3 Design of Unpowered Railway Vehicles

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CONTENTS

3.1 General Vehicle Structure, Main Functions and Terminology ..................................................43
  3.1.1 Car Bodies ..................................................................................................................47
  3.1.2 Running Gears, Bogies and Suspensions .....................................................................49
  3.1.3 Couplers, Automatic Couplers and Draw Bars ..........................................................51
  3.1.4 Pneumatic Brakes .......................................................................................................53
3.2 Design of Car Bodies ..............................................................................................................54
  3.2.1 Passenger Coaches ......................................................................................................54
  3.2.2 Freight Wagons ...........................................................................................................59
3.3 Running Gears and Components ............................................................................................70
  3.3.1 Wheelsets .................................................................................................................... 71
  3.3.2 Axleboxes and Cartridge Type Bearings .....................................................................75
  3.3.3 Wheels ........................................................................................................................ 77
3.4 Design of Freight Wagons Bogies ...........................................................................................79
  3.4.1 Three-Piece Bogies .....................................................................................................79
  3.4.2 Primary Suspended H-Frame Bogies ..........................................................................81
  3.4.3 Double-Suspension Bogies .........................................................................................82
  3.4.4 Auxiliary Suspensions (Cross-anchors, Radial Arms, etc.) .......................................83
3.5 Design of Unpowered Bogies for Passenger Coaches ............................................................86
3.6 Design of Inter-Car Connections ............................................................................................91
  3.6.1 Screw Couplers ...........................................................................................................91
  3.6.2 Automatic Couplers ................................................................................................. 92
  3.6.3 Draft Gear ...................................................................................................................95
  3.6.4 Buffers ...................................................................................................................... 98
  3.6.5 Inter-Car Gangways ..................................................................................................100
3.7 Principles for the Design of Suspensions ..............................................................................103
  3.7.1 Suspension Characteristics in Vertical Direction ......................................................103
  3.7.2 In-Plane Suspension Stiffness ....................................................................................106
  3.7.3 Suspension Damping .................................................................................................108
  3.7.4 Car Body to Bogie Connections ...............................................................................110
References ......................................................................................................................................112

3.1 GENERAL VEHICLE STRUCTURE, MAIN FUNCTIONS AND TERMINOLOGY

Unpowered railway vehicles are mainly operated in trains being drawn by a locomotive or several locomotives distributed along the train. This is the basic technology that is equally used for transportation of freight and passengers, providing high energy efficiency. Further, we refer to freight unpowered vehicles as wagons and passenger vehicles as coaches. Typical examples are shown in Figure 3.1.
For any railway vehicle, the axle load (mass transmitted onto the rails by one wheelset) is the major characteristic, along with the maximum design speed. In freight transportation, axle loads between 22.5 t and 37.5 t are used, with design speeds below 160 km/h (the bigger the axle load, the smaller the design speed), whereas the tendency for passenger coaches is increasing the design speed while maintaining the axle load as small as possible.

Freight train consists can reach up to 48,000 t in total weight and up to 320 wagons, limited by locomotive capabilities and the landscape of the railway, to achieve maximum efficiency (see heavy-haul freight train example in Figure 3.2). Efficiency of the individual wagon is determined by its mass capacity and volume capacity that indicate the possible maximum mass of the cargo and the absolute maximum volume of the cargo that it can carry. The minimum possible mass of the wagon is its tare mass (when totally empty), and the maximum mass is the fully laden mass (tare plus the capacity) that should not exceed the limitation for the axle load.

The length of passenger trains is usually limited by platform size, and speed and comfort are valued more than the mass of the passengers; thus, coaches are characterised by passenger capacity and carry a lot of specific equipment to provide comfort.

The locomotive hauls the train, while each unpowered vehicle produces the running resistance due to interaction between the wheel and rail, oscillations and damping in the suspension, friction in bearings, air and gravity resistance, longitudinal oscillations between vehicles, etc. To stop the train, brakes are applied to each vehicle individually. The lifecycle of unpowered vehicles also includes loading and unloading operations (with external or built-in devices), shunting manoeuvres during assembling and disassembling of the train and coupling to and uncoupling from the adjacent vehicles. When in the train, the locomotive accelerates or decelerates the vehicles, and they pass straight or curved sections of the railway track, change tracks when passing through switches and crossings, experience wind and interact with bridges and tunnels.

All of these described operational scenarios need to be handled by vehicle design. The main parts of the unpowered railway vehicle are the car body, the bogies and the braking and coupling equipment, as shown in Figure 3.3.

**Number of axles:** In general, the vehicles are classified by the number of axles (wheelsets); examples are shown in Figure 3.4. Some special transport vehicles for super-heavy cargo can have two bogies with up to 8 axles in each of them. There may be more than two sections in articulated vehicles or one-axle bogies under the articulation. In addition, bogies in articulated vehicles with different numbers of axles can be positioned under the articulation (three-axle) and under the ends of the car body sections (two-axle). Because the axle load is limited, the number of axles is used to distribute the mass of the vehicle over the length of the railway track.
The following characteristics are used to describe the geometry of railway vehicles:

*Vehicle wheel base:* Wheel base of the vehicle is the longitudinal dimension between the centres of outer bogies or between the centres of outer suspensions of the vehicle. For articulated vehicles, as shown in Figure 3.5, besides the wheel base of the whole vehicle, the wheel base of each section can be used, designating the longitudinal distance between the centres of neighbouring bogies.
Length between coupler axes: The length of the vehicle in the train or how much space it takes up in the train is measured between the axes of the coupling devices. This parameter is also important for spacing the loading and unloading devices at the terminal facilities and for distributing the mass of the vehicles in the train over the length of the track. The bearing capacity of bridges that are longer than one vehicle is determined by the permissible load per unit length of the vehicle, that is, fully laden weight of the vehicle divided by its length between coupler axes. To provide safe curving, the length of four-axle vehicles between coupler axes does not exceed 25 m.
Length of the cantilever part: The difference between the length between coupler axes and vehicle wheel base is double the length of the cantilever part. This parameter becomes important when the vehicles in the train are passing curved sections of the track, because it determines the lateral displacement of the coupling devices.

Car body length: Car body length is its maximum dimension along the track. In most vehicles, car body length equals the length between end beams, but some designs have a protruding top part that increases the loading volume.

Coupler height (above the rail top level): This parameter determines the safety of coupling and prevents uncoupling when the train is moving over summit parts of sloped track. In production, the coupler height is measured in empty vehicles, and it depends on the tolerance for bogie height and tolerances of car body centre sill, as well as the tare weight of the empty car body. In operation, the coupler height is even more variable, as it is dependent on wheel diameter that changes due to turning, suspension deflection that increases when the vehicle is loaded, wear of bogie horizontal surfaces and wear of coupling device surfaces. The railway administrations set safety limits for coupler height in empty and laden vehicle conditions [1,2].

 Clearance diagram: The vehicle dimensions in cross-section laterally to the track are limited by the so-called clearance diagram. Two tracks can be laid next to each other, various platforms and other buildings stand near the track and tracks can go over bridges and into tunnels, and on electrified railways, there is the contact wire running over the track. To limit the vertical (top and bottom) and lateral dimensions of the vehicle and to provide safe clearance between the vehicle and stationary structures on straight track and in curves, the clearance diagram was introduced. Conformity of vehicles to clearance diagrams is a basic safety requirement of all railway administrations [3–5].

3.1.1 Car Bodies

During almost 200 years, the evolution of freight wagons and coaches created a huge number of various car body designs for many purposes. Typical examples are shown in Figure 3.6. There are specific freight wagons to carry bulk named gondola wagons (either drop bottom or solid bottom), to carry grain or mineral fertilisers with the capability of discharge though bottom hatches named hopper wagons, to carry liquids named tank wagons, to carry containers named flat wagons, as well as more specific designs.

FIGURE 3.6 Various types of freight wagon car bodies: (a) drop-bottom gondola, (b) timber flat wagon, (c) tank wagon and (d) hopper wagon.
Passenger coaches differ by either providing travel for sitting or sleeping passengers; there also exist restaurant coaches, laboratory coaches with measurement equipment to monitor railway condition and many other specific types. To increase the passenger capacity of the coach, double decks, as shown in Figure 3.7, can be introduced if the clearance diagram permits.

No matter the variety of car body types, in rail vehicle dynamics, they can be presented either as rigid bodies or as rigid bodies with certain elastic properties. If eigenfrequencies of the car body structure in the empty or laden condition are within the range of the suspension oscillation frequencies (0–20 Hz), then elastic properties of the car body will influence the performance of the vehicle on track and should be taken into account when simulating vehicle dynamics. A typical example of elastic behaviour of a freight wagon is bending oscillations of an 80-foot long container flat [6], as shown in Figure 3.8. The situation of low car body frequencies appears more often for passenger coaches because reduction of tare mass makes the body more flexible. In passenger vehicles, the eigenmodes of the car body not only influence the ride performance but also affect passenger comfort, and standards therefore limit the value of the first eigenmode frequency to be higher than 8 or 10 Hz [7]. Research is being done by Japanese scientists [8] that is aimed to use piezoelectric technology to provide additional damping for eigenmodes of super-light car bodies. Other research tends to implement control in suspensions to dampen the car body’s eigenmodes as well as its rigid-body oscillations [9].

The car body is the part that establishes the existence of the railway vehicle in technical terms; it unites all systems, houses cargo or passengers to provide safety of transportation in all operational modes and bears the registration number of the vehicle as the only non-replaceable part. The car body rests on bogies (or suspensions) and interacts with them through car body to bogie connections. Adjacent car bodies are joined in a train via inter-car connections: coupling devices, passages and additional inter-car dampers to improve ride quality. The car body is used to position braking equipment.

Interaction of wagons and cargo is a separate and very important part of vehicle system dynamics. Semi-trailers have their own road suspension and oscillate while the train is in motion [10]. Liquid cargo sloshes inside a tank, changing its behaviour on straight track and in curves [11,12]. Wood or even bulk cargo can displace under the action of longitudinal impacts.

Interaction of vehicle and passengers is also important, producing much research on the perception of oscillations by people [13] and modelling of human bodies in different postures [14].
The principal difference between a railway vehicle and other types of wheeled transport is the guidance provided by the railway track. The surface of the rails not only supports the wheels but also guides them in the lateral direction. The rails and the switches change the rolling direction of wheels and thus determine the direction of travel of the railway vehicle. Irregularities on the rails produce oscillations of the vehicles.

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

- Transmission and equalisation of the vertical load from the wheels of the vehicle to the rails
- Guidance of the vehicle along the track
- Control of the dynamic forces due to motion over track irregularities, in curves, switches and after impacts between the vehicles
- Efficient damping of oscillations
- Application of 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. The traction and braking forces are transmitted through traction rods or axlebox guides (sometimes known as 'hornblocks') straight onto the wheelset. The typical example is the conventional two-axle wagon (see Figure 3.9), which generates larger forces in tight curves than the equivalent bogie vehicle; therefore, the length of the former 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. Typical examples are shown in Figure 3.10.

Early bogies simply allowed the running gear to turn in the 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 and traction equipment, suspension and...
FIGURE 3.9  Bogie-less two-axle wagon. (From Corporate Media People, Moscow, Russia. With permission.)

FIGURE 3.10  Typical examples of: (a) bogied wagon, (b) two-axle bogie and (c) three-axle bogie.
dampers. It may 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.

The following characteristics are used to describe the bogies:

**Track gauge:** To correspond to the track gauge of the railway and to provide appropriate wheelsets.

**Axle load and design speed:** These are used to check the compatibility of the bogie with the designed vehicle.

**(Bogie) wheel base:** Wheel base of the bogie is the longitudinal dimension between the centres of its outer wheelsets. This parameter is important for distributing the mass of the vehicles over the length of the track. The bearing capacity of bridges that are shorter than one vehicle is determined by the permissible load per unit length of the vehicle, that is, fully laden weight of the vehicle per one bogie divided by bogie wheel base.

**Car body support height over the top of rails:** The height is measured between the top of the rail and the centre bowl surface or the surface of the other elements that support the car body. In production, measurements are usually taken in free bogie condition and can vary depending on wheel diameter tolerance, suspension element tolerances, production tolerances of cast or fabricated frames and beams. After the car body is positioned over the bogie, the height reduces, depending on the suspension deflection under the tare weight. The parameter is important for safety to provide tare vehicle coupler height.

**Clearance diagram:** The bogie dimensions in cross-section laterally to the track are limited by the so-called clearance diagram. Besides requirements that limit the lateral dimensions of the bogies in the same way as the dimensions of the car bodies, there exist special limits on the vertical distance between the bottom of the bogie and on-track devices.

**Suspension deflection under empty/laden vehicle conditions:** The values of deflection are used to check whether the bogie is appropriate under the chosen vehicle and will provide safety limits of coupler height.

### 3.1.3 Couplers, Automatic Couplers and Draw Bars

Various devices are positioned in the end sections of the car body to provide connection between adjacent vehicles in a train, transmitting longitudinal tension and compression forces, providing damping of longitudinal train oscillations and allowing necessary angles of rotation and vertical and lateral displacements for safe passing of curves and humps. These same devices are working when the train is formed out of separate wagons or coaches, facilitating coupling and uncoupling operations when impacts between vehicles need to be handled by the coupling devices.

In general, the coupling devices can work either manually (see screw coupling design in Figure 3.11) or in automatic mode (see automatic coupler design in Figure 3.12). Manual operation is done by the railway staff who lift the screw coupling and connect it to the adjacent vehicle; thus, the screw coupling [15] weight and tension capacity are limited by human capability. Automatic operation is done either by running the vehicle off the sorting yard hump or by moving it by locomotive, with an impact onto the adjacent vehicle. The speed range during sorting is usually below 5 km/h. The special shape of the automatic coupler surfaces [16,17] guides the automatic couplers into their locking position. However, to unlock the automatic couplers, manual force is needed to raise the lock. The introduction of automatic couplers allowed the transmission of much bigger tension as well as compression forces in the train, thus providing for increases in train length and weight. It also speeds up the train sorting operations and provides much higher safety for railway staff who do not need to go between the vehicles during train forming operations.
Two adjacent automatic couplers have longitudinal slack (clearance) between the coupling surfaces that results from tolerances and wear in operation. In train starting mode, the slack allows the locomotive to start moving the vehicles one by one and not the whole train consist at once, thus providing energy efficiency. However, in train transition modes from traction to braking, the slack results in impacts between adjacent vehicles that increase accelerations and make ride comfort poor. Therefore, slackless couplers are often used in coaches instead of automatic couplers. In freight wagons, two or more of them can be connected by slackless draw bars if loading and unloading operations allow it.

Vehicles in one train can have different coupler heights due to wear of wheels or deflection of the suspension that can vary due to oscillations in motion. Therefore, the couplers need to have a possibility of relative vertical displacement and are characterised by a safety limit on coupler height difference.

To provide damping of the oscillations in a train, the screw coupling is usually used together with buffers (Figure 3.11) that work only in compression and provide elastic resistance and friction damping [18].
In automatic couplers, the special damping device named the draft gear is installed inside the centre sill (Figure 3.12). The draft gear is the damper with initial pre-compression that works in compression when tension or compression forces are applied to the coupler. The yoke transforms both tension and compression forces into compression of the draft gear. The main characteristic of the draft gear is its energy-absorption capacity under the impact conditions. Friction draft gears have the lowest energy capacity; higher capacity is provided by friction-elastomer draft gears, elastomer or even hydraulic ones.

### 3.1.4 Pneumatic Brakes

The major brake type for unpowered railway vehicles around the world is the automatic pneumatic brake, where the brake control signals are transmitted from the locomotive onto each vehicle in a train by changing the pressure in the pneumatic train line brake pipe. Each vehicle houses braking equipment such as the brake pipe, air distributor, spare tank, brake cylinder, brake rigging, brake shoes and an empty-load device, as shown in Figure 3.13.

When the pressure in the train line brake pipe increases in charging mode or brake release mode, then the air distributor charges the spare tank and connects the brake cylinder with atmosphere. When the brake pipe pressure decreases in brake application mode, then the air distributor connects the spare tank with the brake cylinder. The braking force from the shaft of the brake cylinder is transmitted through the brake rigging onto the brake shoes that press against the wheels in tread brakes. In disc brakes, the force is transmitted by callipers onto the brake pads that press against the brake discs. Friction between the brake shoes and wheel treads, or between the brake pads and discs, decelerates the vehicle.

![Figure 3.13](image-url)

**FIGURE 3.13** General composition of braking equipment on a freight wagon. (1) train line brake pipe; (2) air distributor; (3) spare tank; (4) brake cylinder; (5) brake rigging; (6) brake shoes; (7) empty-load device; and (8) end cock.
The term ‘automatic’ is used for this type of brake equipment because it automatically applies the brake when the vehicles in the train or the train line brake pipe disconnect in the case of an accident.

Pneumatic brake can be applied in normal operational mode to slowly decelerate the train or in emergency braking mode that corresponds to the same situation of an open mainline in the case of an accident.

The following characteristics are used to describe the vehicle brake system:

- **Shoe force**: The physical value of the force applied from the brake shoe to the wheel tread.
- **Friction coefficient**: The friction coefficient between the brake shoe or pad and the wheel tread or brake disc surface. The coefficient is non-linearly dependent on the vehicle speed and the materials of the friction surfaces.
- **Vehicle stop distance**: The distance that the single vehicle will travel from its initial speed to a full stop after the emergency brake is applied or the train line brake pipe disconnects. The stop distance of a single vehicle is measured in tests on straight horizontal track.
- **Train stop distance**: The distance that the train consisting of identical vehicles will travel from its initial speed to a full stop after the emergency brake is applied. Train stop distance is usually bigger than the vehicle stop distance because the pressure decrease wave needs to travel through the train to reach every vehicle. Train stop distance is highly dependent on track gradient.
- **Skidding**: Skidding between the braked wheel and the rail occurs when the brake force is bigger than the wheel-rail traction force. It increases the stop distance and produces wheel and rail defects.

Wheel-rail traction force depends on the actual axle load of the vehicle. This becomes important for wagons because the weight of the empty one can be five times lower than that of the laden one. To avoid skidding, the pressure in the brake cylinder needs to depend on the actual weight of the car body. Special empty-load devices control the brake cylinder pressure, depending on suspension deflection. The anti-skidding devices are sometimes introduced in passenger coaches, especially with disc brakes.

Braking is a source of longitudinal compression forces originating in couplers of vehicles in a train. Non-synchronous braking of vehicles makes the back part of the train run onto the front part, creating the compression forces. This problem is specifically important in long freight trains and is often dealt with by driving them using several locomotives distributed along the train [19].

When assessing derailment safety, it is necessary to simulate the situation when the heavy train passes through the curve and the emergency brake is applied that produces large quasi-static compression forces on the couplers that press the light (empty) vehicle out of the curve [20]. On straight track, the large compression forces can lead to various train buckling modes [21,22], and again, the assessment of derailment stability is necessary.

### 3.2 DESIGN OF CAR BODIES

#### 3.2.1 PASSENGER COACHES

To provide the requirements of stiffness, deformations and structural strength, the car bodies for passenger coaches bear the external loads from all parts, such as the frame with the floor, two side walls and two end walls and the roof (see Figure 3.14). Inside the car body, there are structures that do not bear main loads, such as internal walls that split the interior into compartments, toilets, staff rooms, etc.

Splitting the bearing structure of the car body by functions, it is possible to distinguish the top compartment (under the roof), the middle compartment (being actually the space utilised by
passengers) and the bottom part (where various items of equipment are installed). The top compartment houses ventilation and conditioning systems, water tanks and tubes. In the bottom part, the frame supports the power generator, batteries, refrigeration units, tanks for used water and other devices providing comfort. In some coaches, the bottom part is open; in others, it is covered with decks or can even be protected by a load-bearing structure to increase the car body stiffness.

Most common material for car bodies is low-alloyed steel. In the desire to make the car body lighter, to increase its reliability and durability as well as improve its external look, designers are constantly searching for new materials, designs and technology solutions. In high-speed trains, the aluminium alloys and stainless steel initially found application. Later, various non-metal materials such as plastics, fibreglass and carbon fiber reinforced polymer composites, multi-layer panels and honeycomb panels were implemented.

Aluminium alloys have better material strength and lower density than steel, do not corrode and are easy to cut and shape. Aluminium car bodies can dissipate much energy to deformation in conditions of an accident, thus providing better safety. At the same time, aluminium alloys have an elasticity modulus that is three times lower than that of steel, which makes the structures prone to stability loss under compressive loads and decreases their stiffness and modal oscillation frequencies. To override these problems, the aluminium-bearing structures need bigger sizes of cross-sections than steel ones.

The bearing structures of coaches can be classified into three types:

**Reinforced sheets:** The main bearing structure are the metal (steel, aluminium or stainless steel) sheets that are reinforced by beams with various shapes of cross-sections. Sometimes, to prevent the sheets from buckling, the corrugated design is used, or the sheets are reinforced with stringers. Beams and sheets are assembled with welding or rivets. Use of welding produces visible deformations of sheets that spoil the visual appearance of the car body.
Carcass systems: The main bearing structures are massive rods with open or closed cross-sections. External sheets just increase the general stiffness of the system. External sheets can be light alloys or plastics that provide a better visual appearance and do not need to bear significant loads.

Panel systems: Loads are transmitted onto the bearing panels that consist of external and internal walls joined together. Such panels are more often produced out of extruded aluminium profiles joined with arc welding or stir welding to form the floor, walls and roof. Extruded panels after stir welding do not have any surface defects and provide high structural strength and good visual appearance. In another case, honeycomb panels (sandwich systems) consisting of external and internal shells joined together with porous polyurethane or foam-aluminium or composite filling are used. Advantages of panels are their ability to damp vibrations and to provide sound and thermal insulation.

Let us consider the car body design with beam-reinforced metal sheets.

The frame (Figure 3.15) consists of the centre sill 1, transverse beam 2, end beam 3, side beams 4, lateral beams 5, lateral floor beams 6 and metal floor sheet 7 that has openings for various service pipes and ducts to pass through.

![Figure 3.15](image-url) Passenger coach frame. (1) centre sill; (2) transverse beam; (3) end beam; (4) side beams; (5) lateral beams; (6) lateral floor beams; and (7) metal floor sheet.
The roof (Figure 3.16) has uniformly spaced arches 1 that are covered with corrugated sheets 2 having a 2 mm thickness and having bent side sheets 3. The roof has air supply openings 5, hatch 4 and covers 6 and 7 that provide access to various equipment in the top part of the coach.

The side wall (Figure 3.17) is the welded whole-metal structure constructed out of corrugated and plane sheets with thickness of 2–2.5 mm supported by carcass bracing on the inside. The side wall consists of the top belt 1 and bottom belt 2, which are interconnected between each other with fabricated beams 3, and windows 4 and 5 comprising the middle belt. The wall is reinforced with profiled elements of the top beams 6, racks for the doors 7, vertical racks 8 and longitudinal beams 9.

FIGURE 3.16 Passenger coach roof. (1) Arch; (2) corrugated sheet; (3) side sheet; (4) hatch; (5) air supply; (6) and (7) covers.

FIGURE 3.17 Passenger coach side wall. (1) Top belt; (2) bottom belt; (3) middle belt with fabricated beams; (4) compartment window; (5) toilet window; (6) top beam; (7) rack for the door; (8) vertical rack; and (9) longitudinal beams.
The end wall (Figure 3.18) consists of the corner racks 1, roof arc 2 and two anti-impact racks 3, which are welded to the bottom of the frame end beam (see Figure 3.15) and to the lateral beam 4, that altogether provide safety for passengers in the case of accidents. The bearing structures are covered with metal sheets 5, additionally reinforced by ribs 6.

Years of operation have shown that relatively simple reinforced sheet structures of car bodies have one significant disadvantage. Their lower bending mode frequency is often less than 8 Hz, and they can come into resonance with suspension oscillations worsening the ride performance and passenger comfort.

In carcass-type designs, the coach frame consists of the centre sill, with side and transverse beams having box cross-sections. Corrugated sheets used in the bottom and floor further increase the stiffness of the whole system. The side walls of the car body are based on a grid of vertical and horizontal beams. Parts of the beams are narrow and higher and others are wider and thinner. This allows joining them by stamping thinner elements into the higher ones. Altogether, this provides a stiff and stable system. The shell is attached to the carcass with spot welding, thus providing a better visual appearance, as spot welding gives less buckling and deformation.

To increase the roof stiffness, the arches are installed. Longitudinal stiffness of the roof can either be provided with stringers or corrugated sheets of the shell.

General stiffness of a carcass-type car body is higher than that for sheet structures; however, the variability of elements makes the production more labour intensive, increasing the cost of the design. Spot welding needs special attention when applying the coatings.

The typical example of carcass design is the TGV train car body shown in Figure 3.19. The frame is not symmetrical because the example shows the end coach that couples to the locomotive when forming the train. The left part of the car body rests on the bogie through a transverse beam (see 7 on side A), and the right part (side B) rests on an articulated bogie that supports two adjacent car bodies simultaneously. Pay attention to the massive carcass of the side walls. The bottom part of the car body consists of frames joined with longitudinal elements, while the floor and the roof are corrugated sheets. Testing showed that the first bending frequency is higher than 10 Hz.
The panel type design allows to have an even lighter, stiffer and more reliable car body structure. The main idea is to use large extruded aluminium profiles for the external shell (see Figure 3.20). Extruded profiles are joined with automatic arc welding, followed by grinding the welds or by stir welding. The resulting panels have large bending stiffness and local structural strength to withstand point loads. This property is used to attach internal and external equipment. The main advantages are low production cost and a smaller number of components.

However, the extruded panels are often too strong and do not provide sufficient sound and thermal insulation; in such cases, they are replaced with sandwich or honeycomb panels.

A typical example of panel design is the Talgo train coach (Figure 3.20a). It uses extruded aluminium profiles for the floor and walls. The bottom part of the floor has guides that are used to support the equipment under it. Side walls have windows and doors reinforced with vertical beams.

By the end of the 1990s, much research work was aimed to choose the best design of car bodies for coaches and locomotives. The lightest car bodies with maximum stiffness and structural strength were found to be provided by soldered honeycomb panels that are, at the same time, the most expensive in production. Riveted airplane-type designs were tested as well and did not find practical application. Currently, the double-skin design constructed out of volumetric extruded aluminium profiles seems to have good prospects; however, it is mainly used in high-speed rollingstock.

### 3.2.2 Freight Wagons

The design of a freight wagon body is determined by its purpose [23–26]. Railway freight wagons are used to transport liquid, loose or bulky product, piece-freight goods, lumber, civil structures, equipment, pipes, wheeled and tracked vehicles, containers and items requiring cooling and maintaining at stable temperatures during transport. To transport each type of cargo, multi-purpose or special...
rollingstock is employed. When in transit, the cargo produces impact forces and can cause corrosive and abrasive wear in the vehicle, whereas the wagon body may contaminate or damage the cargo.

The main characteristics of the freight car body are the volume of the carried cargo and the availability of devices for loading, fastening and unloading of the cargo. Since the wagon is designed in a limited clearance diagram and with a minimum possible car length over the couplers, effective volumetric capacity of the car body should tend to reach the volume of the clearance diagram. The space between the cargo and the clearance diagram is used to accommodate the car body structure (see Figure 3.21).

The freight car body can be nominally represented by a box-section beam mounted on two supports, that is, bogies, and subjected to static and dynamic loads from cargo, bogies and adjacent

FIGURE 3.20 Schematic of various types of panel car body designs: (a) out of volumetric panels, (b) out of honeycomb elements, (c) riveted airplane type design and (d) honeycomb structure.

FIGURE 3.21 Using the clearance diagram to determine the car body structure. (1) clearance diagram, (2) transported cargo, (3) space to position the car body structure, (4) bottom clearance diagram, (5) space to position the coupling devices, (6) maintenance zone and (7) space to position the bogies and running gears.
The body structure shall have a minimum weight and a minimum cost with the given strength and stiffness.

The full diversity of freight car body designs can be divided into several basic types. Enclosed body designs are used in box wagons, hopper wagons, auto carriers, refrigerated wagons, etc. Typical examples are shown in Figure 3.22. A box car body design represents a closed box-section beam. In this type of body, cargo impacts the floor deck by bending the body like a beam lying on two supports, the underframe and the floor deck jointly receive loads in the same fashion as a bottom flange of a nominal beam, the side walls of the car transfer shear stresses and participate in the load transfer in the same fashion as beam webs and the roof with the upper chords.

**FIGURE 3.22** Wagons with enclosed car body: (a) box wagon, (b) hopper wagon with straight walls, (c) hopper wagon with round walls, (d) tank type hopper wagon, (e) dump wagon and (f) refrigerated wagon.
of the side walls functions like the top flange of the beam. The body can have either flat side walls, with the roof reinforced by posts and horizontal sills, or cylindrical side walls and the roof resting upon—two to three transverse partitions. In the latter case, stiffness and stability of plates are ensured by their cylindrical surface. Cylindrical wall designs have shorter welds, yet thicker plates and greater weight.

The lateral pressure from loose or palletised load is transferred to the sheathing and side wall posts, which redistribute it to the car underframe and roof. The underframe, as a rule, is sufficiently lightweight and can have a centre sill extending the entire length of the car or only two middle sills at the ends, where coupling devices are mounted.

The end wall taking longitudinal dynamic loads from cargo usually consists of a flat sheet and a case frame. Stamped corrugated sheets, light and robust, are also used without additional sills, the function of which is carried out by corrugations.

The main challenges faced in developing enclosed body designs are wide openings of sliding doors, protecting the lightweight structure of the floor decking, causing damage to cargo in transit and being able to withstand concentrated loads from loaders, as well as the connection of hopper car pockets with the underframe and location of discharge hatches.

Open body designs are used in gondola wagons with discharge hatches, solid-bottom gondola wagons and open hopper wagons, as shown in Figure 3.23. Owing to an open-top design, the upper chord of the side wall takes up compressive stress resulting from the bending of the car, for which, generally, a reinforced rectangular hollow section is used. The upper chord of the side wall also resists the load when the car is turned over during unloading on a car dumper and can also be subjected to impact loads from a grapple or excavator bucket. Owing to the open-top design, the side wall is attached to the underframe in a cantilever fashion, and the side wall posts resist the bending moment from the load lateral pressure. The largest bending torque is observed at the bottom of the posts and at the junction with crossbearers. An open-top design with internal transverse trusses is occasionally used, which receives a lateral pressure acting on the side walls and transfers forces from the side walls to the centre sill. Although such a technical solution makes it possible to reduce the body weight, it worsens unloading conditions for bulk cargo, in particular if frozen at low temperatures.

Gravity-based unloading of cargo without the use of a rotary dump or grapple can be carried out through discharge hatches made in the floor of the open body. The hatches can take up the entire area of the body, and the end and side walls can then be made vertical, with the maximum body volume attained. If the hatches only take up a part of the car surface area, the lower portion of end and side walls needs to be inclined to direct cargo into hatches. Although the use of the hatches enables the car to be unloaded at an unloading point without technical means, it increases the weight and cost of the body and reduces the volume of the carried cargo.

The main challenges faced in designing the open body are the provision of fatigue resistance of side wall posts to underframe attachments, strength of the underframe with large hatches under longitudinal dynamic loads and strength of the upper chord of the side wall during mechanised loading and unloading.

Load-carrying underframes are employed in flat wagons serving different intended purposes – for containers, for multi-purpose use and for the transportation of wood, large-diameter pipes, coiled and sheet steel, semi-trailer trucks, etc. The flat wagon body consists of an underframe and, when so required by the intended purpose, end walls, side posts, side and end swivel flaps, end platforms and cargo securing devices. As shown by the examples in Figure 3.24, the load can be placed above the underframe, partially lowered into special niches or placed in a well of the underframe.

The flat wagon underframe can have a load-bearing centre sill, load-bearing side sills or a combination of the two. Since the size of the loading area is limited by the upper part of the gauge profile and the top surface of the underframe, developers try to lower the loading surface of the underframe as much as possible. Therefore, in long-wheelbase flat cars, the centre sill and side sills of the underframe extend downwards to the lower part of the loading gauge profile. If the side
sills of the underframe are located above the bogies, the height of the sills is limited, and, in order to obtain the required stiffness and strength, complicated designs comprising plates of varying thickness are used.

The top surface of the underframe may have wooden or metal floor decking used for carrying various cargoes, including wheeled and tracked vehicles, and heavy loose goods, for example, crushed stone.

The flat wagon underframe can be fitted with removable or non-removable end flaps and side posts. They are used for stowing large-diameter pipes and timber. There are underframe designs featuring a middle truss to which pipes or other long cargoes are attached on either side.

Underframes with a deck lowered in the middle are used in well cars for large-size cargoes. Dimensions of the carried cargo do not permit having it placed above the bogies and fit into the loading gauge, so the provision for locating the cargo between the bogies at a minimum height above rail level is made in the car layout. Such an underframe has bends and a minimum height. In order to obtain the necessary strength and especially stiffness, the top and bottom plates of the underframe use the entire width of the car and are connected to one another by several vertical plates.

When designing flat wagon underframes, fatigue resistance and stiffness of sills of a limited height and attachment points connecting the sills into a single structure may pose difficulties.
Tanks are used for the transportation of liquid and powdery bulk cargoes. The tank body consists of a cylindrical hermetically sealed vessel used to carry the goods and an underframe or two half-frames that receive and transfer loads from couplers and undercarriages to the vessel. The material of the vessel and the thickness of the vessel case are determined according to cargo properties and conditions of carriage. Cargoes can be basically divided into cargoes transported under pressure such as compressed natural gas; cargoes unloaded under pressure, for example, acids and other chemical products; and cargoes transported and unloaded in unpressurised condition such as oil. The case thickness depends on the maximum pressure and can range from 6 to 25 mm. If cargo has a corrosive effect on the walls of the vessel or formation of corrosion

FIGURE 3.24 Wagons with load carrying underframes: (a) general purpose, (b) double stack containers, (c) containers and swap bodies, (d) steel rolls, (e) wood, (f) large radius pipes and (g) special cargo.
products leads to cargo contamination, the vessel is made of corrosion-resistant materials or provided with an internal coating.

In the upper or lower part of the vessel, special fittings for filling and draining cargo and monitoring cargo level and pressure are provided. The top or end face of the vessel is equipped with a hatch, providing access to the vessel, which can also be used for loading non-dangerous goods.

Examples of wagons with tanks are shown in Figure 3.25. The tank may have a separate vessel and underframe or have no centre sill. In a tank wagon with the separate vessel, longitudinal loads from coupling devices are received and transferred by the underframe. The vessel is subjected only to static and dynamic loading from cargo. In a tank without the centre sill, the vessel receives and

![FIGURE 3.25 Wagons with tanks: (a) with a frame, (b) with two sub-frames, (c) with insulation and (d) with insulation and heating.](image-url)
transfers all loads from couplers and undercarriages. A tank without a centre sill is usually lighter than centre-sill designs; however, the possibility of assembling vessel and underframe separately at different production facilities and then putting them together at the final stage of the rail car assembly attracts manufacturers.

For the transportation of goods solidifying at low temperatures, the vessel is provided with an insulating jacket. A cargo heating system can also be installed inside the tank or on its external surface to facilitate cargo discharge. For the transportation of liquefied gases at low temperatures, cryogenic vessels, that is, thermoses, are employed. Such a vessel consists of an inner case interacting with the cargo and an external case. The space between the cases is exhausted of air to incorporate vacuum insulation.

Tanks intended for the transportation of dangerous goods are equipped with protective emergency devices, as shown in Figure 3.26. Dished end heads of the vessel have shields, and fittings are protected by arcs or a strong shell. A relief valve is used to prevent pressure build-up in the vessel, and fittings are provided with safety shut-off valves for stopping or slowing the flow of cargo when the fitting is damaged.

The main functional characteristics of the tank, which determine the possibility and efficiency of the carriage of one or several goods, are the volume of the vessel, payload capacity, allowable pressure, material of the vessel and design of the loading/discharge fitting.

The most difficult tasks in designing a tank car are to maximise the volume of the vessel within a limited clearance diagram and car length, achieve the minimum weight of the vessel and ensure strength and fatigue resistance of the vessel to underframe attachment points and those of the vessel in the hatch area.

Combined structure car body designs are used for railcars, the loading and unloading of which involve displacement or detachment of body parts. The bodies of dump wagons consist of a main underframe and a separate freight body with side and end flaps, which in unloading turns about the longitudinal axis to an angle of 45°. When the body is turned, the side wall swivels to the position of the floor plane. Unloading can be done on either side of the track. The body is rotated by pneumatic cylinders with the pressure air supply through a separate main line passing along the train consist from the locomotive. This arrangement and other examples of combined structure car bodies are shown in Figure 3.27.

Flat wagons with a rotary flat deck are used to transport steel sheets up to 4.8 m wide. For loading and unloading of sheets, the flat deck is rotated to the horizontal position. In this condition, the width of a transported sheet and that of the flat deck is greater than the width of the vehicle clearance diagram. To bring the flat deck into the service position, the flat deck loaded with sheets is
rotated by pneumatic cylinders to an angle of 45° to 55°. In this position, the flat deck and cargo are within the clearance diagram. A similar operation principle is employed for flat wagons designed for the transportation of track switches; the only difference is a hydraulic drive used for rotating the flat deck.

The body of a wagon for the transportation of bulk chemical products with a lifting cargo reservoir comprises an underframe and the cargo reservoir with loading and unloading hatches. For unloading, the train consist with lifting bodies moves at a low speed inside an unloading trestle.
Rollers on the side walls of the cargo reservoir move on inclined ways of the trestle, the reservoir carrying cargo rises and rods connected to the underframe open discharge hatches. In addition, designs of an ore-carrying reservoir detachable from the underframe, which are turned over when the train consist enters the unloading trestles, are available.

Flat wagon underframes with a movable additional second top deck are used to transport automobiles. The height of the top deck is changed by means of a mechanical drive for loading automobiles of different heights.

In order to handle large-size heavy loads, for example, electric transformers, underframes of well cars consisting of two separate parts are used. The two underframe halves are attached to the carried cargo on either side, and, during transportation, cargo transfers loads as an underframe component. In the empty condition, the halves of the underframe are interconnected.

Swap bodies mounted on the flat wagon underframe are removed and separated from the underframe while being loaded or unloaded. Swap bodies can be stored in the loaded or empty condition, separately from the car pending unloading or sending for loading. The model range of swap bodies is customised for transportation, loading and unloading of various goods or one in particular. Such designs are used to transport coal, mineral fertilisers, grain and raw materials of woodworking industry. Since the swap body is not intended for road or water transport, it is developed in a vehicle clearance diagram; therefore, its dimensions and volume are much larger than the volume of a multi-purpose container having the same length.

To haul cargo, it is usually loaded at the shipper’s site and unloaded at the consignee’s site. The methods and features of loading and unloading determine a cargo vessel design. For the purposes of loading, securing cargo in transit and unloading, the freight car body is equipped with loading and unloading devices, as well as with cargo securing devices.

**Loading devices** in the form of hatches in the car roof ensure access of cargo during loading and provide integrity of the body and protection of cargo from atmospheric precipitation. The location and size of loading hatches generally correspond to cargo sources on a loading trestle. On some vehicles, the entire roof or parts thereof are shifted or removed to load cargo. When loading cargo with loaders, sliding doors in side walls or sliding side walls are used. In order to protect cargo from atmospheric precipitation, removable hoods or sliding soft tents on arches can be used. In tank cars, a loading device ensures the supply of liquid cargo into the vessel. Auto carriers are loaded and unloaded through doors made in end walls, using stationary ramps and a fold-away crossover platform, allowing automobiles to move through rail cars along the entire consist.

**Unloading devices** can be implemented in the form of unloading hatches intended for gravity-based unloading. Hatches can be made in the floor, in walls or in special pockets at the bottom of the body, providing the supply of cargo to a hatch. Unloading hatches, as a rule, have a driving unit for opening and closing, as well as seals to prevent load spills. The drive of unloading hatches can be actuated manually, by a pneumatic cylinder connected through an external mechanical rotor, in lifting or turning the body on the unloading trestle. On some types of rail cars, such as an automatic-discharge hopper car, an unloading mechanism that allows to control the volume and direct the discharged cargo is used. Unloading of bulk goods can be done using pneumatic vacuum haulage systems installed at the bottom of the body.

Unloading of tank cars is carried out through fittings and pipelines mounted on the top or bottom of the vessel.

**Load-securing devices.** Since the car is subjected to dynamic loading during movement and shunting operations, all cargoes other than liquid, loose and bulk shall be attached to the vehicle by special devices. Swivel and fixed fitting lugs are used to fasten containers of different lengths to flat cars. Coiled steel carried on flat cars is stowed between transverse or longitudinal sections.
Automobiles in special auto carriers are attached to the floor by wheel retainers. Pipes are pulled together using soft-coated tie-downs to prevent longitudinal displacement and damage to a polymer coating. Large machines and equipment are secured to special brackets with restraining straps. Wooden or combined decks of box cars and flat cars are fitted with nailed wooden lugs for fixing cargo. To secure timber lading above the body level, wooden stakes are inserted in special rings on the car sides and underframe.

**Materials.** The most common material used for the production of freight car bodies is a low-alloy steel. Such materials have a high yield strength from 290 to 390 MPa, depending on the thickness, chemical composition, ductility and good weldability, and can be used to manufacture bent and stamped parts. Corrosion resistance of low-alloy steel increases the body service life, when protected by paint-and-varnish coating, to 40 years.

The use of high-strength and more expensive alloy steels in the freight car body makes it possible to reduce the weight only for the elements resisting extreme tensile, compressive or bending loads, mainly under impact conditions. Such elements are centre sills and an end wall frame, and a top sill of the side wall of a gondola wagon. Other body elements receive multi-cycle loads, and, in order to reduce their weight, fatigue resistance characteristics of both the material and welds need to be improved. However, with an increase in the yield strength of steel, fatigue resistance of welds does not improve, or such improvement is insignificant. Therefore, high-strength steels have not become widespread in freight car bodies.

Since pressure vessels of tank cars for handling gases are charged with an internal pressure of up to 30 kPa, and the thickness of their case amounts to 24 mm, the use of a high-strength steel helps to reduce the wall thickness and increase payload capacity.

If the cargo, for example, coal or ore, causes friction during loading and unloading, steels of grades with increased abrasion resistance can be used in body plates that are in contact with the cargo.

Pure steel grade is used for tank car vessels carrying sulphuric acid and for the plates made from this material, and welds are not susceptible to corrosion when in contact with cargo.

Corrosion-resistant steels can be used for bodies that are exposed to corrosive, chemically active cargoes. Since such steels are four to five times more expensive than low-alloy steels, the extent of their application is usually kept at a minimum level to cut back on the cost of the structure. To this end, tank cars only have the vessel made from corrosion-resistant steel, whereas hopper cars only have the body sheathing as such. To reduce the cost, clad steels consisting of a layer of low-alloyed or carbon steel, which is 6 to 12 mm thick, and a layer of corrosion-resistant steel, which is 0.5 to 2 mm thick, are used.

Reducing the vehicle body weight by using aluminium alloys is viewed as a highly promising approach. Weldable aluminium alloys are used in sheathing and the case frame of the bodies of gondola wagons, hopper wagons, box wagons and tank wagons. Since the aluminium alloy structure is lighter but more expensive than steel, the combined body designs are considered optimal. In such bodies, highly loaded elements, for example, underframes, are made of steel, whereas less loaded but sizeable elements, such as sheathing and the case frame of side, end walls and roof are made of light alloys.

Since aluminium materials with a low content of alloying elements are resistant to aggressive chemical products, they are used to manufacture tank car vessels for the transportation of acids and other chemical cargoes.

The use of more expensive light alloys to replace steel is effective for freight car bodies of large volume and carrying capacity. Therefore, aluminium structures are widely used for railway cars with 32 t axle load and on a limited basis on those with an axle load under 25 t. High corrosion resistance to mineral fertilisers and compatibility with food products contribute to improving the effectiveness of Al-alloy. Therefore, aluminium alloys are used for bodies
hauling mineral fertilizers and foodstuffs, for example, grain. Coating is not applied on the internal surface of the vessel carrying cargo.

Composite and polymeric materials, for example, fibreglass, are applied in freight car bodies in a limited way, mainly for loading hatch covers of hopper wagons, roof panels of box wagons, as well as outer case embracing thermal insulation. This is due to their high cost and low strength and stiffness.

Low-alloy and carbon steel structures are susceptible to corrosion when exposed to atmospheric humidity, and as a result of the transportation of cargo, exterior and interior surfaces of the vehicle body are adequately coated. To protect the interior surface of the body from chemical products or increase corrosion protection for foodstuffs, special coatings are applied. Vessels carrying acids are provided with internal rubber lining plates installed using an adhesive.

Connections. Electric arc welding is the most commonly used type of connection of steel parts within the vehicle body structure. Welded joints have the same yield strength as a steel plate, yet their fatigue resistance parameters are 2 to 2.5 times less. Therefore, in structural designing, welds are located in areas with reduced cyclic loads, or the weld area is increased with the use of additional parts to reduce local cyclic stresses. For welds of the vessel case designed to carry liquefied gases, weld ductility and high fracture energy shall be obtained throughout the entire range of operating temperatures. Obtaining such qualities for welds of high-strength steel plates can turn out to be problematic in low-temperature environments with temperatures under $-40^\circ$C and down to $-60^\circ$C.

For Al-alloy structures, both arc welding and friction stir welding are used. Friction stir welding provides a connection with a higher strength and fatigue resistance than those of the parent material, which is also less costly.

Hot riveting is also used to join the parts. Riveted connections are less durable and somewhat more expensive than welded joints, yet they have high robustness and maintainability and are used to fasten parts damaged or worn in service. To connect light-alloy lining sheets with the light alloy or steel case frame, lock bolts are used. The lock bolt connection is strong and has a higher fatigue safety and maintainability but is more expensive than the weld.

Owing to the fact that transported goods have different physical and chemical properties and that conditions of carriage and loading for different cargoes are widely varied, the number of vehicle body designs amounts to several hundreds. Thus, to create new models of rollingstock with a competitive edge over those in service, each car must be designed individually. For the most common types of rail cars, there are conventional time-proven designs in place, and they can be used for new models with technical characteristics similar to those previously produced. However, putting new bogies into operation with increased axle load, as well as the need for a higher payload capacity of a vehicle with a limited axle load, requires a redesign or complete review of the conventional body structure.

### 3.3 RUNNING GEARS AND COMPONENTS

Traditionally, wheelsets provide running support of railway vehicles and their guidance along the rails. In a traditional wheelset, the left and right wheels are rigidly connected to each other via the axle; the axle and wheels work together and are considered as one unit. The wheelsets are guided by normal and tangential forces between profiled surfaces of wheels and rails that act on either the wheel tread or wheel flange.

Interaction between wheels and rails has proved to be able to efficiently support axle loads up to 37.5 t (Australia, Brazil, South Africa) for freight wagons or running speeds up to 550 km/h (experiments in China and France) for passenger vehicles with light axle loads. At the same time, malfunctions in design or maintenance of wheel-rail interaction can lead to fast degradation of the elements and poor economy or even insufficient safety of the railway system.
3.3.1 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 following:

- 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 onto the rails to accelerate and decelerate the vehicle

The design of the wheelset depends on the following:

- 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 3.28. 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 the wheel profile.

In curves, the outer rail will be of 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 the 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 $\gamma$ to the axis of the wheelset (see Figure 3.29). The position of the contact point when the wheelset 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 that 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 the track. Railway administrations normally specify allowable wheel profiles for their infrastructure and the degree of wear permitted before re-profiling is required [27,28,29].

Figure 3.30 shows several examples of new wheel profiles. For understanding the dynamic behaviour of a railway vehicle, the conicity of the wheel-rail 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 to 135 mm, and flange height for vehicles is typically 28 to 30 mm. The flange inclination angle is normally between 65° and 70°. In the vicinity of the tape circle, the
\( \gamma \) is 1:10 or 1:20 for common rollingstock. For high-speed rollingstock, the \( \gamma \) is reduced to around 1:40 or 1:50 to prevent hunting. It can be seen from Figure 3.30 that the wheel profile has a relief towards 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 rollingstock or heavy haul freight wagons, are not conical but designed instead from a series of radii.
FIGURE 3.30 Common wheel profiles: (a) for freight and passenger railcars (Russia), (b) for freight railcars (China), (c) for industrial rollingstock (Russia), (d) for European freight and passenger railcars, (e) and (f) for high-speed trains (Japan), (g) for freight railcars (India) and (h) for high-speed railcars (Russia). (From Iwnicki, S. (Ed.), Handbook of Railway Vehicle Dynamics, CRC Press, Boca Raton, FL, 2006. With permission.)
that approximate a partly 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 as 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 [30]. This is the ratio of the rolling radius difference to twice the lateral displacement of the wheelset:

\[ \gamma_{eq} = \frac{\Delta R}{2y} \] (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 in operation may alter significantly from the initial design shape, 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 3.31) 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 (named hollow wear), damaging stresses may arise at the outer side of the wheel and rail and this is known as false flange damage. Flange wear may lead to increase of the flange angle and reduction of the flange thickness. 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 3.32. 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 the vicinity of the contact patch become very similar.

Many researchers design the optimal combinations of wheel and rail profiles that will provide low wear of both of them as well as reduce the rolling contact fatigue [29,31,32].

3.3.2 Axleboxes and Cartridge Type Bearings

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 onto the other bogie elements. Axleboxes are classified according to:

- Their position on the axle depending on whether the journals are outside or inside the wheel
- 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 bearings (see Figure 3.33) consist of housing 1; bearing 2, which is usually made out of alloy with low friction coefficient (e.g., bronze or white metal); bearing shell 3 that transmits the forces from the axlebox housing to the bearing and a lubrication device 4 that lubricates the axle journal. Front and rear seals 5 and 6 prevent dirt and foreign bodies entering.
the axlebox, while 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 rollingstock, though their use is still rare.

Axleboxes with roller-type bearings (examples shown in Figure 3.34) are classified according to:

- The bearing type (cylindrical, conical, spherical)
- The number of bearing rows in one bearing (two or one row)
- The fitting method (press-fit, shrink-fit and bushing-fit)

The main factor that determines the construction of 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 3.34a). Experience in operation of railway rollingstock 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 3.34d and e) 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 with the radial roller bearings and therefore generate more heat. This produces higher requirements for seals and grease in conical bearings.

Spherical bearings have not been widely applied due to their high cost and lower weight capacity, although they have a significant advantage in providing better distribution of load between 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 rollingstock often has three bearings in the axlebox – two transmitting radial forces and one (often a ball bearing) working axially (see Figure 3.35).
Recently, cartridge-type roller bearings gained wide application for higher speeds and axleloads due to many advantages:

- Resistance to combined axial and radial loads that guarantees increased mileages between repairs
- High accuracy in production that eliminates clearances and reduces vibrations at high rotation speeds
- Compact design with smaller external dimensions and mass
- Cartridge-type design that does not need disassembling during maintenance but is replaced as one piece
- Internal seals, grease and clearances are pre-set during the production and thus greasing and adjustment in operation are excluded

The most common cartridge type bearing (Figure 3.36a) consists of the two-row basic bearing: two rows of rollers inside the polymer cages, two cones separated with a spacing ring, one cup on the outside, seals and grease. The basic bearing is a press-fit onto the axle journal having the backing ring with a seal and is then fastened with a washer and bolts (Figure 3.36b). The cup is used as a seat for the adapter.

### 3.3.3 Wheels

Wheels and axles are the most critical parts of the railway rollingstock. Mechanical failure or exceedance of design dimensions cause derailment. Wheels are classified into solid, tyre and assembly types, as shown in Figure 3.37.

Solid wheels (Figure 3.37a) have three major elements: the tyre, the disc and the hub, and they mainly differ in the shape of the disc.

Tyred wheels (Figure 3.37b) 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 desirability of reducing wheel-rail interaction forces by reducing the unsprung mass has led to development of resilient wheels (Figure 3.37c) that incorporate a layer of material with low...
FIGURE 3.36 Cartridge type roller bearing: (a) bearing and (b) its application with adapter.

FIGURE 3.37 Major types of railway wheels: (a) solid wheels, (b) tyred wheels and (c) assembly wheels. (From Iwnicki, S. (Ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)
Design of Unpowered Railway Vehicles

elasticity modulus (rubber or 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, an independently rotating wheelset inevitably eliminates the majority of wheelset guidance forces. Such wheelsets have found application either on variable gauge rollingstock, providing fast transition from one gauge width to another, or on urban rail transport, where a low floor level is necessary.

3.4 DESIGN OF FREIGHT WAGONS BOGIES

In most cases, freight wagons use two two-axle bogies per vehicle. Articulated freight wagons can have three or more two-axle bogies, with two neighbouring frames resting on one bogie between them.

The majority 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 3.38 that central suspension makes up approximately 6% more of the designs than axlebox suspension. Some wagons use double suspensions similar to passenger bogies to reduce track forces or improve isolation of the load from excess vibrations.

3.4.1 THREE-PIECE Bogies

Bogies with central suspension are common in the countries of the former USSR, the USA, Canada, China, Australia, South America and most countries in Africa, thus outnumbering axlebox and double-suspension bogies by tens of times. Examples of the classical CNII-H3 (type 18-100) Russian bogie and the Barber bogie from the USA are shown in Figures 3.39 and 3.40, respectively. Such bogies are often termed ‘three-piece’ bogies [33].

The frame of a three-piece bogie consists of the bolster and two sideframes that are elastically connected by a coil spring and friction wedge-type central suspension system, which, besides other functions, resists asymmetrical loads and holds the bogie frame square in plane. Such suspension allows independent pitch of the sideframes when passing 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 bearings 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 bearings, the gravitational force providing recovery to the central position. In curves, the car body contacts the side bearings.

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 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 of empty wagons 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 and multi-linear vertical force characteristics.
The sideframes 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 pedestal of the sideframe. Clearances between the adapter (or the axlebox) and the sideframe 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. Owing to the absence of a primary suspension, such bogies have a large unsprung mass, which causes increased track forces on short irregularities or rail joints.

In curves, the three-piece bogies demonstrate the ‘lozenging’ or ‘warping’ effects, when the two sideframes 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. This leads to constant contact between the high-rail flange of the leading wheelset and the rail, causing high levels of wear [34].

Since classic three-piece bogies were introduced in the 1930s, there have been many advances in their design that deal with the known disadvantages and are explained in detail in [35–37]:

- Increase in suspension deflection under the tare wagon as well as under the fully loaded one
- Wedge designs that increase the warping stiffness by spatial configuration of inclined surfaces [38]
- Introduction of elastic pads in primary suspension to provide steering of wheelsets and reduce unsprung mass
- Installation of constant contact side bearings that have an elastic element compressed by the weight of the car body and provide yaw and roll resistance to the car body [39].

### 3.4.2 PRIMARY SUSPENDED H-FRAME BOGIES

The Y25 (and similar bogies such as the Y33) are predominantly used on European freight vehicles [40]. An example of this bogie is shown in Figure 3.41.

**FIGURE 3.41** The Y25 bogie: (a) general view, (b) primary suspension schematic (Lenoir damper) and (c) elastic side bearing schematic. (1) Wheelset; (2) rigid H-shaped frame; (3) braking leverage; (4) centre bowl; (5) side bearings; (6) suspension springs; and (7) axlebox. (From Iwnicki, S. (Ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)
The Y25 bogie has a single-stage primary suspension consisting of a set of pairs of nested coil spring (with a bi-linear characteristic for tare/laden ride) and a Lenoir link friction damper (Figure 3.41b) 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 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 previously. The centre bowl has a spherical surface to reduce asymmetric forces on the frame and elastic side bearings without clearance resist the body roll motions (Figure 3.41c).

Advances in primary suspended freight bogies [37] include:

- Using progressive rubber springs or bell springs instead of coil springs and Lenoir damper
- Introducing the additional pusher spring into the Lenoir damper or using double Lenoir links on each of the axleboxes to reduce longitudinal stiffness and facilitate curving
- Using wedge type friction damper with four rows of springs

### 3.4.3 Double-Suspension Bogies

In the 1980s, the UK had developed a novel, track-friendly bogie using passenger vehicle technology. The LTF25 bogie is shown in Figure 3.42 and is described in [41].

The LTF25 bogie was specifically designed to reduce dynamic track forces, and, as part of this, effort was made to reduce the unsprung mass. Small wheels (813 mm diameter) were used and inside axleboxes, giving a 30% reduction in wheelset mass, although this necessitated the use of on-board hotbox detectors. Primary suspension is through steel coil springs, and secondary suspension is through rubber spring elements and hydraulic dampers. The high cost of the LTF25 bogie and concerns about axle fatigue with inboard axleboxes militated against its adoption, but a modified version known as the TF25 bogies (with suspension reduced to only primary) have achieved considerable production success.

![LTF25 bogie](image-url)

**FIGURE 3.42** LTF25 bogie. (From Etwell, M.J.W., *J. Rail Rapid Transit*, 204, 45–54, 1990. With permission.)
The more recent RC25NT bogie shown in Figure 3.43 has horizontally soft rubber bushes in the primary suspension and flexicoil dual rate springs with friction damping via Lenoir link in the secondary suspension [42]. The bogie is equipped with disc brakes and inter-axle linkages that we discuss in the next section.

### 3.4.4 Auxiliary Suspensions (Cross-anchors, Radial Arms, etc.)

Many novel bogie designs address the fundamental conflict between stability on straight track and good curving [43]. 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, as shown in Figure 3.44, are called radially steered bogies. Such designs have small angles of attack, which leads to significantly decreased 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 3.1. The bogies may be split into three groups depending on the control principle used:

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

The first two groups in Table 3.1 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 the 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 or electric actuators (or a combination of these) are called actively controlled bogies. These are considered in detail in Chapter 15.

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

An example of a freight bogie with passive control using diagonal links between the axleboxes designed by Scheffel [44] is shown in Figure 3.45.

The second group of radially steered bogies are those with wheelsets that are self-steering in curves. The design of such bogies is based on selecting the optimum shear and bending stiffnesses [45]. This may be aided by using designs that allow these stiffnesses to be de-coupled.

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 [46], two of which are shown in Figures 3.46 and 3.47.

The Scheffel radial arm bogie is shown in Figure 3.46, and the generalised bogie stiffnesses for it have the following expressions:

Shear stiffness:

$$K_{\Sigma} = 2k_y + K_s$$

(3.2)

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

Bending stiffness:

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

(3.3)

where \(k_x\) is the longitudinal stiffness of the inter-axle linkage (per side) and \(K_b\) is the bending stiffness provided by the bogie frame. \(2b\) is the distance between axle journal centres.

Thus, the expressions for \(K_S\) and \(K_{b\Sigma}\) contain two independent parameters \(k_x\) and \(k_y\) that allow optimum shear and bending stiﬀnesses to be selected. Scheffel bogies have the axle load of 32 t and provide mileage between wheel turning of up to 1.5 million kilometres, thus proving the high efficiency of the design to reduce track forces.

Shown bogie designs are based on the three-piece bogie consisting of a bolster and two side-frames. They 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 also by the inter-axle links. In order for the inter-axle links to be effective, the bogie must have low longitudinal and lateral primary suspension stiffnesses. These bogie...
designs are therefore effectively double suspended. Infra-Radial [47] and SUSTRAIL [48] projects have shown that links can be effective with primary suspended bogies as well.

3.5 DESIGN OF UNPOWERED BOGIES FOR PASSENGER COACHES

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 [49].

For passenger bogies, the wheelsets are generally mounted in a rigid H-shaped frame that splits the suspension into 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 that 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 the lateral direction is designed to be 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 3.48. The secondary suspension swing consists of the secondary springs and dampers 2 and spring plank 1 that is attached to the bogie frame 3 by swing hangers 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 bolster 5 through bogie centre 6 and part of it is transmitted through side bearings 7. The bogie centre serves as the centre of rotation and transmits the traction forces, whilst the side bearings 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.

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 horizontal direction. Examples include the ETR-500 bogie (Figure 3.49), which uses Flexicoil secondary springs, and the Series E2 Shinkansen (Figure 3.50), 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 be directly mounted on the secondary suspension, as in the bolster-less bogies in Figures 3.49 and 3.50. In bolster-less bogies, the traction forces are transmitted through the centre pivot arrangement, and the bogie rotates under the car body by using the flexibility of secondary suspension in longitudinal direction. In such designs, yaw dampers are often fitted longitudinally between the body and the 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 3.48 Bogie with swing link secondary suspension. (1) Spring plank; (2) springs and dampers; (3) frame; (4) swing hangers; (5) bolster; (6) central column support; and (7) side bearing. (From Iwnicki, S. (Ed.), Handbook of Railway Vehicle Dynamics, CRC Press, Boca Raton, FL, 2006. With permission.)](attachment:image.png)

Various types of elastic elements are used in the primary suspensions of passenger bogies. To achieve high speeds, the longitudinal stiffness of the 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 stiffnesses 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 3.51), coil springs with guide posts (Figure 3.52) and chevron (rubber interleaved) springs (Figure 3.53).

The ETR-460 bogie (Figure 3.51) 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 3.52). 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 3.53). 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 a 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.

3.6 DESIGN OF INTER-CAR CONNECTIONS

Inter-car connections are intended for performing the following functions:

- Connect cars to locomotives and to each other; keep them at a certain distance from each other; and take up, transmit and soften tension and compression forces during the train movement and manoeuvres
- Constrain relative displacement of cars during the train movement in order to improve running smoothness and passenger comfort, as well as to prevent cars from piling up on each other in case of accidents
- Protect passengers passing between cars from environmental impacts

Devices performing the first function are named draw-and-buff gears. Present-day draw-and-buff gears usually include couplers providing inter-vehicle connection and holding strings of cars together, reversive shock absorbers (draft gears) providing energy absorption and return to the initial state, lugs transmitting loads to the car frame and additional irreversible protective devices for taking up excessively large forces, which are intended to prevent cars from piling up on each other in case of emergency.

3.6.1 SCREW COUPLERS

Screw couplers are used in Europe and in some countries of Asia and Africa. They include a hook with a shock absorber, a turnbuckle and two buffers (Figure 3.54). Transmission of draft forces is carried out from one hook to another via the turnbuckle. Compression forces are transmitted through buffers 4; screw coupler 2 is loose and does not affect transmission of compression forces.

Figure 3.55 represents the screw coupler hook with the shock absorber. Hook 1 is connected to thrust plate 4 by means of pin 2 through shackles 3. Draft gear 6 is located between thrust plates 4 and 5. Thrust plate 5 is fixed on the headstock of the car frame, and when draw forces

![Figure 3.54](image-url)
The turnbuckle (Figure 3.56) consists of two nuts 3 and 5 that are screwed onto screw 6. On one side, the thread is right-hand; on the other side, the thread is left-hand. The screw itself is turned by means of handle 4. When the handle is turned, the nuts converge or diverge. Bent coupling link 7 that is placed on the hook of the adjoining car is fixed on trunnions of the right nut. On the trunnions of the left nut, two shackles 2 are installed, and pin 1 passing through the slot in the car hook is installed on the ends thereof.

### 3.6.2 Automatic Couplers

Automatic couplers, in their turn, are divided into non-rigid (allowing reciprocal displacements), rigid (not allowing reciprocal displacements) and semi-rigid (having the possibility of reciprocal displacements only within the clearance limits).

Non-rigid and semi-rigid couplers are applied to freight and passenger cars; rigid couplers are applied to high-speed passenger cars, underground cars and electric trains where electric train-line connections are required. In freight cars, non-rigid automatic couplers SA-3 (for track gauge 1520 mm) and AAR type E (for track gauge 1435 mm, North America, India and Japan) are the most widely used. Their devices are similar, though they have some differences.

The automatic coupler SA-3 (Figure 3.57) includes coupler shank 1, coupler follower 2, draft gear 3, rear 4 and front 5 draft lugs, coupler yoke 6, striker 7 and draft key 8. The front and rear draft lugs as well as the striker are rigidly connected to car centre sill 9. Compression and tension forces are
transmitted in the following way. Under compression, the force coupler head 1 is transmitted via its shank onto coupler follower 2 onto draft gear 3 and further onto car center sill 9 via the rear draft lugs 4. Under tension, the force from coupler head 1 via draft key 8 (in some designs, via the pin) is transmitted onto coupler yoke 6 and from the coupler yoke, via draft gear 3, coupler follower 2 and front draft lugs 5, onto car centre sill 9. In the case of a large impact force, at exhaustion of clearance between type F coupler head 1 and the striker, the force is directly taken up by striker 7. In order to provide curving, the coupler is suspended by means of two pendulum-type suspensions 10 and centring beam 11; they ensure the coupler turning when rounding a curve and its return to the initial state.

As for the American non-rigid type E coupler represented in Figure 3.58, forces thereon are transmitted in a similar way; the differences involve the draft key position (horizontal) and the absence of the centring device.

The American semi-rigid automatic coupler is represented in Figure 3.59. Forces thereon are transmitted in a similar way. The differences involve the elevating stop at the bottom of the automatic coupler head, the spherical end of the coupler shank, the vertical position of the pin connecting the automatic coupler to the coupler yoke and the spring-loaded transverse strap that provides the possibility of turning in the vertical plane for the automatic coupler.

Among rigid automatic couplers, couplers using the bell-and-hopper arrangement connection, which minimises the end play, have become the most widely spread (Scharfenberg patent, 1903). There are also other couplers providing connections of electric and pneumatic trainlines: Tomlinson (the USA), wedge lock couplers (Great Britain), the GF-type conical coupler (Belgium and Switzerland) and zero-clearance couplings BSU-3 (Russia).

In recent years, requirements of energy absorption management in case of train collisions (Crash Energy Management) have been imposed for automatic couplers. Several matched energy-absorbing components, which are integrated into both the coupler, and the car body are used for this purpose.
Components for non-reversible absorption of energy (deformation tubes) and devices preventing cars from climbing up on each other are installed in addition to draft gears of the reverse type.

In Figure 3.60, the modular structure elements of the rigid automatic Voith Turbo Scharfenberg couplers are presented. The coupler includes coupler head 1 with the locking device and the signal transmission unit, deformation tube 2 and lug 3 with resilient damping device 4.
3.6.3 Draft Gear

The draft gear is a device behind the coupler for absorbing longitudinal forces and energy dissipation, which occur during transient modes of a train’s movement, impacting in the course of shunting operations, automatic train shunting, etc. The kinetic energy of interacting car is transformed into potential energy of its resilient elements and work of friction forces. Dry friction, viscous friction and internal friction are used for energy dissipation. Draft gears containing only resilient elements are called spring draft gears. Draft gears using dry friction are called friction draft gears. Draft gears using viscous friction are called hydraulic draft gears. Draft gears using internal friction in material are called rubber or polymer draft gears. Draft gears using compressible bulk polymer of high viscosity as the working mechanism are called elastomeric draft gears.

Performance indicators of draft gears are estimated by means of the force characteristic, which is the relationship between the force affecting the draft gear, its displacement $x$ and the speed $v$. The force characteristic is usually represented in the form of dependences: for the compression stroke ($v > 0$) and for the rebound stroke ($v < 0$) (Figure 3.61).

The work of the force on the route equal to the full travel of the draft gear is called the energy-absorption capacity of the draft gear. It is numerically equal to the area of the diagram under the straight line AB:

$$E = \int_0^x P(x, v)dx$$

After an impact, the draft gear is to return to its initial state. In this case, a part of energy returns. The returned energy is determined by the area under the line CD:

$$E_R = \int_0^{x_{max}} P(x)dx$$  \hspace{1cm} (3.4)
The irreversibly absorbed energy is characterised by the energy absorption factor:

\[ \eta = 1 - \frac{E_R}{E} \]  

(3.5)

In addition, the following parameters of the draft gear are used:
- \( X_0 \) is the full travel;
- \( P_0 \) is the initial tightening force;
- \( P_{st}^{\text{max}} \) is the maximum resistance force at static loading;
- \( P_{dy}^{\text{max}} \) is the maximum force at dynamic loading.

**Friction draft gears.** The wedge thrust is the most common way of pressure generation on friction surfaces (Figure 3.62) [50]. It makes it possible to impose large normal loads on friction surfaces and change them by means of changing the angles of inclination.

Thrust cone 1 takes up forces and transmits them to three wedges 2 providing pressing to hex-shaped nozzle 3 and case 4. The return is carried out by spring 5. The energy-absorbing capacity of such draft gears depends on the breaking-in and wear degree of wedges. That is why, in recent years, ceramic metal plates are applied as friction pairs, and polymer materials are applied as resilient elements in friction draft gears. The design of present-day friction draft gears is shown in Figure 3.63.

**Rubber-metal and polymer draft gears.** Specific properties of rubber as a material, which combines elastic behaviour and relatively large internal friction, are the cause of rubber-metal draft gears application in passenger cars. Structural designs of rubber-metal draft gears are varied. In trial designs, rubber works in shear and in compression with shear. In most cases, rubber-metal draft gears, wherein rubber works in compression, are applied in actual practice.

The disadvantages of rubber-metal draft gears connected with the specific properties of rubber as an engineering material are partly eliminated in polymer draft gears, wherein resilient elements from polyether thermoplastic elastomer are used, as shown in Figure 3.64. Their application makes it possible to considerably increase the energy-absorbing capacity of draft gears.

**Hydraulic and hydraulic-gas draft gears.** They use liquid flow through narrow calibrated orifices for energy dissipation. The main difficulty is caused by transmission of quasi-static loads. For this purpose, resilient elements, such as a spring or pressurised gas, are required in the draft gear. Despite the positive test results for their trial models, they have not become widely spread because of the high cost and low reliability. Information related to these draft gears is presented in [51].
Elastomeric draft gears. Nowadays, on the strength of their technical and economic parameters, draft gears wherein compressible bulk polymer (elastomer) is used as the working medium are considered the most promising. Elastomers, which have the compressibility of about 15% to 20% at the pressure of 500 MPa and high viscosity, perform the function of a spring and viscous fluid. As for their energy characteristics, they are close to hydraulic draft gears; however, their cost is lower and design is simpler.

In the course of loading, the piston movement causes pressure increase; as a result of elastomer bulk compression, additional resilient elements become unnecessary for the transmission of static loads and return to the initial state.

The elastomeric draft gear ZW-73 (Figure 3.65) contains hollow cylindrical case 1 closed with bottom plate 2 on one end. Elastomer shock absorber 3 consisting of cylindrical body 4 filled with elastomer, and rod 5, are partly located inside cylindrical case 1, with the possibility of moving with respect to it. At one end of rod 5, there is piston 6, which divides the working chamber of cylindrical body 4 into two chambers 7 and 8, with the possibility of the elastomer mass flowing from one chamber to the other through calibrated slot 9 between the lateral face of piston 6 and the interior face of cylindrical body 4. Rod 5 rests with its other end on bottom plate 2 of cylindrical case 1. Elastomer...
shock absorber 3 is pressed by coupler follower 10, which is connected with the draft gear case by means of bolts 11 and mounting strips 12, with the possibility of moving with respect to it.

The draft gear operates as follows: When affected by the compression load transmitted through coupler follower 10 to elastomer shock absorber 3, rod 5 enters the cavity of cylindrical body 4 and compresses the elastomer mass, generating high internal pressure, and energy is absorbed as a result of this process. In the course of impact (dynamic) compression of the shock absorber, energy absorption happens owing to the elastomer mass flowing (throttling) through calibrated slot 9 between the lateral face of piston 6 and the interior face of cylindrical body 4.

3.6.4 Buffers

Buffers are intended for absorption of longitudinal forces; however, depending on the rollingstock type and the coupler type applied, they perform a number of additional functions.

Buffers of freight cars with screw couplers are intended for compressive longitudinal forces absorption and energy dissipation; that is, they perform the same function as draft gears of cars with automatic couplers.

The typical buffer (Figure 3.66) consists of plate 1 and case 2, inside of which the absorbing device is located. Depending on the absorbing device type, buffers can be spring, with helical springs, spring with ring springs, hydraulic, hydraulic gas and elastomeric.
According to EN 15551 [18], depending on the type A, B, C or L (long strobes), the dynamic energy absorption capacity of buffers is 30 to 72 kJ per each buffer. The full travel of buffers for freight cars is 105 mm, and, for buffers for passenger cars, it is 110 mm; for type L, it is 150 mm.

Wagons for hazardous materials are equipped with crash buffers. In the case of large impact forces (more than 1.5 MN), crash buffers are plastically deformed and do not return to their initial state.
The Innova System and Technologies buffer is shown in Figure 3.67a. Deliberate weakening of the case is made in the form of slots. When affected by large forces, the case gets plastically deformed, as shown in Figure 3.67b, to absorb the impact energy. The energy-absorption capacity of such buffers is 250 to 400 kJ.

Passenger car buffers with automatic non-rigid coupling are intended for thrusting and clearance adjustment in automatic couplers of passenger cars. Their energy-absorption capacity is not large, because the main energy is taken up by the draft gear. Buffers protrude 65 ± 10 mm over the engagement plane of automatic couplers (Figure 3.68). After automatic couplers have been coupled, the tight condition of the automatic coupler and the clearance adjustment result. The tight condition of the automatic coupler reduces the level of longitudinal acceleration when that train movement modes are changed, which is important for maintaining comfort of passengers. Apart from that, the coupler contour wear of automatic couplers reduces, and the danger of uncoupling for automatic couplers decreases [50].

In some designs of transitive platforms (see later), structures of inter-car gangways rest on buffers.

Passenger car buffers with automatic rigid coupling are intended for absorption of longitudinal forces, because there is no necessity of clearance adjustment in automatic couplers. Their design is essentially similar to designs of buffers in freight cars; the differences are a bigger travel (110 mm) and greater energy-absorption capacity. The example of a buffer with a polymer hydraulic absorber is shown in Figure 3.69.

### 3.6.5 INTER-CAR GANGWAYS

Inter-car gangways are intended for passing from one car to the next one, and they often consist of an enclosure, a sliding frame and a connecting bridge [52].

Inter-car gangways can be with the closed contour, which provides the complete enclosure of the passage, or with an unclosed contour closing only the sides and the top of the passage. Inter-car gangways without the sliding frame, but with rubber tubes, are applied for passenger coaches of 1520 mm track gauge (see Figure 3.70), together with the SA-3 automatic coupler. They consist of three rubber tubes 1 that are fixed onto the car body. In their bottom part, between the pair of
buffers 3, buffer coupling is located, whereon connecting bridge 2 rests, closing the automatic coupler and providing safe passage for passengers. Relative displacements of cars are compensated for by deformation of the rubber tubes.

Inter-car hermetic gangways (Figure 3.71) are used with rigid-type couplers. Relative displacements of the coaches are compensated for by corrugated bellows.

In some high-speed trains, shock absorbers are installed between cars, and they damp reciprocal oscillation of cars. This allows the reducing of input energy from the rail track. The inter-car gangway with inter-car damper of the Talgo train is shown in Figure 3.72.

**FIGURE 3.68** Buffers of 1520 mm gauge passenger coaches: (a) with two consecutive springs, (b) with one spring and (c) with spindle type damper.
FIGURE 3.69 ETH hydraulic combination shock-absorber from Eisenbahntechnik Halberstadt, Germany. (1) Plate; (2) case; (3) hydraulic-polymer elastic elements; (4) foundation; and (5) mounting.

FIGURE 3.70 Inter-car gangway without the sliding frame with rubber tubes. (1) rubber tubes; (2) connecting bridge; and (3) buffers.

FIGURE 3.71 Inter-car hermetic gangways. (1) Frame and (2) corrugated bellows.
3.7 PRINCIPLES FOR THE DESIGN OF SUSPENSIONS

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

- There is sufficient reserve of critical speed with respect to design speed
- Ride quality, track forces and safety factors satisfy the standards on a straight track and in curves for all range of operational speeds
- 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 later [53]. To make sure that the parameters are optimised, further refinement is usually done using computer simulation.

3.7.1 SUSPENSION CHARACTERISTICS IN VERTICAL DIRECTION

Suspension should control and damp the motion of both the sprung and unsprung masses in the vehicle to obtain the best possible ride qualities whilst strictly fulfilling the safety requirements and satisfying specific service limitations such as ensuring that the vehicle remains within the clearance diagram.

Bogie elastic elements have various constructions, which, for example, can be cylindrical, rubber, leaf or pneumatic springs. From the dynamic behaviour from the vehicle point of view, the specific construction of the elastic element is not important, but the force characteristic that it provides is significant, that is, the dependence of the vertical load on the element $P$ from its static deflection $f$: $P = P(f)$. 

![FIGURE 3.72 End wall of Talgo Pendular passenger coach. (1) One-axle bogie frame; (2) wheel; (3) axlebox and gauge changing mechanism; (4) bottom inter-car hydraulic damper; (5) top inter-car hydraulic damper; (6) pneumatic suspension bellow; (7) support column; (8) inter-car gangway; (9) bridge and door; (10) coupler element.](image-url)
The static deflection of a suspension with linear characteristics (constant stiffness) is determined by the formula:

\[ f_{st} = \frac{P_{st}}{c}, \]  

(3.6)

where \( P_{st} \) is the static load on the suspension, and \( c \) is the stiffness of the suspension.

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

\[ \omega^2 = \frac{c}{M} = \frac{g}{f_{st}}, \]  

(3.7)

where \( M \) is the sprung mass of the vehicle, and \( g \) is the gravity acceleration.

Research has shown that decreasing the suspension vertical stiffness (increasing the static deflection) is favourable for the dynamic performance and force influence of the rail vehicle on the railway track if other conditions do not change.

In general, a low suspension stiffness gives lower acceleration, but practical considerations dictate that there must be a relatively small height difference between tare and laden conditions. To overcome such a limitation, new designs of coupling devices and brake systems are necessary [54]. In addition, the human perception of vibration over a range of frequencies must be considered. For passenger vehicles, the car body bounce frequency is generally in the range of 0.9 to 1.2 Hz, whilst this frequency for freight wagons can rise to 2.5 Hz in laden and up to 4 Hz in the tare conditions.

In order to realise an increased suspension deflection, modern suspensions use non-linear springs to provide optimal stiffness in the vicinity of the static deflection corresponding to the required load.

In suspension elements with variable stiffness (Figure 3.73), 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:

\[ c_{eq} = \frac{dP}{df} \bigg|_{P=P_{st}} ; \quad f_{eq} = \frac{P_{st}}{c_{eq}}, \]  

(3.8)

In a suspension with a stiff force characteristic, as shown by curve b in Figure 3.73, \( f_{eq} < f_{1st} \) and, for a soft characteristic, as shown by curve b, \( f_{2eq} > f_{2st} \), where \( f_{1st,2st} \) is the total static deflection.

In general, for the suspension with variable stiffness (bilinear characteristic), 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 (see trace c in Figure 3.73):

In this case:

\[ c_{eq} = \begin{cases} c_1, & f \leq \Delta \\ c_2, & f > \Delta \end{cases} ; \quad f_{eq} = \begin{cases} \frac{P_{st}}{c_1}, & P_{st} \leq c_1 \Delta \\ \frac{P_{st}}{c_2}, & P_{st} > c_1 \Delta \end{cases}, \]  

(3.9)

where \( \Delta \) is the deflection corresponding to the breakpoint of the characteristic.

In freight wagon bogies, a non-linear vertical characteristic is usually provided by combinations of cylindrical springs having different free heights. In such bogies, the characteristic can be multi-linear, containing up to four linear pieces.
In theoretical calculations of the suspension force characteristic, there are sharp transition points from one piece of multi-linear characteristic into the other; these are much smoother in the real world, where the springs have free height production tolerances. Therefore, the transition zones in simulation are often smoothed with fillets as well.

To provide structural strength of the elastic elements and no impacts in the suspension, especially with coil springs, they should provide no solid state during the oscillations of a fully laden car body while moving in a train. To consider this situation, the deflection reserve coefficient $K_{res}$ is introduced for suspensions in general:

$$K_{res} \geq 1 + k_d,$$

with a limit value that should be bigger than $(1 + k_d)$, where $k_d$ is the maximum possible ratio of dynamic vertical force in car body oscillations to the static vertical force in the suspension [55].

For suspensions in general, the deflection reserve coefficient is the minimum value among all elastic elements. For each of the coil springs in the suspension, the deflection reserve coefficient can be calculated using the formula:

$$K_{res} = 1 + \frac{h_f - h_s - (f_{st} - e)}{f_{eq}},$$

where $h_f$ and $h_s$ are the spring free height and solid height, respectively; $e$ is the height difference between the tallest spring and the chosen spring; $f_{st}$ is the full static deflection of the spring and $f_{eq}$ is the equivalent deflection of the suspension.

To provide the safe operation of coupling devices in the train, the height difference between the longitudinal coupler (or buffer) axes of two neighbouring cars should not exceed the prescribed value. 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 (e.g., wheel profile wear or wear of centre bowls and side bearings).

**FIGURE 3.73** Non-linear elastic force characteristics of a suspension: stiff (curve a), soft (curve b) and bilinear (trace c). (From Iwnicki, S. (Ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)
In service, the car body roll must also be limited to prevent the risk of overturning on highly canted curves and to ensure that the vehicle remains within the required clearance diagram. Once the maximum allowable roll angle for the vehicle body and the maximum lateral force (centrifugal, wind and lateral components of the interaction force between the vehicles in curves) have 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.

### 3.7.2 In-Plane 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 non-linear, and the relation 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 guiding 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 requirement 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 the ride quality in an analytical or graphical form. The simplified approach described in [32] is useful as a starting point.

The natural vibration modes shown in Figures 3.74 and 3.75 can be obtained from the linear equations of motion for a two-axle bogie [46].

Analysis of the modes shows:

- For in-phase yaw, as shown in Figure 3.74a, there is a relative lateral displacement between the centres of wheelsets $O_1$ and $O_2$ and the bogie centre.
- Similar lateral displacements appear for the anti-phase mode, shown in Figure 3.75b.
- Relative rotation between wheelset centres $O_1$ and $O_2$ occurs only for anti-phase yaw of wheelsets (Figure 3.75a).

![FIGURE 3.74](https://example.com/fig3_74.png)  
Wheelset modes for a two-axle bogie: (a) in-phase yaw and (b) in-phase lateral displacement.  
Thus, two generalised parameters can be introduced for the bogie:

- A stiffness corresponding to relative lateral displacement between the centres of wheelsets, referred to as the shear stiffness ($K_s$)
- 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 3.76, represented 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 3.76) allows the in-plane bogie stiffnesses to be chosen, without considering its specific design.

Solution of the stability problem [36] shows that the critical speed of a conventional railway vehicle is a function of its shear and bending stiffnesses, as shown in Figure 3.77. 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 stiffnesses, as shown in Figure 3.78.

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.

### 3.7.3 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 necessity of a separate damper.

The selection of the optimum damping levels is a more complicated problem than the choice of suspension stiffness, although damping is less dependent on existing operational limitations. 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:
where \([B]\) and \([M]\) are the damping and inertia matrices of the vehicle multi-body model, respectively; \([v_i]\) is the column-vector of the \(i\)-th eigenmode and \(\omega_i\) is the natural frequency of the \(i\)-th eigenmode.

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

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.

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:

\[
\Delta A'' \geq \Delta A'.
\] (3.13)

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

\[
F \geq \frac{\pi q}{4} c_{eq},
\] (3.14)
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 when using a relative friction coefficient that equals the ratio of friction force to the static vertical load:

$$\varphi = \frac{F}{P_{st}} \geq \frac{\pi q}{4f_{eq}}, \quad (3.15)$$

where $f_{eq} = \frac{P_{st}}{c_{eq}}$.

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 a nonlinear suspension characteristic, the vertical load. For freight wagons, the recommended optimum relative friction coefficient lies within the range of 0.2 to 0.4 for the empty condition and 0.07 to 0.13 for the fully laden condition [36].

### 3.7.4 Car Body to Bogie Connections

Car body to bogie connections have the following functions:

- 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

These functions depend on the type of the rollingstock: 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 car body to bogie connections may not be necessary.

Designs generally aim to make the bogie to car body connections as simple as possible by the use of a small number of elements and reduction of the number of elements with surface friction. The conventional design uses the centre bowl to centre plate connection (flat or spherical) in the middle, with side bearings (elastic constant contact or rigid with a clearance) by the sides. From the railway vehicle dynamics point of view, it is described by the following parameters:

- Vertical and longitudinal stiffness of the elastic element of the side bearing (zero for rigid side bearings with a clearance)
- Static deflection of the constant contact side bearing under the car body (usually expressed in percentage of car body weight carried by the side bearings)
- Friction coefficient on the side bearing surface
- Friction yaw torque and roll recovery in the centre bowl
- Maximum possible deflection (or clearance for rigid) of the side bearing limited by the bump stop

The weight per one constant contact side bearing is determined by the formula:

$$P_{st} = c_{spr} \cdot f_{st}, \quad (3.16)$$
where $c_{sp}$ is the vertical stiffness of the elastic element, and $f_{st}$ is the static deflection of the side bearing.

To provide uniform distribution of car body weight to the bogie, the sum of vertical forces per side bearing is usually limited to not more than 85% of empty car body weight:

$$\eta = \frac{4P_s}{M_bg} \times 100\%,$$

where $M_b$ is the minimum car body mass.

The constant contact side bearings experience not only the static weight of the car body but also the dynamic roll oscillations, providing the recovery torque and thus improving the ride performance [39]. The dependence of mode damping on the side bearing vertical stiffness is presented in Figure 3.79. It appears that the damping is highly dependent on the longitudinal mass moment of inertia of the car body (23,000 kg·m² for the gondola and 131,000 kg·m² for the hopper). With the increase of the vertical stiffness of the constant contact side bearing, the damping of roll oscillations increases. In addition, there is an increase in overturning safety in curves but a decrease in derailment safety on straight track (Figure 3.80). The rational values of the side bearing vertical stiffness that provide the necessary compromise correspond to damping coefficients in the range of 0.25 to 0.3.

**FIGURE 3.79** Dependence of car body roll damping coefficient on side bearing vertical stiffness for gondola car and hopper car.

**FIGURE 3.80** Impact of side bearing vertical stiffness on the probability that the Y/Q coefficient is greater than the limit value for: (a) the empty wagon and (b) the fully laden wagon.
To keep down the car body roll angle and provide the structural strength of the side bearing elastic elements, their vertical deflection is usually limited with a bump stop or locking between the cap and the cage. The maximum vertical force on the side bearing appears when the laden wagon passes through sharp curves, with large rail superelevation at maximum possible speed.

The longitudinal stiffness and friction force in the side bearing (that can be viewed in series with elastic element shear deflection limited by the bump stop) influence the critical speed and wheel-rail lateral forces in curving. Increase in the shear longitudinal stiffness of the side bearing (see Figure 3.81) leads to the increase of the critical speed with a larger rate of increase for stiffness below 3.5 MN/m and with a much smaller rate for stiffness above this. Analysis of curving using the nonlinear models showed that, within the possible range of yaw friction torque and the range of side bearing longitudinal stiffness, providing necessary critical speed level, no variation of curving qualities was obtained. However, standards tend to limit the total yaw torque provided by constant contact side bearings down to 12 to 15 MN·m/rad.

FIGURE 3.81 Dependence of the critical speed on longitudinal stiffness and viscous damping being modelled in series in the side bearing.

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