 4. The force P causes elastic torsion of rod 1. The most common application of this type of spring in railway vehicles is the roll bar.

Rubber-metal springs (picture F in Table 3.1) consist of the rubber blocks 1 interleaved with or reinforced by steel plates 2. This type of spring is widely used in passenger rolling stock, particularly on primary suspensions as it allows damping of high frequency vibrations and reduction of maintenance costs due to the elimination of wearing friction components. Some types of rubber-metal springs are illustrated in Figure 3.10. The elastic properties of rubber can be exploited to make springs that can carry significant loads in both compression and shear (Figure 3.10b, d, and e).

Air spring (picture G in Table 3.1) consists of the mounting 1, and rubber-cord elastic chamber 3 filled with compressed gas (usually air). This type of elastic elements is characterised by its small mass, excellent noise and vibration isolation and ability to maintain a constant ride height for different vehicle load conditions. Such springs are found almost universally in the secondary suspension of modern passenger vehicles. Air springs are often arranged in series with a rubber or rubber-interleaved spring to provide some compliance in the suspension if the airspring becomes deflated.

The operation of a typical air suspension with pressure control to maintain constant ride height is shown in schematic form in Figure 3.11.

TABLE 3.1
Principal Types of Elastic Elements

Features	Schematic of Elastic Elements: A	B	C	D	E	F	G	H
								
Application	Attenuation of vibrations and impacts in one direction	Multidirectional attenuation of vibrations	Multidirectional attenuation of vibrations	Multidirectional attenuation of vibrations and impacts	Attenuation of vibrations and impacts in one direction	Multidirectional attenuation of vibrations and impacts	Multidirectional attenuation of vibrations and impacts, various characteristics are possible	Attenuation of vibrations and impacts in one direction
Advantages	Stiffness of elastic and damping properties in a single component	Good vibration and noise insulation	Good vibration and noise insulation	Easy production and maintenance, small mass and dimensions	Possible to obtain large deflections, small mass and dimensions	Damping of high frequency vibrations, stiffness and damping in a single component	Ability to vary stiffness and maintain constant ride height, good noise and vibration insulation	Easy to maintain and repair
Disadvantages	Unpredictable damping	Complicated control of parameters, production, and maintenance	Complicated control of parameters, production, and maintenance	Inability to vary characteristics in operation	Complicated production, vertical displacements cause longitudinal displacements	Ageing of rubber, inability to vary characteristics in operation	Complicated air supply system and maintenance	Wear, large size

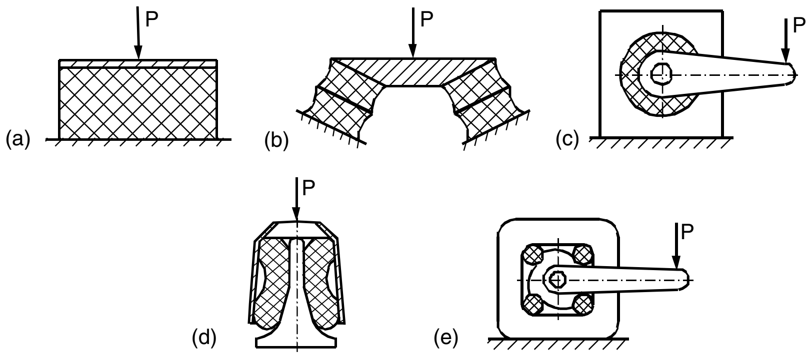


FIGURE 3.10 Rubber-metal springs: (a) compression; (b) compression and shear; (c) torsion; (d) bell type; (e) cam type.

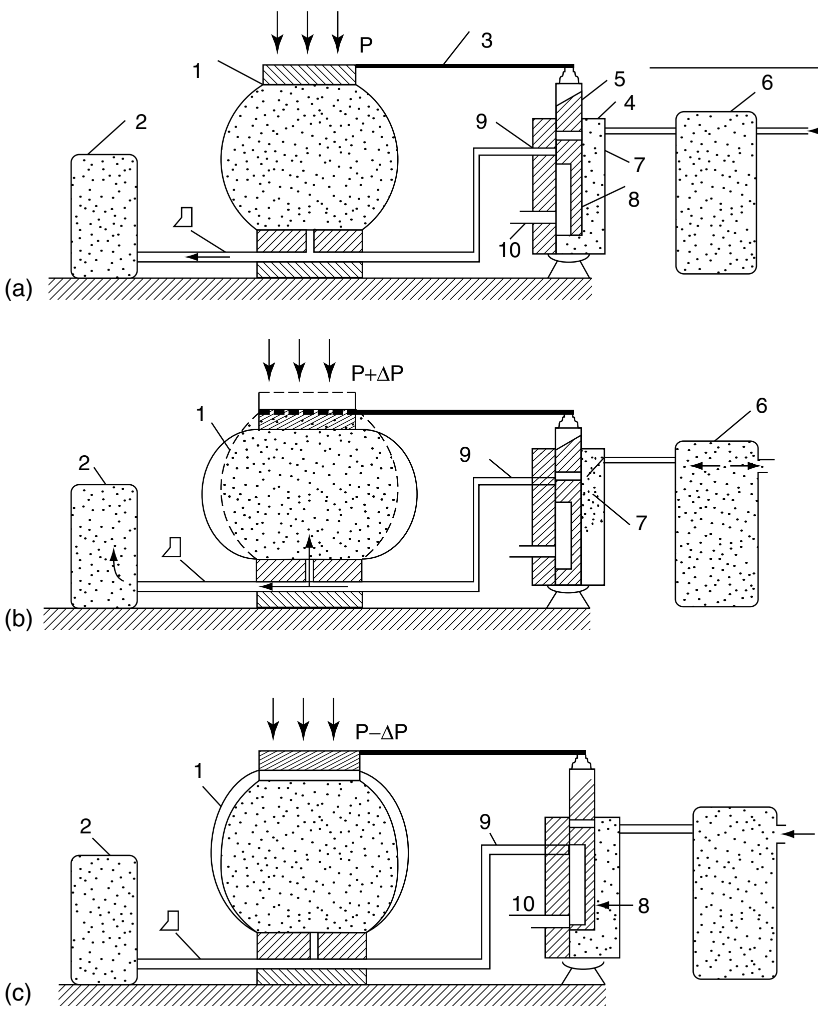


FIGURE 3.11 Schematic showing the operation of a typical air suspension: (a) Equilibrium position; (b) Upstroke; (c) Downstroke.

In position (a) above, the system is in static equilibrium when the pressure inside the elastic chamber (airbag) 1 provides the prescribed ride height P. To reduce the spring stiffness the elastic chamber is connected to the surge reservoir (additional volume) 2. When the load increases (position (b)) the airbag 1 is compressed and moves the valve 5 of control system 4 down. This causes the compressed air from the main reservoir 6 to be admitted to the airspring system through the pipe 9 and orifice 7, thus increasing the pressure. This restores the spring 1 to the equilibrium position (a) again and control valve 4 stops the flow of air from the main chamber 6 into the airbag. Reduction of the load (position (c)) makes the airbag rise and control valve 5 moves up. In this case pipe 9 connects to the atmosphere 10 through orifice 8 and drops the pressure in airbag 1. The spring height reduces and returns to the equilibrium position again.

The surge reservoir 2 and the damping orifice 7 are important features in the operation of the airspring. Increasing the surge reservoir volume leads to decreasing spring stiffness. Reducing the size of the damping orifice increases the damping properties of the spring (by increasing the kinetic energy dissipation), but also increases the stiffness. The lateral stiffness of the pneumatic spring depends on the shape of elastic chamber.

Railway vehicles often use devices whose stiffness is derived from gravitational forces, as for example in the swing link arrangement shown in picture H in Table 3.1. Rollers on inclined planes and various lever systems have also found applications in vehicle suspension arrangements. The swing link suspension is the most common application of those listed above. It consists of swing links 1 that are attached to mountings 2 and connected with a beam or spring plank 3. Swing link suspension effectively acts as a pendulum and is often used in secondary suspensions to give constant lateral frequency.

Typical force characteristics of elastic elements are shown in Table 3.2.

A piecewise linear characteristic (Table 3.2) is typical for coil springs arranged with clearance between springs working in parallel (picture B) or for springs with initial compression (picture C). Parabolic characteristics (picture D) are obtained from coil springs with variable step or wire diameter, conical springs, rubber, or pneumatic springs without pressure control system. An S-shaped force versus deflection characteristic is typical for combinations of elements with a jump between two working modes (picture E). An automatically controlled parabolic characteristic may be obtained from airsprings with automatic pressure control that is dependent on the vehicle loading (picture F). These may be combined with an elastic bump stop acting in compression.

The advantage of coil springs, rubber and pneumatic springs is that they are flexible not only in vertical, but also in longitudinal and lateral direction.

F. Dampers

Damping is usually provided in railway vehicle suspension by the use of viscous or friction damping devices.

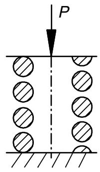
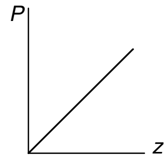
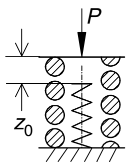
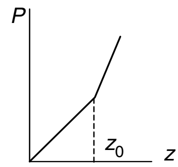
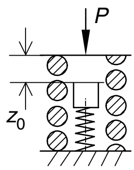
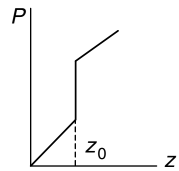
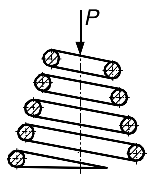
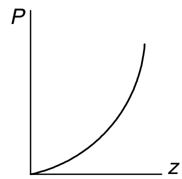
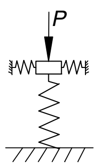
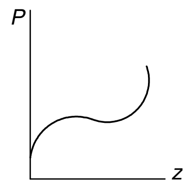
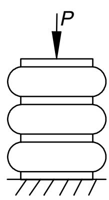
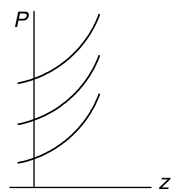
Dry friction results from the relative slip between two rigid bodies in contact. The friction force can be constant or dependent on the mass of the car body, but always acts to resist the relative motion. Friction force is proportional to friction coefficient μ, pressure between surfaces Q, and contact surface area S. This dependence can be represented by the following formula:

$$F_{\mathrm{dry\ fric}} = -\mu SQ\frac{\dot{z}}{|\dot{z}|} = -F_0\frac{\dot{z}}{|\dot{z}|} \tag{3.2}$$

where F_0 is the magnitude of friction force; $\dot{z}$ is the relative velocity of motion; $|\dot{z}|$ is the magnitude of velocity. The minus sign denotes that the friction force is always in the opposite direction to the velocity.

Viscous damping develops between two parts separated with a layer of viscous liquid (lubricant) or in devices known as hydraulic dampers, where the viscous liquid flows through an

TABLE 3.2
Typical Force Characteristics of Elastic Elements

Number	Designation of the Element	Scheme of the Element	Designation of Force Characteristic	View of Force vs. Deflection Characteristic
A	Cylindrical springs with equal height		Linear	
B	Cylindrical springs with different height		Bilinear	
C	Cylindrical springs with different height and initial compression of internal one		Trilinear with a jump	
D	Conical springs, rubber springs		Parabolic	
E	Elements with non-equilibrium characteristics		S-shaped	
F	Controlled pneumatic suspension		Controlled	

orifice and dissipates the energy. The damping force in viscous case is proportional to velocity:

$$F_{\text{hydr fric}} = -\beta \dot{z}^n \qquad (3.3)$$

where β is the coefficient; $\dot{z}$ is the velocity of relative motion; n is the power. Depending on the construction of the device and the liquid properties the power n can be greater, equal or less than 1.

If the liquid flow is laminar then $n \approx 1$ and damping is described linear viscous damping:

$$F_{\text{lin visc fric}} = -\beta_1 \dot{z} \tag{3.4}$$

where β_1 is the coefficient, named the damping coefficient for the hydraulic damper.

For $n = 2$ damping is called turbulent or quadratic:

$$F_{\text{turb visc fric}} = -\beta_2 |\dot{z}| \dot{z} \tag{3.5}$$

Gases are also viscous. Therefore, driving the gas through a throttle valve (damper orifice) may also produce sufficient force for damping the oscillations of railway vehicles.

Intermolecular damping (hysteresis) originates mainly in rubber and polyurethane elastic elements. In such cases, the damping force is proportional to oscillations velocity and is inverse to the frequency:

$$F_{\text{molec fric}} = -\frac{\beta_0}{\omega} \dot{z} \tag{3.6}$$

Damping of vibrations can also be obtained by other means such as the introduction of active dampers being controlled proportionally to velocity.

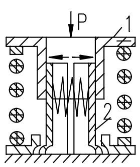
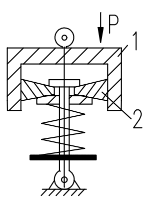
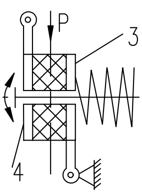
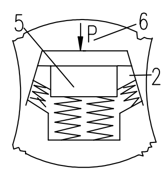
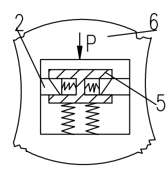
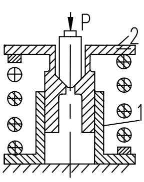
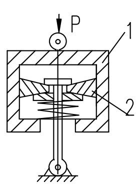
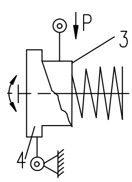
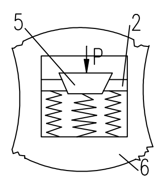
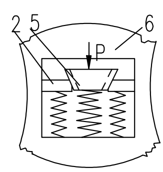
A damper is the device that controls oscillations in the primary or secondary suspension of the vehicle by energy dissipation.

Friction dampers are the devices that transform the energy of oscillations into the heat energy by dry friction. Friction dampers are mainly used in freight vehicle suspensions due to their low cost and simplicity.

Depending on their construction friction dampers may be classified as one of four types: integrated with the elastic element, integrated into the spring suspension, telescopic, and lever (Table 3.3).

Dampers integrated with an elastic element consist of the barrel 1 and friction wedges 2 that are held in contact by a spring. When the elastic element deforms, the friction forces act on the contacting surfaces between the barrel 1 and the wedges 2 transforming the kinetic energy into the heat.

TABLE 3.3
Classification of Friction Dampers

	Integrated with Elastic Element	Linear Action		Planar Action	Spatial Action
		Telescopic	Lever	Integrated in the Suspension	
Constant friction					
Variable friction					

Telescopic friction dampers consist of the body 1 that contains the piston with the system of friction wedges 2 clamped by a spring.

Dampers integrated in the suspension are mostly used in three-piece bogies and consist of friction wedges 2 that move relative to side frame 6 and bolster 5. Construction of the dampers (Table 3.3) differs by the position of friction wedges 2 (inside the bolster 5 or inside the frame 6), by the number of springs and their inclination angles, as well as by the design of the friction wedges. For example, the Russian CNII-H3 bogie has wedges 2 with inclined faces contacting with the bolster 5 and pressed to the side frame 6 by springs underneath.

Simultaneous and integrated friction dampers are connected to the springs in the suspension, whereas telescopic dampers are independent devices. Friction dampers may be arranged to produce either constant or variable friction force and can be designed to act in one (linear), two (planar), or three (spatial) directions.

Friction dampers integrated in elastic elements have found wide application in freight bogies in Russia, the USA, and many other countries, due to the following advantages: simplicity of design and fabrication, low cost, and easy maintenance. Disadvantages of such dampers include suboptimal damping in the partially laden condition, the difficulties of controlling friction to the desired design values and changes in friction levels as the faces wear or become contaminated in service.

Telescopic dampers have the advantage of being autonomous, protected from the environment (which reduces the likelihood of contamination of the friction surfaces), can be installed at angles other than vertically and hence can be used to damp vertical or horizontal vibrations of sprung elements of the vehicle. They can be inspected and repaired without lifting the car body. One of the reasons that such telescopic dampers are not widely used in freight vehicles using the popular three-piece bogie is that an integrated friction wedge as shown above is required to resist warping in vertical and horizontal planes.

In case of the bogies with a solid frame, friction dampers in the primary suspension must resist wheelset displacements. It is desirable that in primary suspensions the damper has an asymmetric characteristic providing lower damping forces in compression than extension. Hydraulic dampers are superior in this respect.

The main advantage of plane and spatial friction dampers is their ability to damp vibrations in several directions and in certain cases provide friction–elastic connections between parts of bogie frames. Such properties allow significant simplification of the bogies whilst retaining reasonable damping of complex vibrations. They are therefore widely used in freight bogies despite a number of disadvantages including providing unpredictable friction forces, and the fact that repair and adjustment of friction forces may require lifting the car body and disassembling the spring set.

Typical force characteristics for friction dampers are presented in Figure 3.12. Different designs of friction damper have varying arrangements for transmitting the normal force to the friction surfaces. Depending on the design, the damper may provide constant or variable friction. In the latter case, such damper is usually arranged such that a component of the force in one or more of the suspension springs is transmitted via a linkage or wedge to the friction faces.

Force characteristic (a) describes a constant friction damper, where the friction force does not depend on deformation of the spring set and is the same for compression and tension. The dashed line shows the characteristic of the same damper, where the friction pairs are elastically coupled. This can occur, for example, as a result of the friction surface having an elastic layer underneath. Force P first deforms the elastic pad and when the shear force equals the friction breakout force (i.e., μN), then relative displacement of friction pair occurs.

Characteristic (b) is common for most friction dampers used on freight bogies. The friction force depends on the deflection of the suspension, and is different for tension and compression.

Characteristics (c) and (d) are typical for multi-mode dampers, where the friction forces vary according to the given law and depend on the spring set deflection in tension or compression.

It can be seen that the variety of force characteristics available from friction dampers allows freight vehicle to be designed with suspensions providing satisfactory ride qualities.

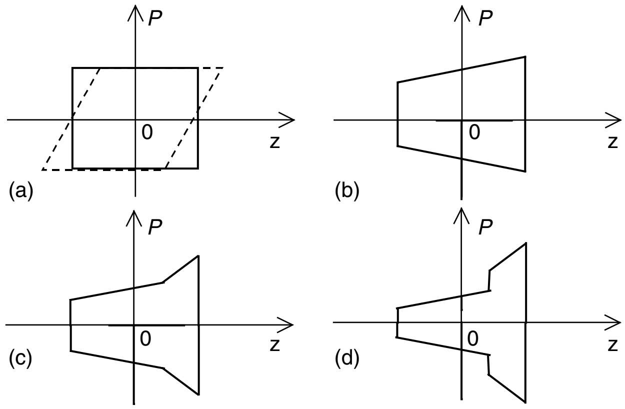


FIGURE 3.12 Typical force characteristics of friction dampers.

Hydraulic dampers are almost universally used in passenger bogies and are sometimes also used in modern freight bogies.

The energy dissipated in a hydraulic damper is proportional to velocity, and therefore to the amplitude and frequency of vibration. Thus the hydraulic damper is self-tuning to dynamic excitations and provides reliable and predictable damping of vehicle oscillations.

Railway vehicles use the telescopic hydraulic dampers as shown in Figure 3.13. The hydraulic damper operates by forcing the working fluid through an orifice (flow control valve) from one chamber into the other as the vehicle oscillates on the suspension. This produces viscous damping and the kinetic energy of the oscillations is transformed into heat.

Telescopic hydraulic dampers (Figure 3.13) consist of the body 1 with the sealing device, the working cylinder 2 with valves 4 and the shaft 3 with a piston 5 that also has valves 6. When the piston moves relative to the cylinder, the working fluid flows through the valves from the chamber over the piston to the chamber under it and back.

The reliability of hydraulic dampers mostly depends on the sealing between the shaft and the body. Occasionally malfunction of this unit causes excessive pressure in the chamber over the piston resulting in leakage of the working fluid. The capability of hydraulic damper to dissipate

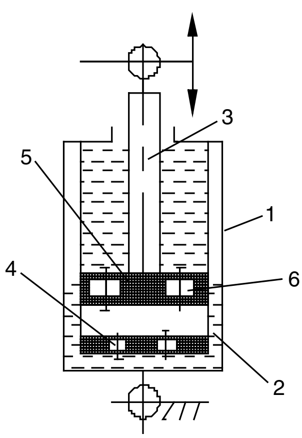


FIGURE 3.13 Telescopic hydraulic damper.

energy is characterised by its force versus velocity characteristic, which is the dependence between the resistance force developed in the hydraulic damper, P, and the piston displacement velocity $\dot{d}$.

The damper characteristic may be either symmetrical, when the resistance forces are the same for extension and compression, or asymmetric. Dampers with symmetric characteristics are typically used in secondary suspensions. In primary suspensions, asymmetric dampers are often used as the motion of the wheel over a convex irregularity causes larger forces than negotiating a concave one. As a result, dampers may be designed with an asymmetric characteristic providing a smaller force in compression than in extension. However, large damping forces in extension can significantly decrease the vertical wheel load, thus increasing the risk of derailment. Therefore the railway dampers are less asymmetric than the automobile ones.

Common force characteristics of hydraulic dampers are shown in Figure 3.14.

Figure 3.14A shows a hydraulic damper with a resistance force proportional to velocity and not exceeding the "blow-off" (saturation) force. When a predetermined pressure value is reached inside the working chamber, the "blow-off" valve opens to prevent excessive forces being developed by the damper.

Hydraulic dampers having characteristic B have a resistance force proportional to the velocity and the displacement. Such characteristic is obtained by the provision of specially calibrated needles (or other devices) into the flow control valve to change its cross-section. The size of the valve cross-section is controlled depending on static deflection of the suspension.

Scheme C is typical for devices where the dissipative force is proportional to velocity, but the operation of "blow-off" valve is controlled depending on the displacement and velocity of the piston. In scheme D, the size of the valve cross-section and the saturation limit for emergency valve are controlled together depending on the relative displacement and velocity of the piston.

Attachment of hydraulic dampers to the vehicle is usually done using the elastic mountings or bushes to prevent the transmission of high frequency vibrations. The internal pressure in the damper often gives it elastic properties. Therefore, hydraulic dampers are often modelled as a spring and viscous damper in series.

In some designs, the hydraulic dampers are united with the elastic elements. The schematic of a hydraulic damper integrated into coaxial rubber-metal spring is shown in Figure 3.15.

G. Constraints and Bumpstops

Constraints are the devices that limit the relative displacements of bogie units in longitudinal and lateral directions.

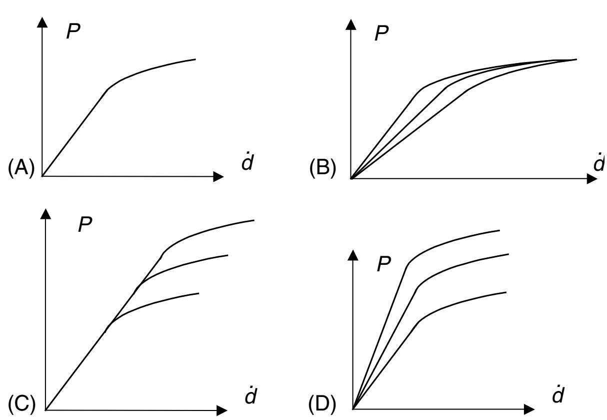


FIGURE 3.14 Common force characteristics of hydraulic dampers.

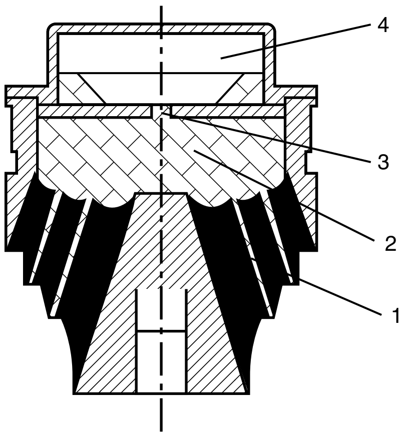


FIGURE 3.15 Hydraulic spring: (1) rubber-metal conical spring; (2) working fluid; (3) flow control valve; (4) compensation reservoir with rubber diaphragms.

1. Horn Guides

A simple primary suspension design uses horn guides to limit the movement of the axlebox (Figure 3.16).

This design has several disadvantages, including fast wear of friction surfaces leading to the increases in clearances, lack of elastic longitudinal and lateral characteristics, and increased friction force in vertical direction in traction and braking modes, when the axlebox is pressed against the slides. The design could be improved by the application of anti-friction materials that do not require lubrication and have high resistance to wear.

2. Cylindrical Guides

These comprise two vertical guides and two barrels sliding along them. Typically the vertical guides are attached to the bogie frame and the barrels to the axlebox as shown in Figure 3.17. The barrels are attached to the axlebox through rubber coaxial bushings and therefore provide some flexibility between the wheelset and the bogie frame in the longitudinal and lateral directions. Due to axial symmetry of the rubber bushes, the stiffness in longitudinal and lateral directions is the same, which may limit the provision of optimal suspension characteristics.

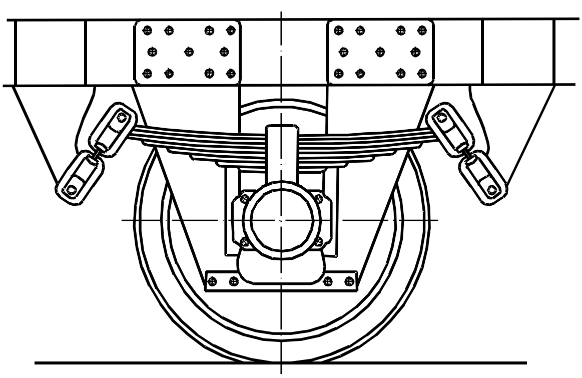


FIGURE 3.16 Axlebox located by horn guides.

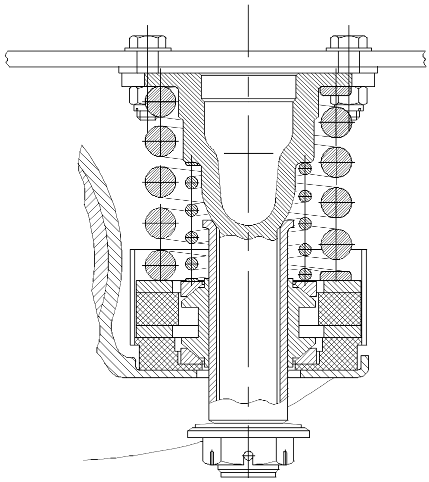


FIGURE 3.17 Connection between the axlebox and bogie frame using cylindrical guides.

Axlebox constraint with cylindrical guides, where the displacement of the axlebox along the guides occurs by shear deformation of multi-layer rubber-metal block is free from disadvantages of classical construction. Such axlebox designs are used on French TGV Y2-30 bogies. In order to obtain the optimum relationship of horizontal and vertical stiffness this block consists of two longitudinally oriented sections (Figure 3.18).

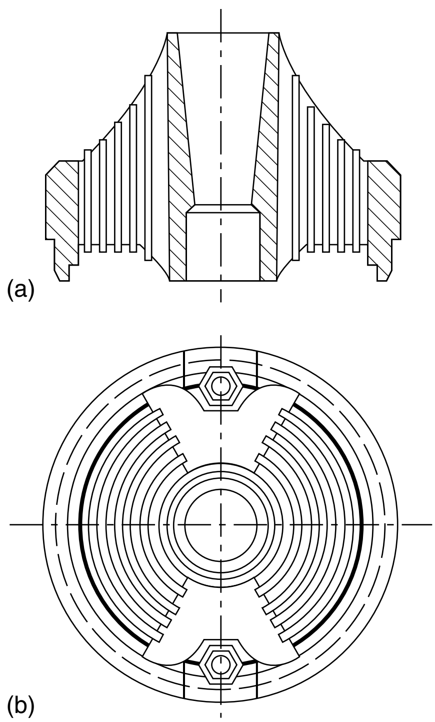


FIGURE 3.18 Two-section rubber-metal block used to connect the axlebox and bogie frame.

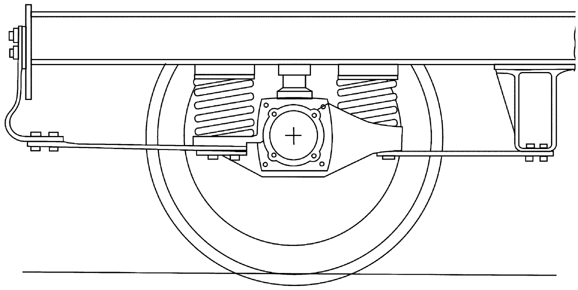


FIGURE 3.19 Connection between the axlebox and bogie frame using beam links.

3. Beam Links

The desire to avoid wear led to the development of links in the form of thin elastic beams that hold the wheelset in the longitudinal direction (Figure 3.19).

When primary suspension springs deflect, the beam links bend, whereas for traction and braking they experience tension or compression. To provide vertical flexibility in such construction it is necessary for at least one of the links to have longitudinal flexibility. This is achieved by attaching the beam to a longitudinally flexible spring support (the Minden Deutz link) or by attaching the links to the frame through radially elastic joints (IS primary suspension of Japanese trains).

The main disadvantage of such designs is high stress which develops around the joints at either end of the beam.

4. Constraints Using Radius Links

The use of rubber-metal bushes avoids surface friction and corresponding wear. The main problem with a radius link arrangement is obtaining linear motion of axleboxes when the links rotate. Alstom designed such an arrangement where the links are positioned on different levels in anti-parallelogram configuration (Figure 3.20) and this has found wide application. Links that connect

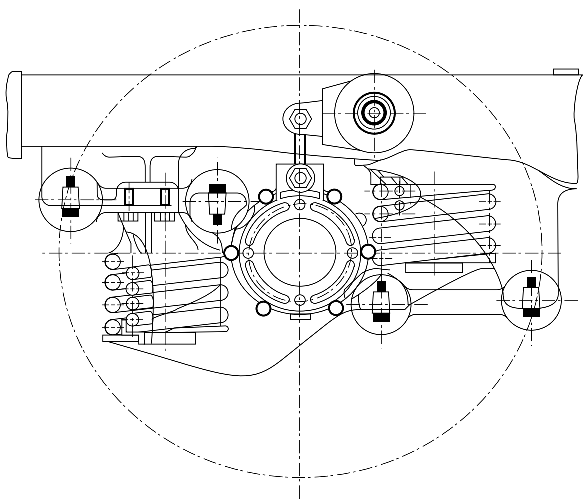


FIGURE 3.20 Radius links positioned at different heights in an anti-parallelogram configuration.

the axlebox to the frame provide linear displacement of its centre. By careful choice of size and the material of the rubber elements it is possible to obtain the required stiffness values in different directions. Due to the position of the links, lateral displacements do not cause misalignment of the axlebox therefore providing optimum conditions for the bearings.

Disadvantages of the radius link design include the significant vertical stiffness of the connection due to torsion stiffness of the bushes. Increasing the length of the levers would decrease the vertical stiffness, but it is limited by the space available in the bogie frame.

5. Constraints Using Trailing (Radial) Arms

Trailing arm suspensions allow the design of shorter and lighter bogie frames. Such designs are now widely used in passenger vehicle primary suspensions, such as the Y32 bogie shown in Figure 3.21.

The disadvantages of such designs include the longitudinal displacement of the axleboxes caused by vertical displacement of the suspension and torque applied to bogie frame due to wheelset lateral displacement.

6. Traction Rods

These are normally used to transmit longitudinal (traction and braking) forces in either the primary or secondary suspension. They are typically comprised of a rod with a rubber "doughnut" or bushes at each end. They may be adjustable length to maintain the necessary linear dimensions as wheels or suspension components wear (Figure 3.22).

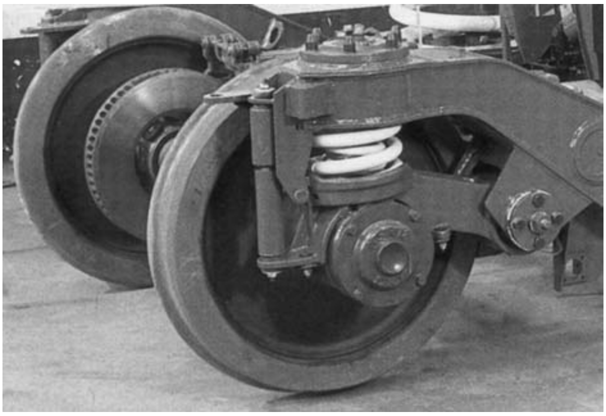


FIGURE 3.21 Trailing arm suspension on a Y32 bogie.

FIGURE 3.22 Traction rod.

F. Car Body to Bogie Connection

The connection between the car body and bogie must:

- Allow the bogie to turn relative to the car body in curves
- Transmit the vertical, traction and braking forces
- Provide additional control of lateral suspension inputs
- Assist in maintaining the stability of the bogie
- Provide longitudinal stability of bogie frames and equal distribution of load over the wheelsets (for traction rolling stock)

These problems are solved differently depending on the type of the rolling stock — traction or trailing, passenger or freight, moderate or high speed.

If the vehicle is stable up to the design speed, then introduction of additional yaw resistance torque is not necessary. If the static deflection of the suspension is sufficient, then vertical flexibility in the car body to bogie connection may not be necessary.

Designs generally aim to make the bogie to car body connection as simple as possible by the use of a small number of elements and reduction of the number of elements with surface friction.

1. Flat Centre Plate

In three-piece freight bogies the most common connection is the flat circular centre plate, that is secured by pin pivot at the centre (Figure 3.23).

The plate transmits the majority of the car body weight and the longitudinal and lateral interaction forces. The pin pivot has large in-plane gaps to the car body and only provides emergency restraint. When the car body rocks on the flat centre plate a gravitation resistance torque having soft characteristic is produced. The centre plate allows the bogie to rotate in curves and creates a friction torque that resists bogie rotation. Hence the circular centre plate provides a connection between the bogie and the car body in all directions.

Such a unit is of simple construction, but has several disadvantages. Firstly, clearances exist in the lateral and longitudinal directions. Secondly, relative motion occurs under high contact pressure and hence the surfaces are subject to significant wear. In curves, the car body leans on the side bearer creating additional friction torque that resists bogie rotation and increases wheel–rail forces. When the car body rocks on straight track, the contact surface becomes very small and high contact pressures can lead to cracks in the centre plate. To combat these problems, modern designs use a flat centre plate combined with elastic side bearers which resist car body rock and reduce the load on the centre bowl.

2. Spherical Centre Bowl

In this case, the car body rests on the spherical centre bowl and elastic side bearers (Figure 3.24).

The advantage of this design is the lack of clearance in the horizontal plane and no edge contact during car body roll. This results in reduced levels of contact stress and increases the centre bowl

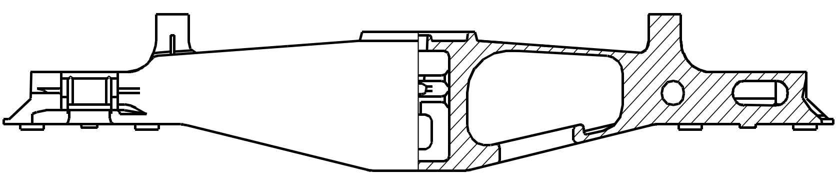


FIGURE 3.23 Flat centre plate.

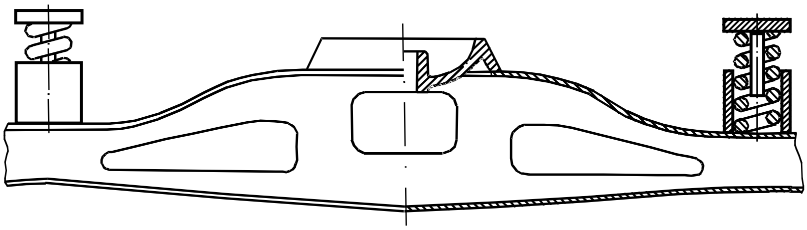


FIGURE 3.24 Spherical centre bowl.

service life. Such centre bowls are widely used in UIC freight bogies, electric trains, and underground cars in Russia.

3. Centre Pivot

The desire to exclude edge contact and increase the friction torque to resist bogie yaw led to development of bogies with centre pivots as shown in Figure 3.25. The majority of the car body mass is in this case transmitted to the side bearers and the car body can only turn relative to the bolster about the vertical axis.

This design is widely used in passenger coaches of former USSR. The disadvantages include the clearances in longitudinal and lateral directions. The design provides sufficient ride quality only for bogies having low lateral stiffness of secondary suspension.

4. Watts Linkage

This arrangement, illustrated in Figure 3.26, allows the bogie to rotate and move laterally whilst restricting longitudinal movement. It therefore provides a means of transmitting traction and braking forces. Pivots in the linkage are provided with rubber bushes to prevent the transmission of high frequency vibrations through the mechanism.

5. Pendulum Linkage

The pendulum linkage consists of a vertical rod connected at each end to the body and bogie frame by conical rubber bushes as shown in Figure 3.27. The mechanism is held in a central position by two precompressed springs. Elastic side supports provide lateral stability to the car body. For the small displacements that are typical of bogie hunting on straight track the pendulum support provides almost infinite stiffness determined by initial compression of springs. When large

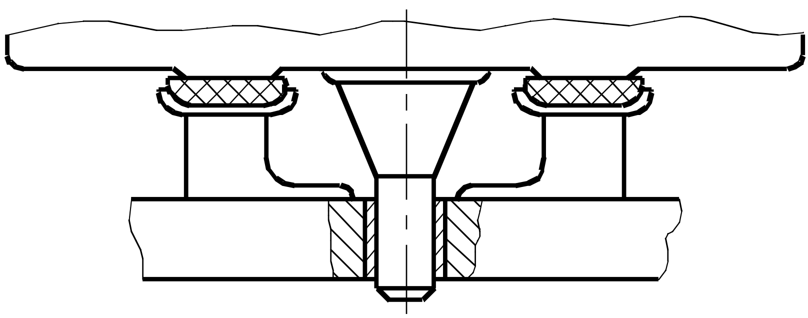


FIGURE 3.25 Centre pivot.

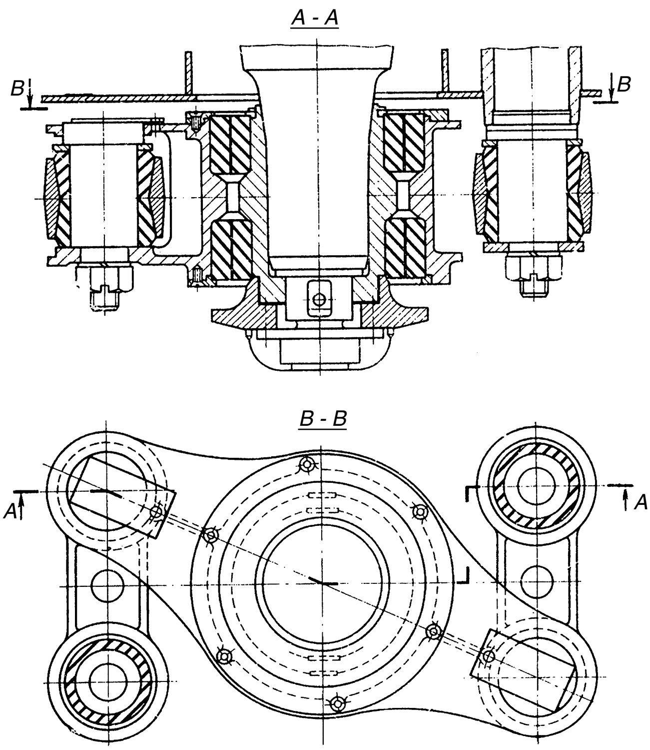


FIGURE 3.26 Watts linkage.

displacements develop in curves, the support provides low stiffness. Thus, the pendulum support has a soft nonlinear characteristic.

The drawback of such an arrangement is the rigid connection with a gap in the longitudinal direction, complex tuning requirements for the precompressed springs and friction forces in the additional sliding supports.

6. Connection of Car Body to Bolsterless Bogies

The complexity of the designs described above accounted for the development of modern bolsterless bogies using either flexicoil springs or air springs. In such suspensions the springs can

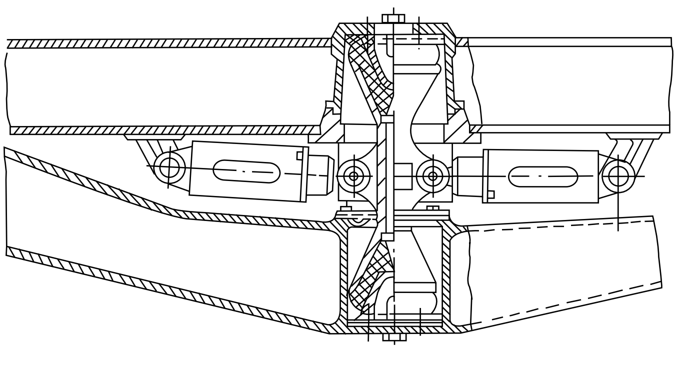


FIGURE 3.27 Pendulum linkage.

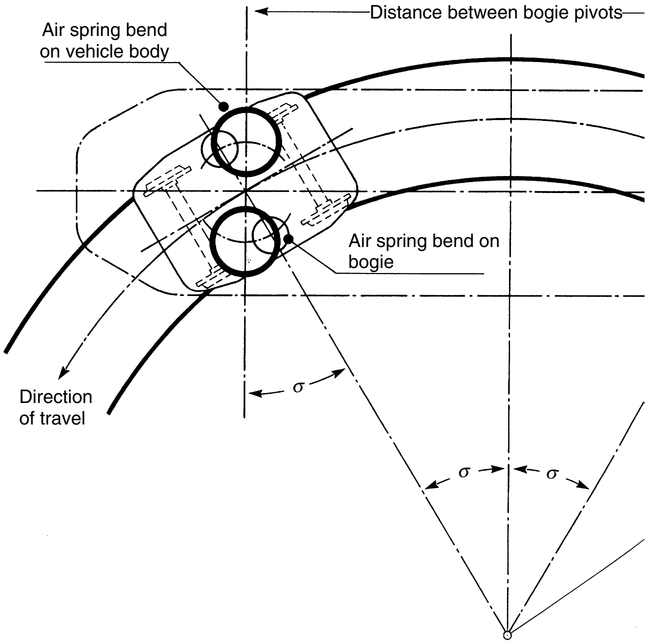


FIGURE 3.28 Schematic showing a bolsterless bogie passing a curve.

achieve large deflections in shear providing sufficiently large longitudinal displacements to allow the bogie to rotate in curves (Figure 3.28).

The top of the flexicoil springs rests on resilient blocks arranged to provide a cylindrical joint with rotation axis perpendicular to the track axis (Figure 3.29).

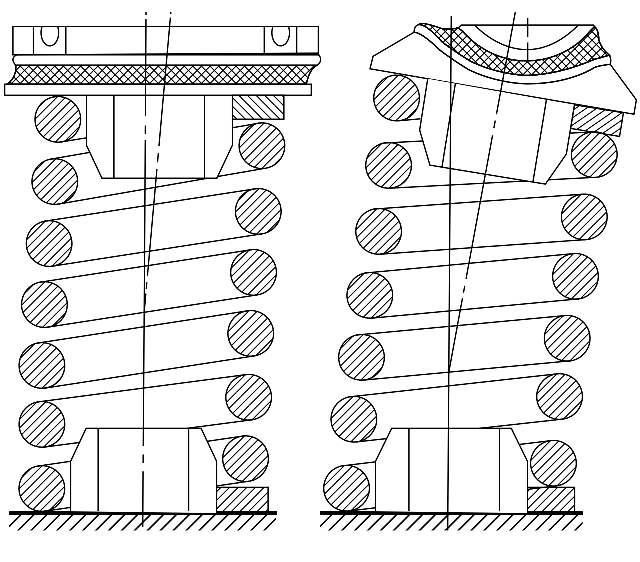


FIGURE 3.29 Spring resting on rubber-metal cylindrical joints.

A similar approach is used in bogies with secondary air suspension. In this case the air spring is often arranged in series with a rubber-metal spring to provide some suspension if the air spring deflates. Transmission of longitudinal forces is done through the centre pivot, Watts linkage, traction rods, or in the case of a Y32 bogie, through the backstay cables. Bolsterless bogie designs typically achieve reductions in bogie mass of around 0.5–1.0 t.

III. COMMON PASSENGER VEHICLE BOGIE DESIGNS

The most common passenger vehicle designs use a pair of two-axle bogies on each vehicle. However, in articulated trains, for example the French TGV, two-axle bogies are positioned between the car bodies, whilst the Spanish Talgo trains use single-axle articulated bogies.[1]

For passenger bogies, the wheelsets are generally mounted in a rigid H-shaped frame that splits the suspension in two stages. The primary suspension transmits forces from the wheelsets to the bogie frame and the secondary suspension transmits forces from the bogie frame to the car body.

The principal functions of the primary suspension are guidance of wheelsets on straight track and in curves, and isolation of the bogie frame from dynamic loads produced by track irregularities. The secondary suspension provides the reduction of dynamic accelerations acting on the car body which determines passenger comfort. The source of these accelerations is excitation from the track irregularity/roughness profile and the natural oscillations of the bogie frame and car body on their suspension elements. It is particularly important to reduce the lateral influences, to which the passengers are more sensitive, and therefore the stiffness of secondary suspension in lateral direction is designed as small as possible.

An example of a traditional type of secondary suspension (used on passenger vehicles for over 100 years) is shown in Figure 3.30. The secondary suspension swing consists of the secondary springs and dampers (2), spring plank (1) that is attached to the bogie frame (3) by swing links (4). This arrangement provides low lateral stiffness, and the height of the secondary springs remains comparatively small.

When curving, the bogie should rotate under the car body to reduce track forces, whereas on straight track it should resist yawing motion. In the case of bogies with swing link secondary suspension, part of the car body mass is transmitted to the bolster (5) through the bogie centre (6) and part through the side bearers (7). The bogie centre serves as the centre of rotation and transmits the traction forces whilst the side bearers provide friction damping to the bogie yaw motion. The traction rod usually limits longitudinal displacements of the bolster relative to the bogie frame.

Swing link secondary suspension may be acceptable for speeds up to 200 km/h. Its disadvantage is the large number of wearing parts that require relatively frequent maintenance to prevent deterioration of ride quality.

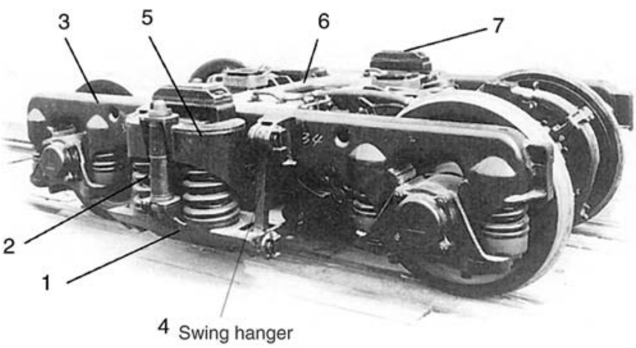


FIGURE 3.30 Bogie with swing link secondary suspension.

Modern bogie designs have a smaller number of parts in the secondary suspension and thus reduce maintenance costs. They typically use elastic elements that have a small stiffness in the horizontal direction. Examples include the ETR-500 bogie (Figure 3.31) which use Flexicoil secondary springs, and the Series E2 Shinkansen (Figure 3.32) which uses an air spring secondary suspension.

In such secondary suspension arrangements, the vehicle body may rest on a bolster (as in the swing link bogie), or directly mount on the secondary suspension, as in the bolsterless bogie in Figure 3.31 and Figure 3.32. In bolsterless bogies the traction forces are transmitted through the centre pivot arrangement, and the bogie rotates under the car body using the flexibility of secondary suspension in longitudinal direction. In such designs, yaw dampers are often fitted longitudinally between the body and bogie to damp hunting motion on straight track.

Modern bogies are normally equipped with separate secondary dampers to damp oscillations in vertical and lateral directions. Lateral damping is normally achieved with a hydraulic damper whilst vertical damping may be hydraulic or orifice damping within the air spring.

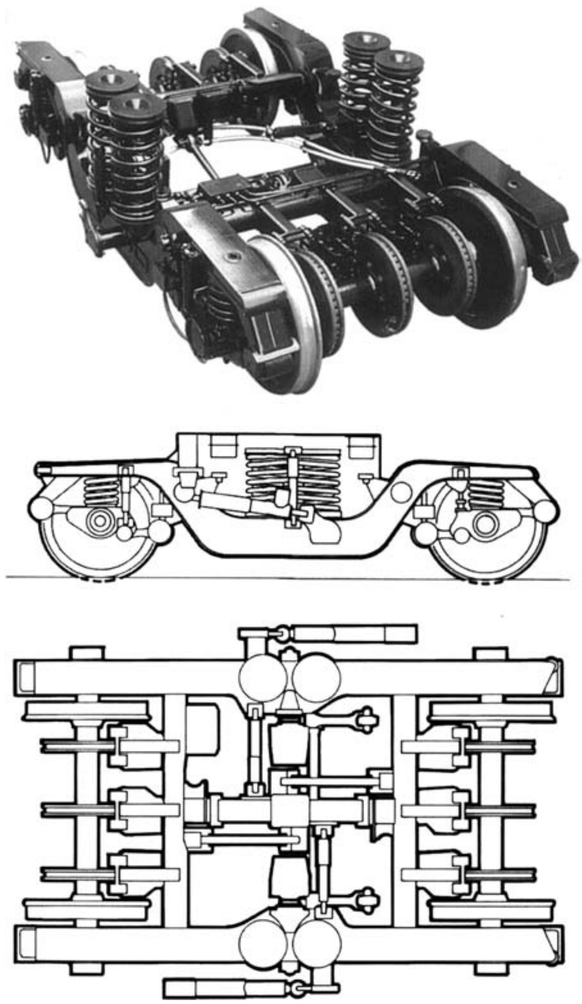


FIGURE 3.31 ETR-500 train bogie (Italy).

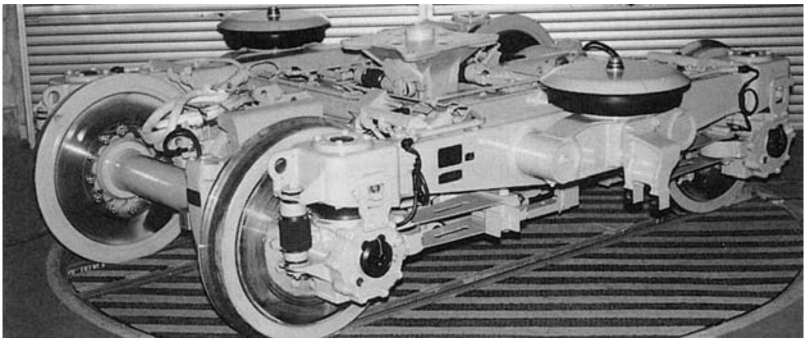


FIGURE 3.32 Series E2 Shinkansen bogie (Japan).

Various types of elastic elements are used in passenger bogie primary suspensions. To achieve high speeds the longitudinal stiffness of primary suspension should be high, whereas the lateral stiffness may be lower. In curves the high longitudinal primary stiffness leads to the increase of contact forces between the wheels and rails causing increased wear. Similarly, high lateral stiffness may lead to increased dynamic force when negotiating lateral track irregularities. For passenger bogies, it is therefore preferable if the suspension design can provide different stiffness in the lateral and longitudinal directions.

The three most common types of primary suspension are those with coil springs and longitudinal traction rods or links (Figure 3.33), coil springs with guide posts (Figure 3.34), and chevron (rubber interleaved) springs (Figure 3.35).

The ETR-460 bogie (Figure 3.33) is an example of a primary suspension using traction links with resilient bushes. The wheelset is guided by two links with spherical joints, and the vertical and lateral loads are mainly reacted by the coil springs.

In the primary suspension of Series 300 Shinkansen bogies, coil springs are used together with cylindrical guide posts containing rubber-metal blocks (Figure 3.34). The springs bear the vertical load whilst the rubber-metal block provides different stiffnesses in the longitudinal and lateral directions. It also acts to damp high frequency vibrations.

The X-2000 high speed train bogie primary suspension uses chevron (rubber-interleaved) springs (Figure 3.35). In this type of spring, rubber blocks are separated by steel plates arranged at an inclined position to the vertical. In this way vertical forces on the spring cause both shear and compression forces in the rubber blocks. Depending on the V-angle and material properties of the chevron spring, the longitudinal stiffness can be made three to six times higher than the lateral stiffness. The disadvantage of such design is that the mechanical properties are highly dependent on temperature and this may become a significant factor when operating in climates where extremes of temperature are common.

IV. COMMON FREIGHT WAGON BOGIE DESIGNS

In most cases, freight wagons use two two-axle bogies per vehicle. However, some articulated freight vehicles have been designed, principally flat wagons for container transportation.

The majority (Figure 3.36) of freight bogies have single-stage suspension, either between the wheelsets and the bogie frame (similar to passenger bogie primary suspension and often termed "axlebox suspension"), or between the bogie frame and the bolster (similar to passenger bogie secondary suspension and often termed "central" suspension). It can be seen from Figure 3.36 that central suspension make up approximately 6% more of the designs than axlebox suspension. Some

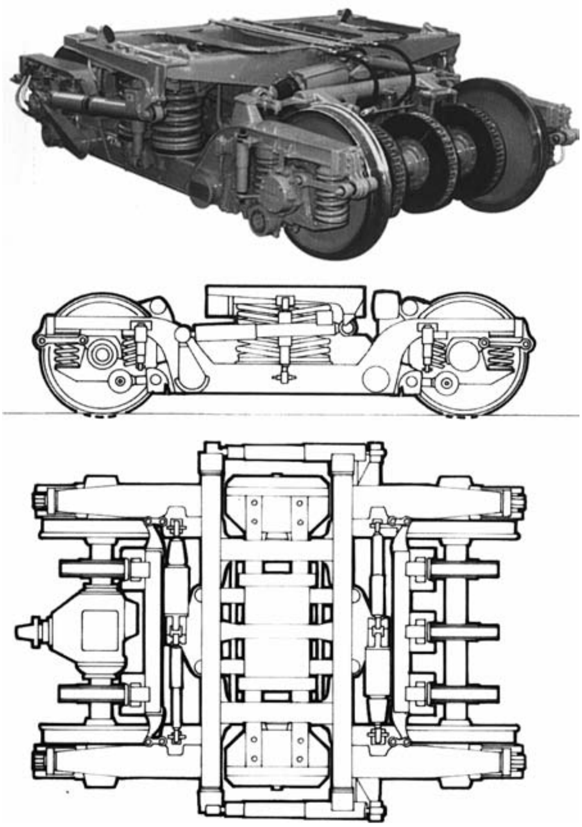


FIGURE 3.33 ETR-460 bogie (Italy).

wagons use double suspensions similar to passenger bogies to reduce track forces or improve isolation of the load from excess vibrations.

Bogies with central suspension are common in the countries of the former USSR, USA, Canada, China, Australia and most countries in Africa. Examples of the CNII-H3 (type 18-100) Russian bogie and the Barber bogie from USA are shown in Figure 3.37 and Figure 3.38, respectively. Such bogies are often termed "three-piece" bogies.

The frame of a three-piece bogie consists of the bolster and two side frames that are elastically connected by a coil spring and friction wedge-type central suspension system, that beside other functions resists asymmetrical loads and holds the bogie frame square in-plane. Such suspension allows independent pitch of the side frames when negotiating a large vertical irregularity on one rail, allowing the bogie to safely negotiate relatively poor track.

The vehicle body is connected to the bogie with a flat centre bowl and rigid side bearers having clearance in the vertical direction. When moving on straight track, the car body rocks on the centre bowl and does not touch the side bearers, the gravitational force providing recovery to the central position. In curves, the car body contacts the side bearers.

The central suspension consists of a set of nested coil springs and the wedge arrangement that provides friction damping in the vertical and lateral directions. The inclination of the friction wedges may vary between designs: in the 18-100 bogie the angle is 45°, whilst in the Barber bogie it is 35°.

Freight wagons suspensions have to operate under a wide range of load conditions from tare to fully laden, when axle loads can change by more than four times and the load on the spring set more

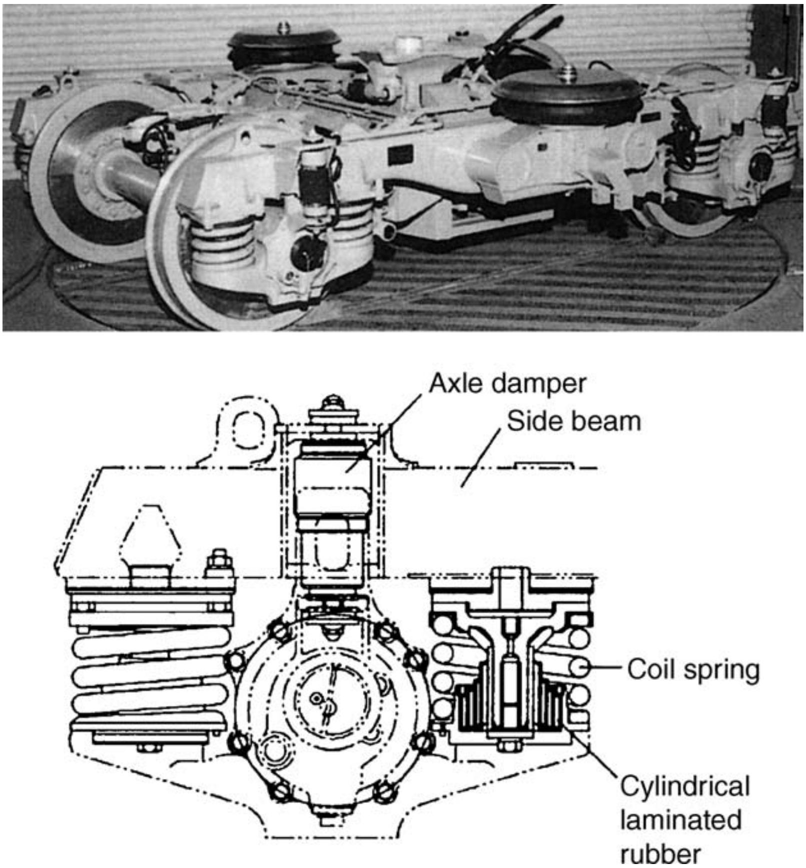


FIGURE 3.34 Series 300 Shinkansen bogie (Japan).

than five times. In the 18-100 bogie, the stiffness of the spring set is independent of the load, which leads to poor ride and increased derailment risk due to small deflections of the springs. For the Barber bogie, a range of suspension spring sets are available for axle loads from 7 to 34 t, that include spring sets with bilinear vertical force characteristics.

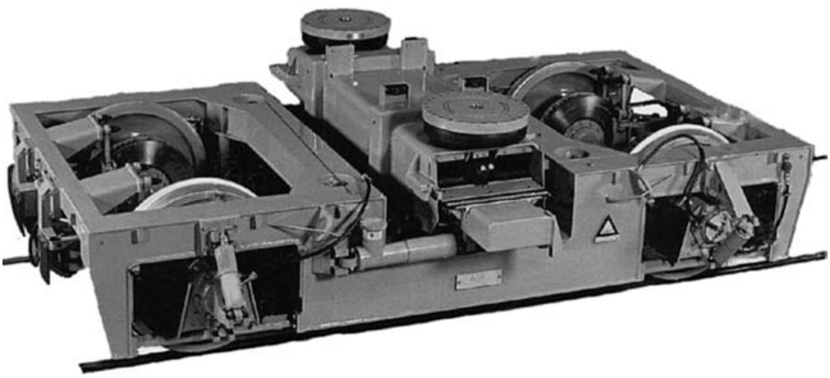


FIGURE 3.35 The X-2000 bogie with chevron spring primary suspension (Sweden).

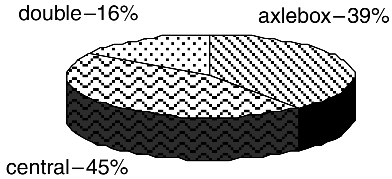


FIGURE 3.36 Proportions of freight bogies by suspension type.

The side frames of a three-piece bogie rest on the wheelsets. In the 18-100 bogie, the bearing is mounted inside the axlebox, whilst the Barber bogie has an adapter between the cylindrical cartridge-type bearing and the flat surface of the side frame. Clearances between the adapter (or the axlebox) and the side frame in the longitudinal and lateral directions allow the wheelsets to move in curves and when passing the large horizontal irregularities. Thus the axlebox unit does not steer the wheelsets, but damps their displacements by friction forces. Due to the absence of primary suspension, such bogies have a large unsprung mass which causes increased track forces.

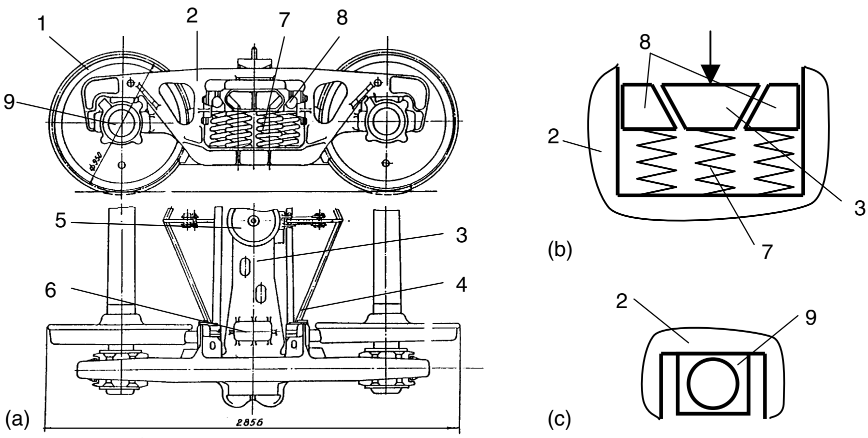


FIGURE 3.37 Type 18-100 bogie: (a) general view; (b) central suspension scheme; (c) primary "suspension" scheme (1, wheelset; 2, side frame; 3, bolster; 4, braking leverage; 5, centre bowl; 6, rigid side bearings; 7, suspension springs; 8, friction wedge; 9, axlebox).

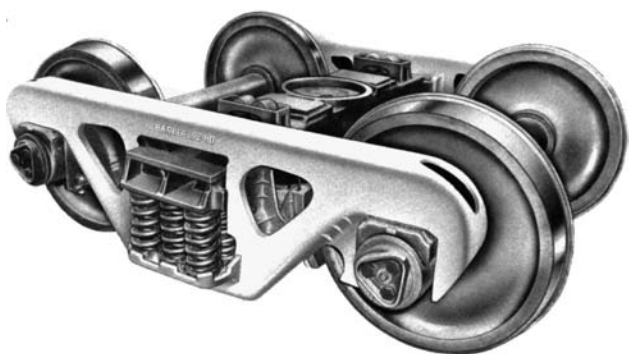


FIGURE 3.38 Barber S-2 bogie.

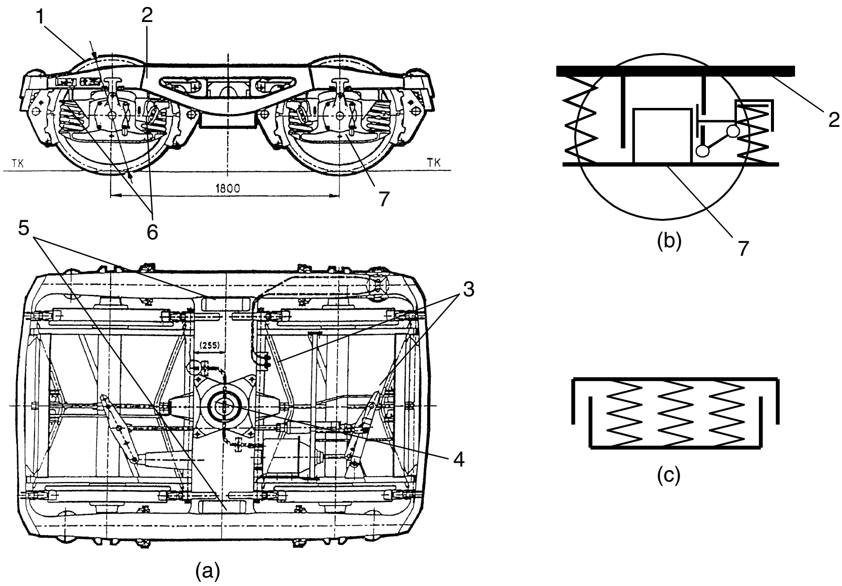


FIGURE 3.39 The Y25 bogie: (a) general view; (b) primary suspension scheme (Lenoir damper); (c) elastic side bearing scheme (1, wheelset; 2, rigid H-shaped frame; 3, braking leverage; 4, centre bowl; 5, side bearings; 6, spring set; 7, axlebox).

In curves, the three-piece bogies demonstrate the "lozenging" or "warping" effects, when the two side frames adopt a parallelogram position (in plan view). In this instance, the wheelsets cannot adopt a radial position in the curve, and generate large angles of attack, which leads to constant contact between the high-rail flange of the leading wheelset and the rail causing high levels of wear.

The Y25 (and similar bogies, such as the Y33) are commonly used on European freight vehicles. An example of this bogie is shown in Figure 3.39.

The Y25 bogie has a single-stage primary suspension consisting of a set of pairs of nested coil spring (with a bilinear characteristic for tare/laden ride) and a Lenoir link friction damper (Figure 3.39b) providing vertical and lateral damping. The friction force depends on the vertical load on the spring set, a component of which is transferred to the friction face by the inclined Lenoir link. Derailment safety is improved by the provision of vertical clearance between the inner and outer springs in each pair giving a lower stiffness in tare than in laden. Whilst improving the ride in both conditions, problems may still arise with the part-laden ride, when the bogie is just resting on the inner "load" spring making the suspension relatively stiff for the load being carried.

The bogie has a rigid H-shaped frame that consists of two longitudinal beams, one lateral and two end-beams and may be either cast or fabricated. The connection of the vehicle body is different to the three-piece bogies described above. The centre bowl has a spherical surface to reduce asymmetric forces on the frame and elastic side bearers without clearance resist the body roll motions (Figure 3.39c).

V. COMMON TRAM BOGIE DESIGNS

Trams and light rail vehicles (LRVs) are generally designed to negotiate very small radius curves and be compact enough for street running with overthrows minimised to avoid contact with cars. Modern trends for the low floor trams to improve accessibility leads to a requirement for very compact running gears.

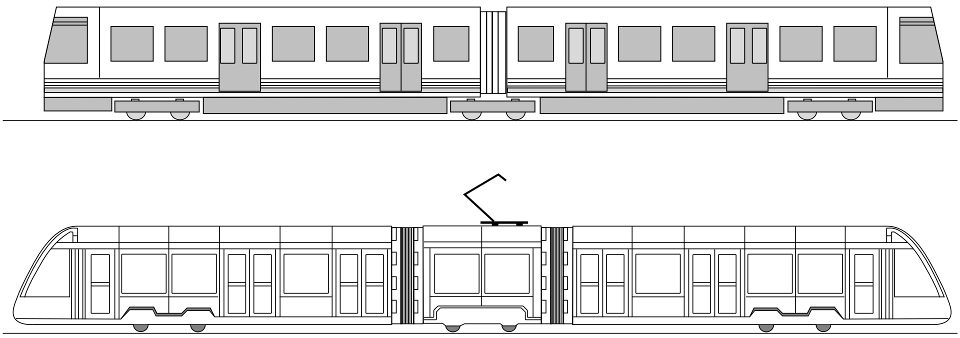


FIGURE 3.40 Typical tram configurations.

Traditionally tram bogies have a rigid frame and double suspension similar in design to conventional passenger bogies. Modern tram designs typically use one of the articulated arrangements shown in Figure 3.40.

In the example above, one short body section is rigidly fixed to the centre bogie, such that body section and bogie rotate together when passing a curve (Figure 3.41). Slewing rings are often used to join tram car bodies and bogies if it is necessary to allow for large rotation angles (Figure 3.42).

Wheelsets for tram bogies traditionally have a rigid axle (as in Figure 3.41 and Figure 3.42) and a small wheel diameter. However, low floor trams require alternative arrangements and in this case independently rotating wheels are often mounted in a common subframe (Figure 3.43).

Tram bogie primary suspensions typically use small rubber interleaved (chevron) springs. The primary suspension and bearings are generally located inside the wheels to reduce the overall size of the bogie.

Tram secondary suspensions use coil or air springs. The latter have the advantage of allowing constant floor height to be maintained for various loading conditions. Trams have smaller vertical deflection of the suspension compared to conventional passenger coaches as they operate at lower speeds and may have less stringent ride quality requirements due to shorter passenger journey times.

VI. PRINCIPLES OF SELECTING SUSPENSION PARAMETERS

The parameters of a rail vehicle may be considered optimal if its dynamic characteristics meet three groups of requirements:

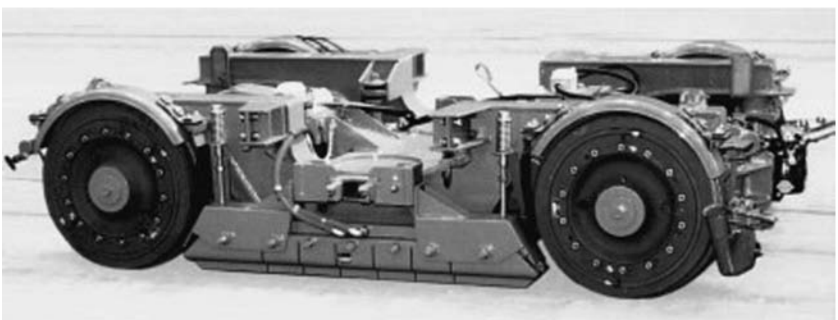


FIGURE 3.41 Tram bogie with rigid connection to the car body.

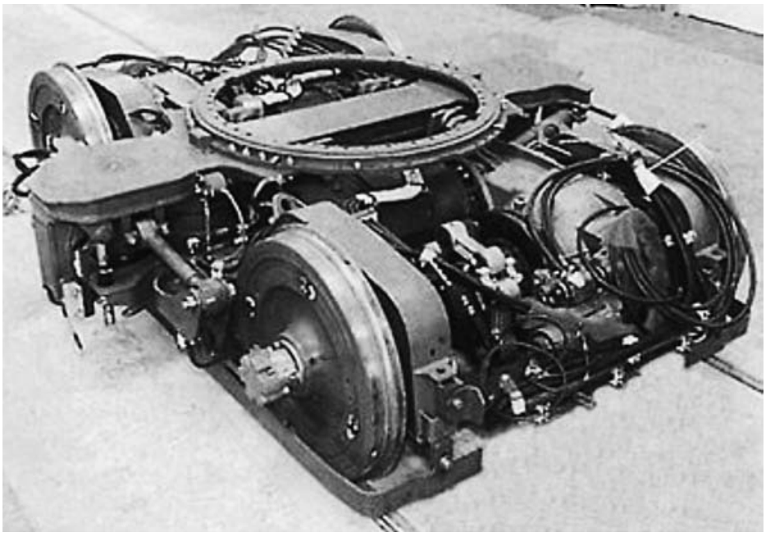


FIGURE 3.42 Bogie with ball bearing support of car body.

1. There is sufficient reserve of critical speed with respect to design speed
2. Ride quality, track forces, and safety factors satisfy the standards on straight track and in curves in all range of operation speeds
3. Wear of friction elements and wheel profiles is within acceptable limits.

Experience in the development of rail vehicles shows that at the preliminary stage the suspension parameters can be estimated using the simple engineering approaches described below. To make sure that the parameters are optimised, further refinement is usually done using computer simulation.

A. Selecting Vertical Suspension Characteristics

Suspension should control and damp the motions of both the sprung and unsprung masses in the vehicle to obtain the best possible ride qualities whilst strictly fulfilling the safety requirements

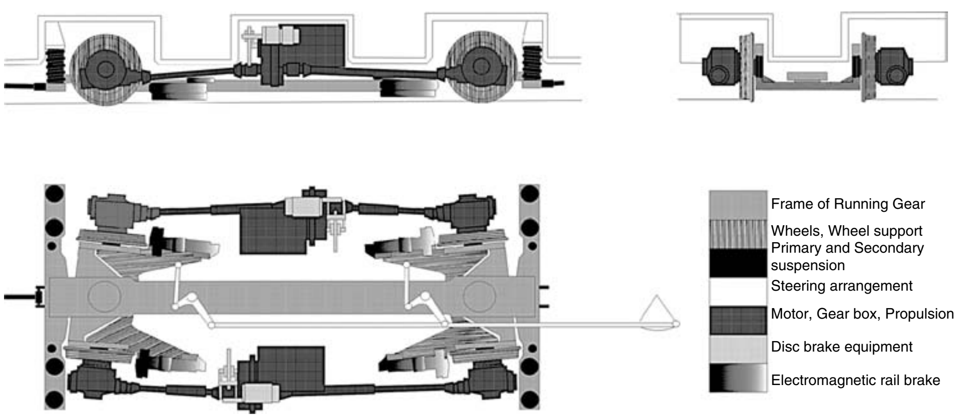


FIGURE 3.43 Tram bogie with independently rotating wheels.

satisfying specific service limitations such as ensuring that the vehicle remains within the loading gauge.

Bogie elastic elements have various constructions, as described above. In a simple initial analysis of the vertical behaviour, it is not the specific construction of the elastic element that is important, but the force characteristic that it provides, i.e., the dependence of the vertical load on the element P from its static deflection f: $P = P(f)$.

The static deflection of a suspension with linear characteristics (constant stiffness) is determined by formula:

$$f_{\mathrm{st}} = \frac{P_{\mathrm{st}}}{c} \tag{3.7}$$

where P_{st} is the static load on the suspension; c is the stiffness of the suspension.

For a linear suspension, there is a dependence between the bounce natural frequency and the static deflection:

$$\omega^2 = c/M = g/f_{\mathrm{st}}$$

where M is the sprung mass of the vehicle; g is the gravity acceleration.

Research has shown that decreasing the suspension stiffness is favourable for the dynamic performance of rail vehicles if other conditions do not change. In general, a low suspension stiffness gives lower accelerations but practical considerations dictate that there must be a relatively small height difference between tare and laden conditions. In addition the human perception of vibrations over a range of frequencies must be considered. For passenger vehicles, the body bounce frequency is generally in the range 0.9–1.2 Hz, whilst for freight wagons this frequency can rise to 2.5 Hz in laden and up to 4 Hz in the tare condition.

In order to avoid excessive suspension deflections, modern suspensions use nonlinear springs to provide optimal stiffness in the vicinity of the static deflection corresponding to required load.

In suspension elements with variable stiffness (Figure 3.44), the dynamic oscillations appear around an equilibrium position given by static force P_{st}. To estimate the oscillation frequency in this case the equivalent stiffness and equivalent deflection are used:

$$c_{\mathrm{eq}} = \left.\frac{\mathrm{d}P}{\mathrm{d}f}\right|_{P=P_{\mathrm{st}}}; \qquad f_{\mathrm{eq}} = \frac{P_{\mathrm{st}}}{c_{\mathrm{eq}}} \tag{3.8}$$

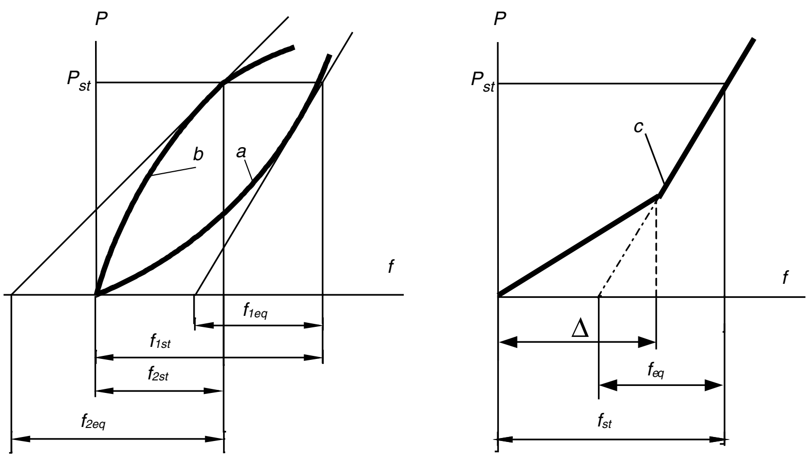


FIGURE 3.44 Nonlinear elastic force characteristics of a suspension: (a) stiff characteristic; (b) soft characteristic; (c) bilinear characteristic.

In a suspension with stiff force characteristic as shown in Figure 3.44a $f_{eq} < f_{st}$, and for a soft characteristic (Figure 3.44b) $f_{eq} > f_{st}$, where f_{st} is the total static deflection.

Freight rolling stock often uses a suspension with a bilinear force characteristic as shown in Figure 3.44c. The first part of the characteristic has a constant stiffness c_1 chosen to give the required frequency for the tare condition, whilst the second part with constant stiffness c_2 gives the required frequency for the laden wagon. In this case:

$$c_{eq} = \begin{cases} c_1, & f \le \Delta \\ c_2, & f > \Delta \end{cases}; \qquad f_{eq} = \begin{cases} \dfrac{P_{st}}{c_1}, & P_{st} \le c_1 \Delta \\ \dfrac{P_{st}}{c_2}, & P_{st} > c_1 \Delta \end{cases} \tag{3.9}$$

where Δ is the deflection corresponding to the breakpoint of the bilinear characteristic. The value of Δ is usually chosen so as to ensure that the breakpoint is not reached during normal suspension movements of the wagon in the tare condition.

Some further limitations on the value of suspension static deflection due to service conditions are discussed below.

An important limitation is imposed by the need to restrict height differences between the couplings of adjacent vehicles. The worst case is calculated from the coupling height of the gross laden car with maximum possible wear of bogie components and the height of the tare vehicle with new bogies (without wear). The difference in the coupler levels is due to static deflection of the suspension under the maximum load, aging of elastic elements and wear of bogie components (for example, wheel profile wear or wear of centre bowls and side bearers).

In service, the car body roll must also be limited to prevent the risk of overturning on highly canted curves and to ensure the vehicle remains within the required loading gauge. Once the maximum allowable roll angle for the vehicle body and the maximum lateral force (centrifugal, wind, and lateral component of the interaction force between the vehicles in curves) has been established, the equilibrium equation gives the minimum acceptable vertical stiffness of the suspension.

The final value of vertical stiffness for the suspension is chosen to be the maximum of the minimum values calculated using the service and design limitations.

B. Selecting the Lateral and Longitudinal Primary Suspension Stiffness

Theoretical investigations and experiments show that wheelset stability increases with increasing stiffness of the connection to the bogie frame. However, the character of this dependence is highly nonlinear and the relationship between suspension stiffness and the mass and conicity of the wheels influences the critical speed. Increasing the longitudinal stiffness of the primary suspension impairs the guidance properties of the wheelset in curves whilst increasing the lateral stiffness reduces the ability of the wheelset to safely negotiate large lateral irregularities.

A fundamental conflict therefore exists between the requirements for high speed stability on straight track and good curving with safe negotiation of track irregularities. The "in-plane" (lateral and longitudinal) stiffnesses must therefore be selected to give the best compromise for the conditions under which the vehicle will operate.

In order to make a preliminary choice of bogie in-plane stiffness, it is useful to know the relationship between stiffness and ride quality in an analytical or graphical form. The simplified approach described in Ref. 2 is useful as a starting point.

The natural vibration modes shown in Figure 3.45 and Figure 3.46 can be obtained from the linear equations of motion for two-axle bogie.[2,3]

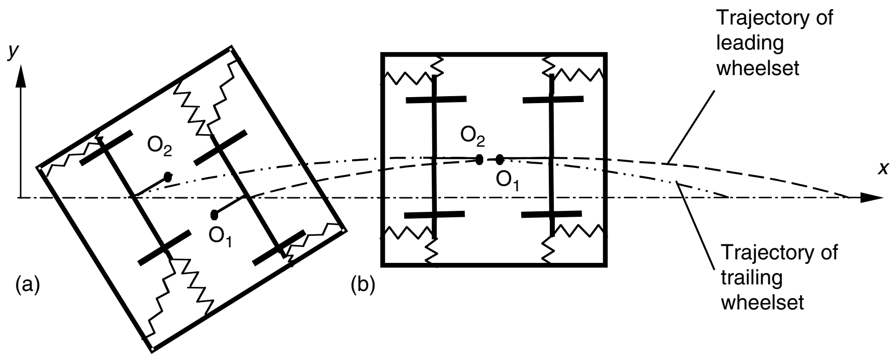


FIGURE 3.45 Wheelset modes for a two-axle bogie: (a) in-phase yaw; (b) in-phase lateral displacement.

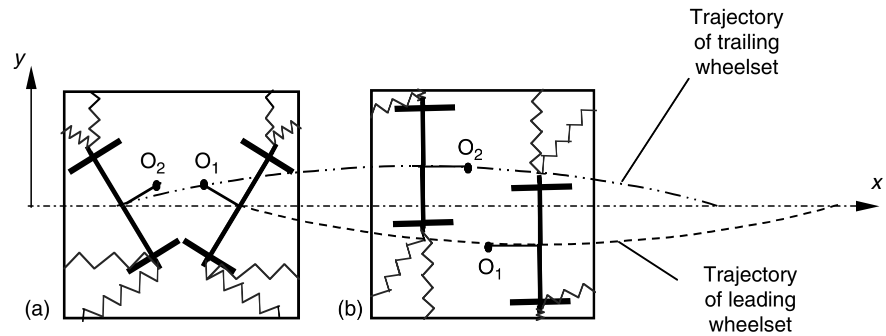


FIGURE 3.46 Wheelset modes for a two-axle bogie: (a) anti-phase yaw; (b) anti-phase lateral displacement.

Analysis of the modes shows:

- For in-phase yaw according (Figure 3.45a) there is a relative lateral displacement between the centres of wheelsets O_1 and O_2 and the bogie centre
- Similar lateral displacements appears for the anti-phase mode shown in Figure 3.46b
- Relative rotation between wheelset centres O_1 and O_2 occur only for anti-phase yaw of wheelsets (Figure 3.46a).

Thus, two generalised parameters can be introduced for the bogie:

1. A stiffness corresponding to relative lateral displacement between the centres of wheelsets referred to as the shear stiffness (K_s)
2. A stiffness corresponding to the relative yaw angle between the wheelsets referred to as the bending stiffness (K_b)

The conventional representation of bogie shear and bending stiffness is shown in Figure 3.47 as translational and torsion springs, respectively. The generalised stiffnesses K_s and K_b have a particular physical meaning. The shear stiffness K_s has a greater influence on the critical speed of the vehicle, whilst the bending stiffness K_b mainly determines the wheelsets' angles of attack in curves.

The use of shear and bending stiffness to give a simplified representation of the primary suspension without consideration of the bogie frame inertia (Figure 3.47) allows the in-plane bogie stiffnesses to be chosen without considering its specific design.

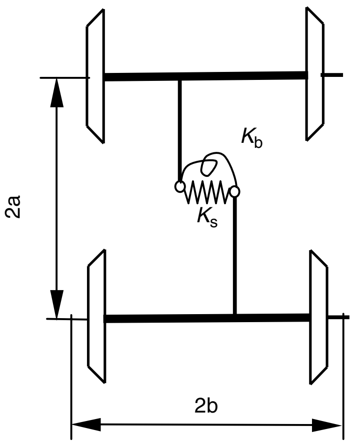


FIGURE 3.47 Representation of the primary suspension using shear and bending stiffness.

Solution of the stability problem[4] shows that the critical speed of a conventional railway vehicle is a function of its shear and bending stiffness as shown in Figure 3.48. The quality of curving can be estimated using the relationship of the wear number (the sum of creep force power for all wheels of the vehicle) to the shear and bending stiffness as shown in Figure 3.49.

These relationships show that the chosen bending stiffness of the bogie should be the minimum that provides the required critical speed and the shear stiffness should be within the critical speed range for the chosen bending stiffness.

C. Selecting Suspension Damping

Damping is typically provided within the suspension by either friction or hydraulic devices. Some types of elastic elements, such as leaf springs, have sufficient internal friction damping to avoid the neccesity of a separate damper.

The selection of the optimum damping levels is more complicated than the choice of suspension stiffness. High levels of damping decrease the amplitudes of vibrations in resonance situations but significantly increase the accelerations acting on the vehicle body for higher frequency inputs such as short wavelength track irregularities.

Hydraulic dampers are almost universally used for passenger vehicles. Let us consider the simplified case of linear dependence between the damper force and the velocity. In this case attenuation of vehicle vibrations is determined by the ratio of the real part of the eigenvalue to the corresponding natural frequency. This is termed the damping coefficient and is different for different natural vibration modes:

$$d_i = \frac{1}{2\omega_i} \frac{\{v_i\}^{\mathrm{T}}[B]\{v_i\}}{\{v_i\}^{\mathrm{T}}[M]\{v_i\}} \tag{3.10}$$

where $[B]$, $[M]$ are the damping and inertia matrices of the vehicle multi-body model, respectively, $\{v_i\}$ is the column-vector of ith eigenmode and ω_i is the natural frequency of ith eigenmode. In a simple multibody model, the vehicle body, wheelsets, and bogie frames are represented by rigid bodies connected with the elastic and damping elements.

Effective damping of the vibrations of railway vehicles is typically obtained with damping coefficients which lie in the following ranges: 0.2–0.3 for vertical oscillations; 0.3–0.4 for horizontal oscillations, and 0.1–0.2 for vehicle body roll.

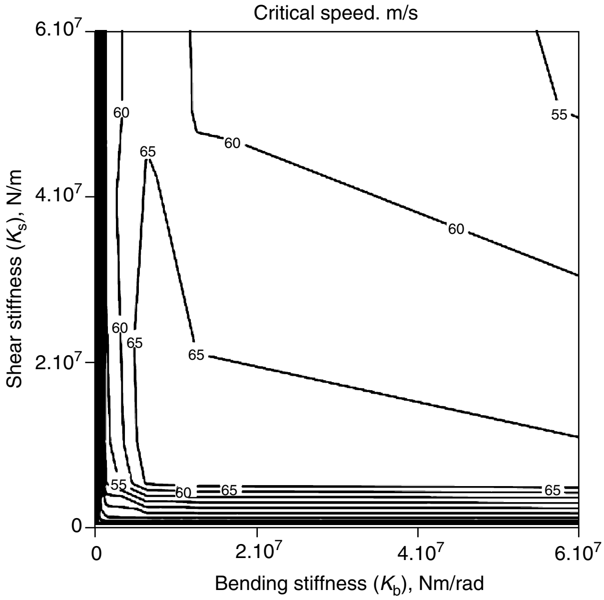


FIGURE 3.48 Critical speed as a function of shear and bending stiffness.

In freight bogies, friction dampers are commonly used. When making the preliminary choice of parameters, the friction force in the damper is estimated on the basis that the amplitude should not increase in the resonance case.

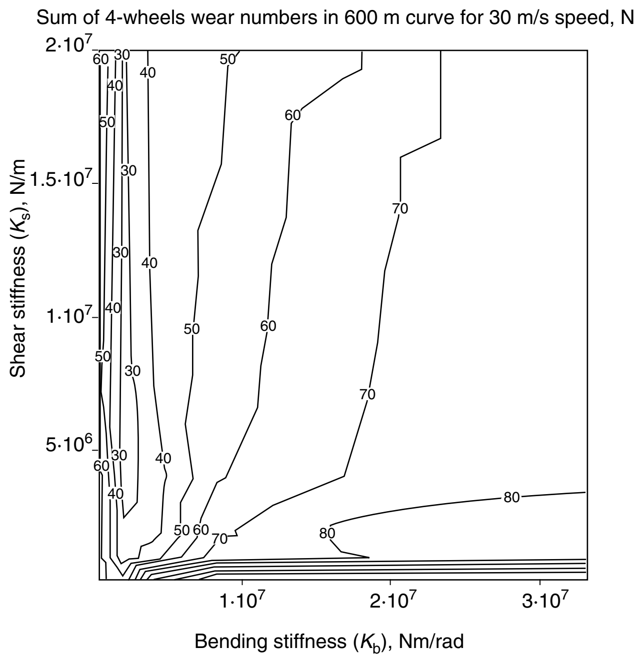


FIGURE 3.49 Wear number as a function of shear and bending stiffness for a 600 m curve at 30 m/sec.

Assuming that the amplitude of oscillations at resonance increases by $\Delta A'$ during one period, and the friction force F acting in the suspension reduces it by $\Delta A''$, the following conditions must apply to prevent the amplitude increasing in the resonant case:

$$\Delta A'' \geq \Delta A' \tag{3.11}$$

The equations of oscillation for the system with dry friction under periodic excitation give:

$$F \geq \frac{\pi q}{4} c_{\mathrm{eq}} \tag{3.12}$$

where q is the estimated amplitude of periodic track irregularity (prescribed in regulations) and c_{eq} is the equivalent stiffness of the suspension.

Estimating the magnitude of the friction force is easier using relative friction coefficients that equal the ratio of friction force to the static vertical load:

$$\varphi = \frac{F}{P_{\mathrm{st}}} \geq \frac{\pi q}{4 f_{\mathrm{eq}}} \tag{3.13}$$

where

$$f_{\mathrm{eq}} = \frac{P_{\mathrm{st}}}{c_{\mathrm{eq}}}$$

For freight cars the recommended value of relative friction coefficient is typically in the range 0.05–0.15.

The relative friction coefficient is a general parameter of the wagon, and the optimal value of friction force depends on the equivalent static deflection of the suspension, or for the case of nonlinear suspension characteristic, the vertical load.

VII. ADVANCED BOGIE DESIGNS

Many novel bogie designs address the fundamental conflict between stability on straight track and good curving described above. It is clear from the foregoing discussion that the bogie should maintain stable conditions on straight track but allow the wheelsets to adopt a radial position in curves.

Bogies where the wheelsets adopt or are forced to take an approximately radial position in curves (Figure 3.50) are called radially steered bogies. Such designs have small angles of attack which leads to significant decrease of flange wear and lower track forces.

Radially steered bogies fall into two groups: those with forced steering of the wheelsets in curves and those with self-steering of the wheelsets. In the first case, the wheelsets are forced to adopt a radial position due to linkages between the wheelsets or linkages from the wheelset to the vehicle body. Various methods of obtaining forced steering for radially steered bogies are shown in Table 3.4. The bogies may be split into three groups depending on the control principle used:

1. Wheelsets yawed by the wheel-rail contact forces
2. Wheelsets yawed by the relative rotation between the bogie frame and vehicle body (either yaw or roll)
3. Wheelsets yawed by an external energy source (electric, hydraulic, or pneumatic actuators)

The first two groups in Table 3.4 have passive control systems that change the kinematic motion of the wheelset depending on the curve radius. Designs where the energy source is provided by

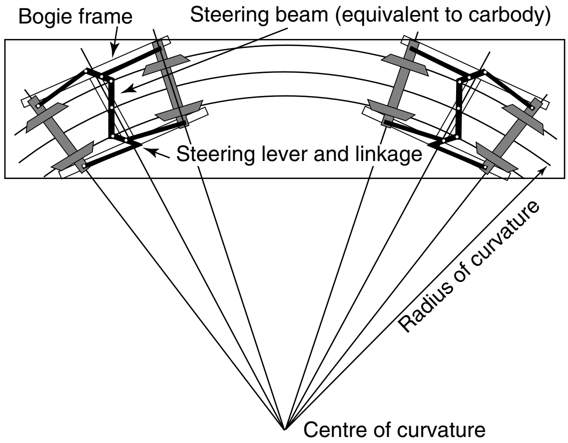


FIGURE 3.50 Radial position of wheelsets for a bogie with inter-axle steering linkages.

steering force in wheel–rail contact, may be considered preferable as the behaviour of systems relying on interconnection to the car body is dependent upon vehicle speed.

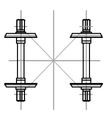
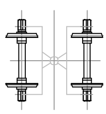
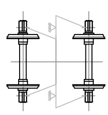
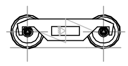
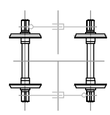
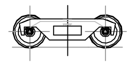
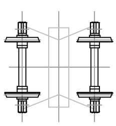
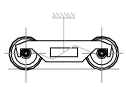
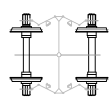
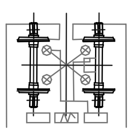
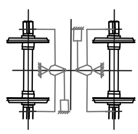
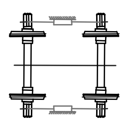
Designs where the wheelsets are forced to adopt a radial position by hydraulic, pneumatic, electric actuators (or a combination of these) are called actively controlled bogies. These are considered in detail in Chapter 6.

Three main groups of executive mechanisms are common: those using links between wheelsets, those using an arrangement of levers, or those using sliders.

Figure 3.51 shows a passenger bogie using a passive steering system with Watts linkage.

An example of a freight bogie with passive control using diagonal links between the axleboxes designed by Scheffel is shown in Figure 3.52.

TABLE 3.4
Classification of Forced Steering Mechanisms

Energy Source — Control Mechanism	Steering Mechanisms		
	Links	Levers	Sliders
Wheel–rail interaction	 	 	 
Car body yaw or centrifugal force action		 	
Active external force			

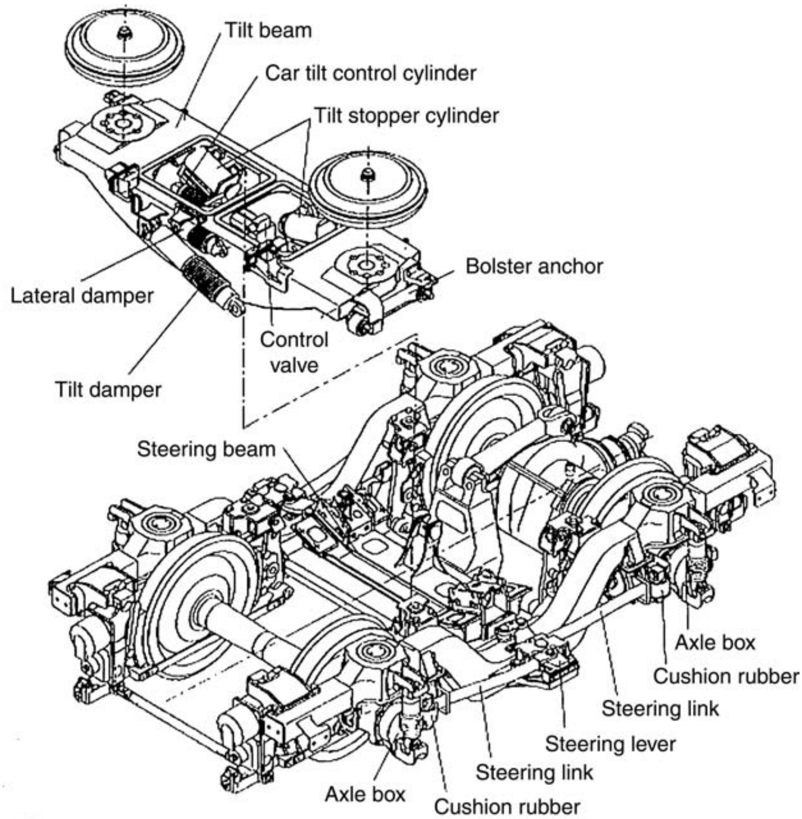


FIGURE 3.51 Passenger car bogie with passive steering system.

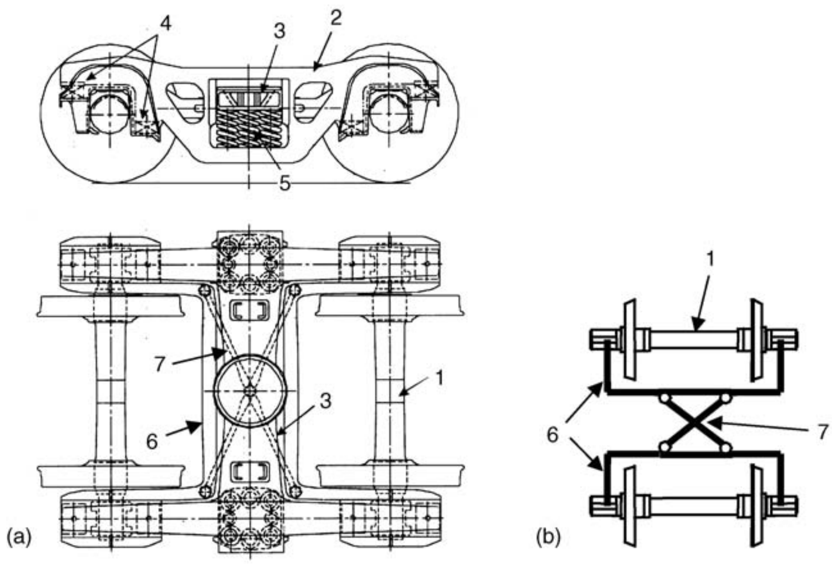


FIGURE 3.52 Scheffel HS bogie with diagonal linkage between wheelsets: (a) general view; (b) the principal scheme of inter-axle linkages (1, wheelset; 2, side frame; 3, bolster; 4, primary suspension; 5, secondary suspension; 6, subframe; 7, diagonal links).

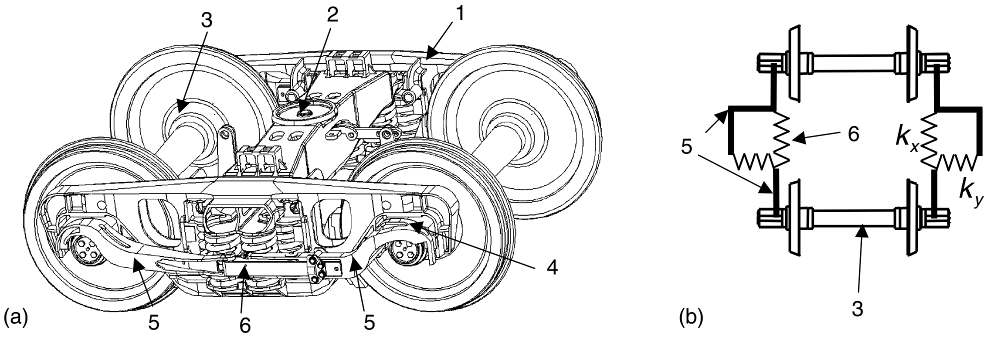


FIGURE 3.53 Scheffel radial arm bogie: (a) general view; (b) principal scheme of inter-axle linkages (1, side frame; 2, bolster; 3, wheelset; 4, primary suspension; 5, two longitudinal arms; 6, elastic elements between the arms).

The second group of radially steered bogies are those with wheelsets which are self-steering in curves. The design of such bogies is based on selecting the optimum shear and bending stiffnesses. This may be aided by using designs that allow these stifnesses to be decoupled.

In conventional suspension arrangements, the bending and shear stiffness are not independent. Decreasing the bending stiffness leads to a reduction of shear stiffness, which means that improving the curving qualities leads to reduced stability on straight track. Inevitably, therefore, the bogie in-plane stiffness is chosen to give the best compromise between curving and stability.

In order to resolve the curving–stability controversy, Scheffel proposed several arrangements of inter-axle linkages,[4] two of which are shown in Figure 3.53 and Figure 3.54.

The Scheffel radial arm bogie is shown in Figure 3.53, and for it the generalised bogie stiffness has the following expressions:

Shear stiffness:

$$K_{s\Sigma} = 2k_y + K_s \qquad (3.14)$$

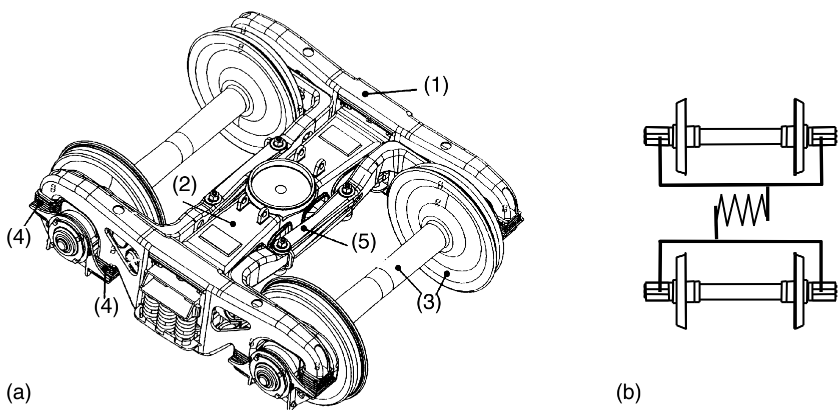


FIGURE 3.54 Scheffel bogie with A-frame inter-axle linkages: (a) general view; (b) principal scheme of inter-axle linkages (1, side frame; 2, bolster; 3, wheelset; 4, primary suspension; 5, elastic connection between the subframes).

where k_y is the lateral stiffness of inter-axle linkage (per side) and K_s is the shear stiffness provided by the bogie frame.

Bending stiffness:

$$K_{b\Sigma} = 4b^2k_x + K_b \tag{3.15}$$

where k_x is the longitudinal stiffness of inter-axle linkage (per side) and K_b is the bending stiffness provided by the bogie frame.

Thus, the expressions for $K_{s\Sigma}$ and $K_{b\Sigma}$ contain two independent parameters k_x and k_y that allow optimum shear and bending stiffnesses to be selected.

Such bogie designs are based on the three-piece bogie consisting of a bolster and two side frames. The Scheffel designs retain the advantages of the three-piece bogie when negotiating large track irregularities and carrying the asymmetric loads. However, wheelset steering is provided not by the frame (as in traditional designs), but by the inter-axle links. In order for the inter-axle links to be effective, the bogie must have a low longitudinal and lateral primary suspension stiffness. These bogie designs are effectively therefore double suspended.

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