Chapter 11
Mechanical and Hydraulic Gears

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11.1 Definition of Rack-and-Pinion Gears

The family of steering equipment covers hydrostatic steering systems, recirculating ball-with-nut gears, manual rack-and-pinion steering and rack power steering. The final steering system mentioned has a rack-and-pinion gear which transforms the rotation of the steering wheel into the translation of the rack and again into the steering movement of the tyres.

Table 11.1 shows a classification of steering system designs. The classifying features of the various models chosen are active principle, drive, primary gear, force introduction and integration. The naming of the steering and the corresponding abbreviation complete the presentation.

In a manual rack-and-pinion steering the parameters are the wheel angle and the wheel torque, which is initiated at the steering wheel and transformed by the dovetailing components, pinion and rack, into rack force and rack shift. Beyond these mechanical variables, no further energy is supplied to the manual rack-and-pinion steering to move the rack.

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Table 11.1 Classification of steering designs

	Manual steering	Power-assisted steering								
Active principle	Mechanically	Hydraulically	Electric-hydraulically	Electromechanically						
Drive	Manual	Engine	E motor	E motor						
Primary gear		Belt drive	Direct drive	Worm gear			Bevel gear	Belt drive	Direct drive
Power introduction		Hydraulic pump	Hydraulic pump	Steering column	Single pinion	Double pinion	Recirculating Ball and Nut Steering Gear		
Integration	Mech. Rack	Mechanical gear hydr. assist	Mechanical gear hydr. assist	Mechanical gear	Steering gear with integrated electrical drive				
Nomenclature	Manual steering	Hydraulic assist power steering	Electro hydraulic assist steering	Column EPS	Pinion EPS	Double pinion EPS	Axis offset EPS	Axis parallel EPS	Rackcentred EPS
Abbr.	MSG	HPS	EHPS	C-EPS	P-EPS	DP-EPS	RC-EPS	B-EPS	CR-EPS

The rack power steering works essentially by the same principle, but an 'assistant' supports the driver's steering activities. This assistant can be either hydraulic, electric-hydraulic or electromechanical. In an electromechanical rack-and-pinion steering, for example, the assistant moving the rack is an electric motor which allows contributing a suitable energy supply by means of an additional power-assist gearbox. A hydraulic assistant is a pump, powered by the internal combustion engine, that supplies the steering system. In contrast to both, the steering system of the electric-hydraulic version is decoupled from the internal combustion engine, and the hydraulic pump is driven by an electric motor. The structures these power steerings have in common, is that the existing wheel torque is exploited to control the assistant. The base for receiving the wheel torque is usually a torsion bar which allows activating the steering valve of hydraulic and electric-hydraulic steering, or a suitable twisting angle in the torque sensor of the electromechanical steering. The manual steering has no torsion bar.

This chapter will first discuss the mechanical rack-and-pinion gears, either driven by hand or by the above-mentioned power-assist gearbox, separated from the mechanical rack-and-pinion gear.

11.2 Applicability/Pros and Cons

Manual rack-and-pinion steering and rack power steering are almost exclusively used in vehicles with independent suspensions at the front axle. These are passenger cars of almost any vehicle class, as well as light commercial vehicles.

A direct comparison of the recirculation ball with nut gears that had once been widespread in these vehicle classes, too, shows the pros (+) and cons (−) in Table 11.2. Rack-and-pinion steering has replaced the recirculating ball gear in

Table 11.2 Comparison between rack-and-pinion steerings and recirculating ball gears

Assessment criterion	Rack-and-pinion steering	Recirculating ball gear
Freedom of action	+	–
Efficiency	+	–
Bumpiness	–	+
Steering elasticity	+	–
Stroke limitation	+	–
Tie rods—radial forces	–	+
Usability with rigid axles	–	+
Drop of the full ratio over the wheel angle	–	+
Complex structure	+	–
Space required	+	–
Production costs	+	–
Weight including steering linkages	+	–

almost any vehicle with independent suspensions of the front axle, because of the lower steering elasticity, less need for space, less weight of the full steering system and lower production costs. The cons of the rack-and-pinion steering—less damping of externally excited power pulses (bumpiness), the curve of the steering ratio and the lateral forces from the tie rods—are compensated by constructive measures.

11.3 Kinematic Differentiating Features of Gears

The first basic criterion for the kinematic distinction of the gears is the on-board position of the wheel, distinguishing left-hand or right-hand driving. To supply all the markets, both versions are developed and offered for most vehicles in Europe. In the easiest case, if the structure of the vehicle permits it (package), the gear of either version is laid out symmetrically to the longitudinal axis.

In addition, there are special versions of sports or racing cars that feature a central steering wheel. They will not be discussed here any further.

11.3.1 Position of the Gear in Relation to the Front Axle

The on-board positioning of a gear in relation to the front axle provides four basic options, depending on whether the rack axis of a gear (gear axis) and the steering triangle of steering arm and tie rod are in front of or behind the front axle (see Figs. 4.19, 4.22, 4.23 and 4.24).

Most cars with front-wheel drive and laterally arranged engine/gearbox unit feature both gear and triangle behind the front axle. This arrangement enables easy integration of the steering column into the vehicle package. On the other hand, an unfavourably high thermal load will often accompany this arrangement, because the exhaust system is close to the gear.

Standard-driven vehicles preferably have both gear and steering triangle in front of the front axle. This avoids conflicts in the package between the longitudinal drive unit and the lateral gear, and the steering column can be easily guided past the engine.

Gears and steering triangles are rarely arranged at different positions (e.g., gear behind the front axle and steering triangle in front of the front axle; see Chap. 4, segment 4).

The position of the steering triangle gives ultimately the direction of the rack's motion upon steering, therefore it is essential for the dovetailing design, defining the way the pinion dovetailing is inclined.

11.3.2 On-board Gear Interfaces

The majority of vehicles has a steering mounted on a separate subframe that is in turn mounted to the body. Since the wheel suspension at the chassis is at least partially joined to this subframe, there is a stiff connection between steering and front tyres. This creates a basis for precise steering, even under the influence of high lateral forces in the wheels. Furthermore, the subframe is acoustically decoupled from the car, so that less chassis noise is transmitted by the gear into the body.

In addition, there are vehicles whose steering is separated from the front axle and directly joined to the body. This mounting is often given by the package, for example, in cars with front-wheel drive and longitudinal engine in front of the front axle (example: any Audi A6 older than 2011). Sometimes the steering is directly mounted at the engine bulkhead, i.e. the separation between engine and passenger compartment, this is a very cost-efficient solution. If any acoustic setbacks of this arrangement cannot be accepted, they are compensate by additional measures, such as damped tie rods and elastic mounting of the gear at the engine bulkhead.

The mounting points can be either rigid or flexible, i.e. by means of silent blocks or rubber mountings. A rigid arrangement provides highest stiffness of the steering system, but it has setbacks in acoustics and bumpiness. Rubber mountings offer the chance to tune the response and damping characteristics of the steering and to achieve the best compromise for the vehicle in question.

A rubber (metal) mounting consists in principle of a jack, usually made of steel, that is vulcanised into a rubber body. It acts like a progressive spring and provides the kind of damping that is native to rubber. Lateral and axial stiffness, as are necessary for the respective application, and the desired damping parameters are identified by means of vehicle simulations and road tests. Figure 11.1 shows typical one-piece and two-piece versions of rubber mountings.

The number of fastening points can vary from at least two, used to precisely adjust the on-board gear, up to four or more, which enables precise, controlled elastokinematics, together with the properly tuned rubber mountings.

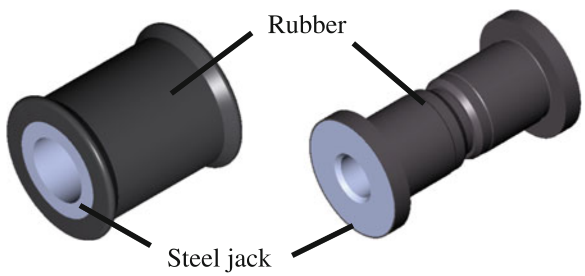


Fig. 11.1 Typical rubber mountings as one-piece and two-piece edition

11.3.3 Adjustment of the Gear Case

The co-ordinates of the main points of a rack-and-pinion gear—more precisely: the points of the axial joints at the tie rods, the lower universal joint of the steering column and the junction point of the rack and pinion axis—are given by the package of the full vehicle, as shown in Fig. 11.2. Therefore the pinion case can be turned outward or inward from the car's central plane. An inward orientation of the pinion case, vertical to the rack axis, would be unusual, though. It is chiefly found in racing vehicles.

11.3.4 Tie Rod Interfaces

Depending on how the two tie rods are connected with the rack, there is a distinction between a rack-and-pinion steering with central tap and others with end tap at one or both sides. A rack-and-pinion steering with end tap at both sides has the inner joints of the tie rods (axial joints) screwed directly onto both ends of the rack. This configuration is the current state-of-the-art, Fig. 11.3.

It merges high stiffness with low weight, because the rack axis is passing through the points of both axial joints. Torsion torques in the rack develop only from lateral forces from the tie rods which are a function of the tie rod angle (see also Fig. 11.2). The rack of a mechanical gear for standard strokes can be short, so

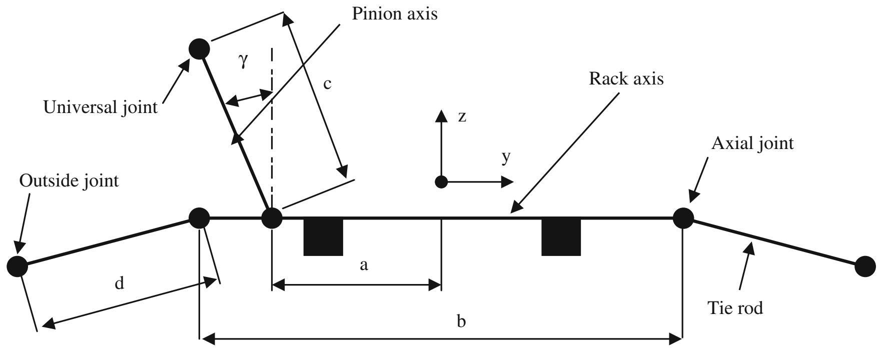


Fig. 11.2 Typical outlines of a rack-and-pinion gear

Fig. 11.3 Mechanical gear with end tap on both sides (Renault Scenic 3)

usually there is enough space for it in the package. Rather long tie rods can be used which display low torsion angles even at full deflection and rebound of the front axle, so that low lateral forces are applied to the rack. A rack-and-pinion steering with central tap has only one mounting point for both tie rods, and it is in the centre of the car, if the steering is set on straight driving. This steering model is rarely used in new applications, on account of the weight and the required space (see also Sect. 11.9.1). Short gears with end taps on one side are not used any more, either. They had been very compact, so that they could be mounted on the driver's side. By analogy with a gear with central tap, the tie rods were mounted at one end of the rack, while the other side remained free. This construction was suitable for purely manual steering with correspondingly low tie rod peak forces.

11.3.5 Kind of Gear Ratio

The stroke of the rack during one turn of the pinion is called the gear ratio of a rack-and-pinion gear. This means that for a higher gear ratio (more stroke per turn of the pinion), the whole steering ratio on-board will drop (wheel-angle-to-steer-angle ratio). A high gear ratio means that there is less effort for a wheel angle, for example on curvy roads or when cornering in the city. On the other hand, a high gear ratio complicates the exact adjusting of very small steer-angles, for example, when slight corrections of the straight course should be carried out at high speed. Moreover, a high gear ratio produces high steering forces in a manual steering and high power-assist torques in electromechanical steering systems.

The majority of gears produced has a steady gear ratio. If it changes as a function of the wheel angle or the rack stroke, this is called a variable gear ratio. This variability can be either constant, rising or falling with increasing wheel angle on different sections. This suggests a solution of the above-mentioned problems. The gear ratio curves of manual and mechanical rack-and-pinion gears are dropping with rising steer-angles, to counteract the kinematic rise of the wheel torques. There are also M-shaped (rising–falling) versions which merge this quality with a low gear ratio during straight driving. Typical steering ratios of different versions are compared in Fig. 11.20. For relevant differences, resulting gear ratio curves, layout criteria and ways to produce a variable gear, see Sect. 11.5.

11.4 Design and Main Components of a Mechanical Rack-and-Pinion Gear

Some essential components of a mechanical rack-and-pinion gear that are important for the steering are shown in Figs. 11.4 and 11.5:

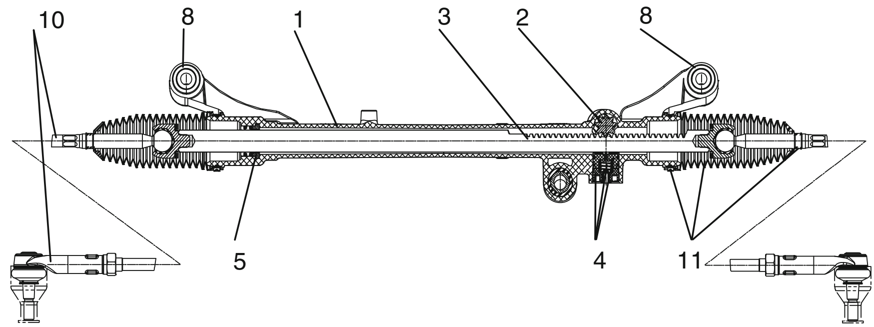


Fig. 11.4 Cut of a mechanical rack-and-pinion gear

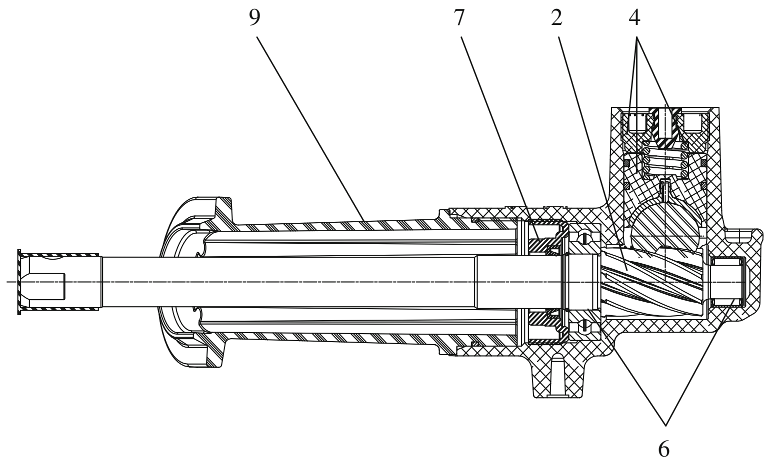


Fig. 11.5 Cut of the pinion case of a mechanical rack-and-pinion gear

1. gear case
2. pinion
3. rack
4. rack yoke elements, including spring and adjuster bolt of the free travel
5. rack bushing
6. upper and lower roller bearings of the pinion
7. upper pinion screw to secure the axial mounting of the pinion
8. joining elements to receive and mount the gear case on-board
9. gear case gasket towards vehicle interior, near the junction of the pinion to the column (use depends on the on-board implementation of the gear)
10. tie rods (inside and outside joint)
11. gaiters including coupling clamps.

11.4.1 Gear Case

The gear case receives and joins all the components of the gear. It is either rigidly or elastically mounted at the vehicle by rubber mountings (see Fig. 11.1). The case has the additional task to accept any steering forces present, transmitted as wheel torques, and any forces acting from the tyres on the rack via the tie rods. Other functions are the limitation of the stroke and the reception of the end stop forces.

One-piece and two-piece gear cases are the most common designs. They are described in more detail in the following.

11.4.1.1 One-Piece Gear Case

The one-piece type is the most common one. For an example, see Fig. 11.6. This version of a gear case consists of die-cast aluminium and offers the following benefits: The flanges to mount the gear case in the car are made right in the casting process. This technique also allows adding holders and mounts. The die-cast aluminium case is light and requires no costly welding or additional surface cover, unlike steel parts.

A few typical die-cast aluminium materials for gear cases are GD AlSi9Cu3 and GD AlSi12Cu. A listing of proper characteristics is shown in Table 11.3. The sand-cast G-AlSi7Mg0.3 and its mechanical qualities are also given in the table, as it is often used in prototype building and for small numbers of pieces.

11.4.1.2 Two-Piece Gear Case in Composite Design

The two-piece version has a pinion case of aluminium and a rack case made of a steel pipe, see Fig. 11.7. The parts of the case are connected by grouting both components or by casting around the steel pipe during the die-casting process. The flange at the pinion case that mounts the gear case to the vehicle is shaped during the casting process. At the steel pipe, the case is joined either with a holder welded to the pipe or with an aluminium flange that is cast around the pipe. A very cheap way to mount a two-piece case to the vehicle can be realised with a separate securing clip and a rubber insert at the steel pipe.

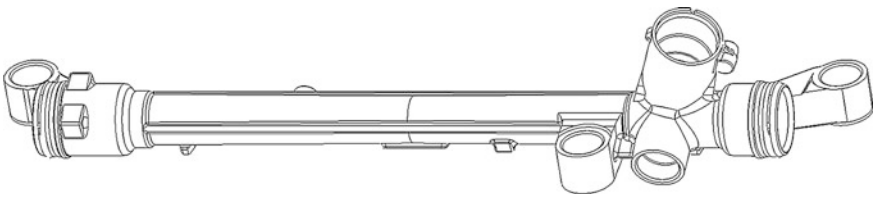


Fig. 11.6 One-piece gear case from aluminium die-casting

Table 11.3 Typical aluminium die-cast and sand-cast materials and their mechanical parameters

	GD Al Si 9 Cu 3 Die casting	GD Al Si 12 Cu Die casting	G Al Si 7 mg 0.3 Sand casting (T6)
Tensile strength (N/mm^2)	240–310	220–300	230–310
Yield strength (0.2 %) (N/mm^2)	140–240	140–200	190–240
Stretch (%)	0.5–3	1–3	2–5
Brinell hardness (250 kg, 5 mm)	80–120 HB	60–100 HB	75–110 HB

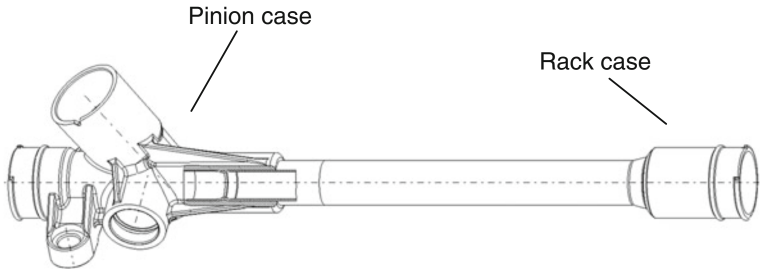


Fig. 11.7 Two-piece gear case in composite structure

One advantage of this two-piece implementation, compared with the one-piece aluminium case, is less space needed for the steel pipe, on account of the lower thickness of the wall. Nevertheless, the steel pipe needs a separate galvanic or varnished surface protection.

11.4.2 Steering Pinion

The pinion is connected to column and wheel by the intermediate steering shaft, and it is dovetailing with the rack. Therefore the main task of the steering pinion is to transform the rotation of the steering wheel into a translation of the rack.

11.4.2.1 Pinion Bearing

One common arrangement of the pinion bearing of mechanical rack-and-pinion gears is a lower needle bearing, which is floating, combined with an upper deep groove ball bearing, which is fixed, that is axially mounted with the pinion, Fig. 11.8. The upper ball bearing and the pinion are again mounted in the gear case by the upper pinion screw, without axial free travel. The bearing therefore supports the axial dovetailing forces which develop from the usual diagonal dovetailing of

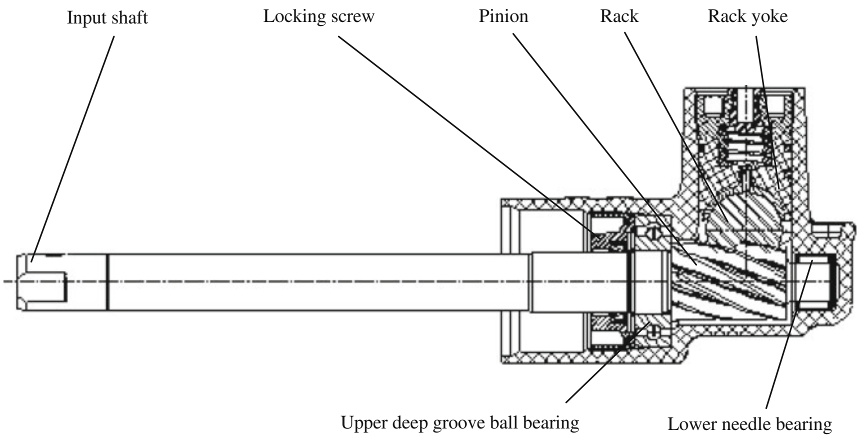


Fig. 11.8 Pinion bearing arrangement with deep groove ball bearing (*top*) and needle bearing (*bottom*)

pinion and rack. It also bears the radial dovetailing forces and those forces which act from the tie rods about the rack on the pinion, together with the lower needle bearing.

This typical pinion support by means of deep groove ball and needle bearing is complemented by the less common variety with two angular contact ball bearings, see Fig. 11.9. In this case, an angular contact ball bearing is arranged both above and below the pinion dovetailing. The two bearings receive axial and radial forces synchronously. This version of a pinion bearing has not prevailed, because the costs of the parts are higher.

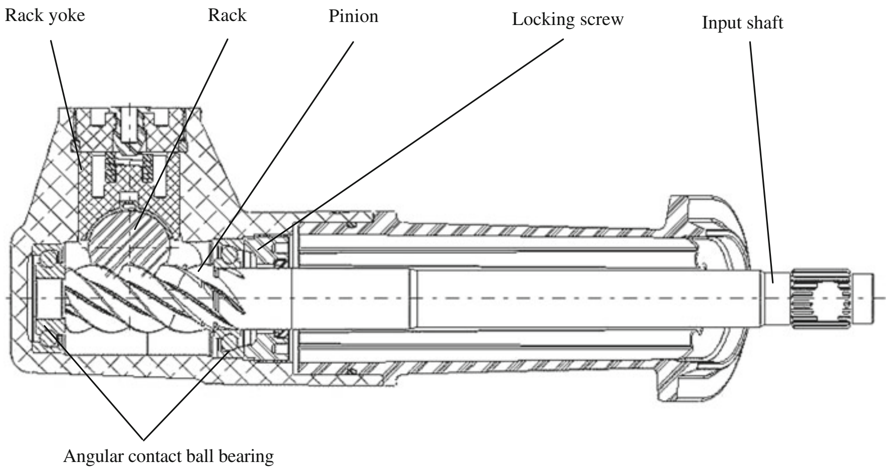


Fig. 11.9 Pinion bearing arrangement with two angular contact ball bearings

11.4.2.2 Implementation of the Pinion Dovetailing

The steering pinion is usually toothed askew to achieve more dovetailing coverage when interacting with the rack dovetailing. This ensures that more than just one cog is dovetailing and that even dovetailing and uniform torque transmission between pinion and rack is maintained. Mechanical gears are basically designed with big dovetailing modules, so that the cogs are firm enough, on account of the high steering torques to be transferred.

Common manufacturing methods for the pinion dovetailing are the most often applied hobbing and the less common gear-tooth forming. There are high requirements for the run-out deviation of the pinion dovetailing (up to 40 μm) to achieve a steady frictional curve over the rack stroke with low free travel of the rack yoke.

11.4.3 Rack and Rack Guide

The rack has the task to convert a rotation, given by the steering wheel on the pinion, into a translation of the rack and the attached tie rods about the dovetailing between pinion and rack. The rack has to transfer the highest applying steering forces and tie rod forces in axial and radial direction. Diameter and material of the rack belong therefore to the main layout criteria of a rack-and-pinion gear. Primarily, the rack should be sufficiently resilient against bending to resist the high lateral forces that the tie rods may issue. Typical materials of racks and their mechanical qualities are shown in Table 11.4.

The rack is generally made of a round-bar steel. The dovetailing is made either by forming or by machining. To optimise the weight, it is partially or entirely hollowed out. There are also versions of racks for which a pipe is transformed according to Sect. 5.5. The making of racks is described there in more detail.

11.4.3.1 Rack Guide

The translation of the rack is enabled by two bearings. Opposite the steering pinion is the rack passing into a plain bearing bush (see Fig. 11.4) that receives the lateral forces of the rack in radial direction.

Table 11.4 Typical rack materials and their mechanical qualities

	SAE 1040 C40-C43	EN8C	37CrS4 41CrS4
Tensile strength (N/mm^2)	640	695	775
Yield strength (0.2 %) (N/mm^2)	440	495	620
Stretch (%)	19	17	13

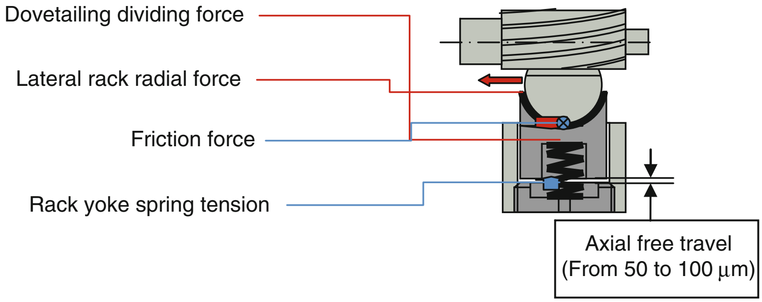


Fig. 11.10 Forces acting on the rack yoke and the axial free travel

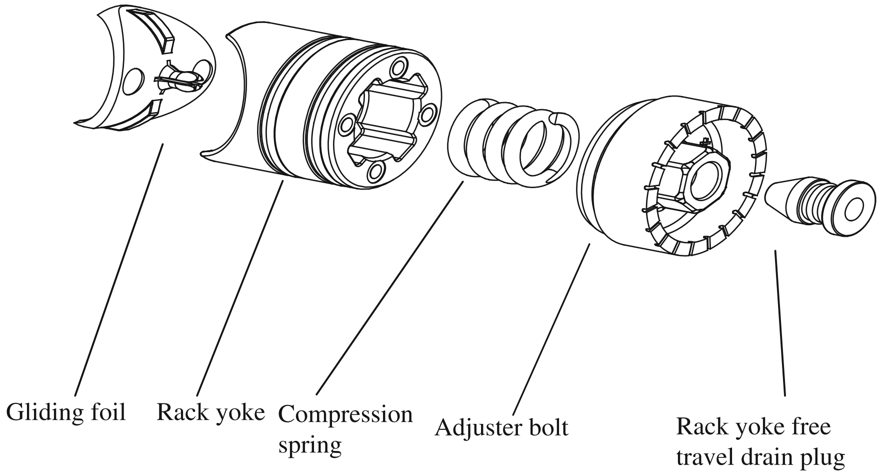


Fig. 11.11 Construction of the rack yoke assembly in the gear case

The rack is guided on the side of the dovetails between the pinion and a spring-loaded rack yoke, see Figs. 11.4, 11.8 and 11.9. The rack yoke almost completely surrounds the diameter of the rack on the side that is turned away from the pinion (looking at the cross section) and presses the rack dovetailing against the pinion dovetailing by means of a compression spring, without free travel, if possible. The rack yoke is guided in a case drilling without free travel. Thus the bearing also enables the reception of the lateral rack forces in radial direction. Figure 11.10 shows the forces acting on the rack yoke, combined of the dovetailing dividing force, the lateral rack force and the rack yoke spring tension. Reliable functioning of the rack support is commonly ascertained by an axial free travel of the rack yoke between 50 and 100 μm. The structure of the rack yoke and its components is shown in Fig. 11.11.

This bearing constellation reduces the movement of the rack to two degrees of freedom: the desired translation and a possible undesirable roll of the rack around its longitudinal axle. The contact area between the dovetailing of pinion and rack along the pinion's axis is rather long, so that a rolling movement will push the rack away from the pinion. First, the spring pretension has to be overcome, then, the rack yoke spring has to be compressed, until the rack yoke touches the adjuster bolt. Consequently, the possible rolling movements are limited to a range which does not affect the function of the gear.

11.4.3.2 Rack Yoke

The rack yoke has to perform the following functions during gear operation:

- Establish a plain bearing of axial rack strokes without free travel
- Establish a dovetailing between pinion and rack without free travel or noise
- Support radial and axial dovetailing dividing forces between pinion and rack
- Support tie rod forces at the pinion
- Compensate dovetailing tolerances between pinion and rack from manufacturing
- Tune steering qualities by varying the frictional mating between rack and rack yoke
- Suppress mechanical noise.

Construction of the rack yoke

To meet the requirements, the following construction of the rack yoke assembly is prevailing in reality. The rack yoke is a rounded plain bearing element which is axially moving in an appropriately sized gear case drilling. The depiction of the rack yoke in Fig. 11.11 shows that there is a plain foil at the rack which is fixed at the rack yoke and allows a low-friction movement of the rack. To achieve radial damping of the rack yoke, one or two O-rings can be integrated, they are fitted into corresponding grooves. The compression spring and the adjuster bolt enable a safe support of the rack. The drilled hole, made into the adjuster bolt to directly observe the free travel of the rack yoke, can be sealed with the drain plug after measurement.

Figure 11.12 shows potential noise sources in the system of the rack support by the rack yoke. This constellation can produce rattle under certain loads, like varying tie rod forces (for example, in a journey over uneven cobblestones) or

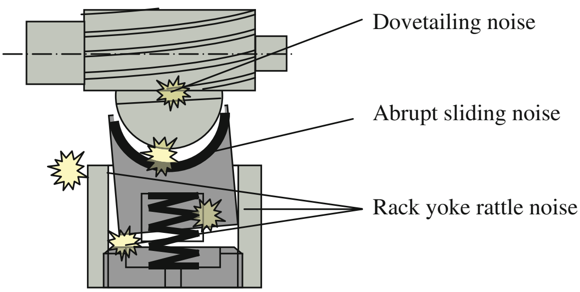


Fig. 11.12 Potential noise sources in the dovetailing and rack yoke area

when movement of the rack reverses (‘reverse knock’). This noise should be fully prevented by suitable measures.

To avoid noises of the rack yoke, one aims at as little radial and axial free travel of the rack yoke in the case drilling as possible.

The radial bearing clearance results mainly from the accessible tolerance margins for the outer diameter of the rack yoke and the inner diameter of its drilling. It depends on the chosen way these diameters are processed and on the mating of materials, because different thermal expansions require measures to prevent the rack yoke from blocking.

The axial bearing clearance is given by the adjuster bolt for the rack yoke clearance that ultimately represents a stop unit for the rack yoke, preventing further movement. Here again, the free travel should be as low as possible. Still, note that an insufficient rack yoke clearance entails a powerful rise of the friction in the rack, which the driver will perceive as very unpleasant and may interpret as a blocked rack. Hence, the rack yoke clearance has to be adjusted in such a way that it can also compensate the permissible deviations of the cog treads of the rack and the pinion, the run-out deviation of the pinion and the permissible static sag of the rack, beside the already mentioned tolerance and stretch differences. The permissible tolerance of the mentioned parts are therefore integrated into the definition of the rack yoke clearance. A clearance of up to 100 μm is generally common. It is measured during the adjustment of the bolt straight through a drilling in its centre. After setting the clearance, the drilling is sealed in the bolt with the drain plug.

Optionally, an integrated O ring can be added to the adjuster bolt in the contact zone to the rack yoke, so that the noise in this area is further damped, if the operation of the steering should cause the rack yoke to strike on the adjuster bolt.

Possible rack yoke materials are aluminium, zinc-aluminium alloys, sintered metal or plastic. The friction contact surface of metal rack yokes to the rack consists of an additional foil of plastic or composite metal with bronze and PTFE support. The different sliding materials, combined with the pretension of the rack yoke spring, enable a tuning of the friction at the desired level. In general, one aims at very low friction to provide a good steering feel to the driver. At the same time, the steering has to damp any disturbances and vibrations acting from the outside; a defined friction is contributing to this. However, all sliding materials have in common that the difference between sliding friction and static friction should be as low as possible, this avoids stick-slip effects and resulting noise or abrupt movements of the rack during small steering corrections.

The friction contact surfaces of the rack yoke are adapted to the rack according to its geometry in the dovetailing area, i.e. semicircular, Y or V (see Sect. 11.5.5). Using Y or V rack yokes prevents the rolling of the rack due to the dovetailing for the most part.

Rack yoke tuning

Altogether, the rack yoke and its parts allow tuning the steering qualities, damping mechanical noise and rack vibrations and optimising the steering response. Structural factors are the level of the rack yoke spring tension, the sliding parameters of the contact surface of rack yoke and rack, or their friction coefficient, the free travel of the rack yoke bearing and the insertion of O-rings in the perimeter. These factors affect response, damping and noise of a mechanical gear and can within limits influence the steering qualities.

Changes in the area of the rack yoke are often used in practice to damp noises and vibrations which are recognised only late in the process of development, without having to essentially modify the gear.

11.4.3.3 Rack Bushing

The rack bushing—the second rack bearing of a mechanical gear, beside the rack yoke with pinion—is a sliding fit which supports the lateral forces from the tie rods in the rack and ensures that the translation of the rack is low-friction and free of noise. In some applications, the rack bushing also assumes the function of the end stop, limiting the rack stroke.

Mechanical rack gears and gears for pinion EPS most often feature plastic sliding fits, versions of sintered metal are less common. By analogy with the contact area of rack and rack yoke, the desired friction can be adjusted by choosing the material mating. To counteract rattling noises, two O rings are sometimes applied at the outer diameter of the rack bushing of plastic sliding fits. This facilitates good noise damping. With the help of these O rings, some further damping of axial rack movements can be achieved for small stroke movements as well. In addition, sudden radial load impulses are damped by this kind of bearing. An example of a plastic rack bushing with two O rings is shown in Fig. 11.13.

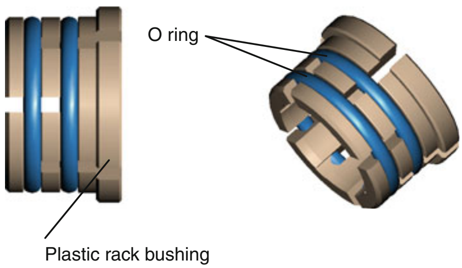


Fig. 11.13 Rack bushing with two O rings

11.5 Dovetailing and Gear Ratio

The purpose of the dovetailing design is to include different limiting criteria, like high load-carrying capacity, high efficiency, low noise and high bending strength of the rack. Some of the individual measures generate opposing effects, demanding a compromise. For any given rack diameter, one tries to achieve the highest possible overlapping of profile and jump. The remaining cross section may not be weakened too much, otherwise the bending strength of the rack will be impaired. Constant and variable gear ratio are distinguished.

11.5.1 Constant Gear Ratio

The use of gears with constant gear ratio (CGR) is widespread in the car industry, where customary steering is concerned. The historical reason is the easier production by broaching. Broaching is acknowledged as an established manufacturing method for 'common' rack dovetailing, and it is most often applied; sanding and milling the dovetailing belong to the less widespread processes.

The full ratio of a consistently translated gear is computed from the centre gear ratio and the steering axis geometry of the vehicle. Consequently, the choice of a suitable gear ratio is a compromise between the preferred ratio in the centre area (or 'straight ahead' position), the desired manoeuvrability of the vehicle and the full number of steering wheel rotations, stop unit to stop unit.

The possibility to individually adapt the steering qualities by a variable gear ratio is a comfortable solution to find the best compromise for layout and adaptation of a steering system.

With regard to rules for the dovetailing design, the CGR is a special case of the variable gear ratio, so that the layout criteria discussed in the following sections also apply to CGR dovetailing.

11.5.2 Variable Gear Ratio

The above-mentioned manufacturing methods are not applicable for variable dovetailing with a flank geometry changing over the stroke. Hence, racks with variable dovetailing in mass production are either hot or cold transformed.

Dovetailing with variable gear ratio (VGR) uses a helical pinion, just like the CGR versions. The flanks of the rack cogs are bent, so that the same angles of the engaging cogs are present at any time of engagement. The left side of Fig. 11.14 shows the front cut through rack and pinion dovetailing. In the distance of the pitch circle diameter $d1$, the cogs of the rack have a frontal division $pt1$ (normal division to the pinion axis) and a related engagement angle. If the engagement angle rises,

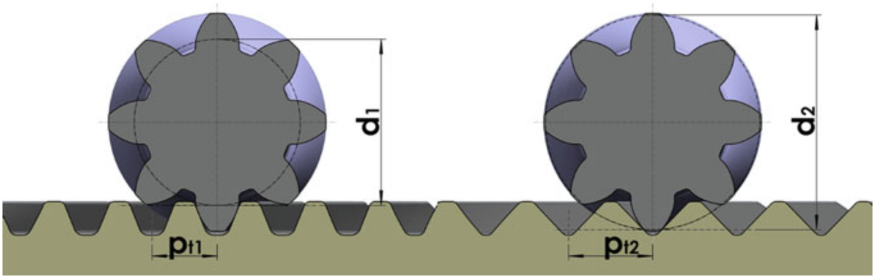


Fig. 11.14 The different pressure angles of the pinion for different pitch circle diameters

the contact lines are shifted towards the head of the respective pinion cog. The right side of Fig. 11.14 shows a rise of the engagement angle in the front cut. Accordingly, the contact lines between pinion and rack cogs are separated by the pitch circle diameter near the cog heads of the pinion. The chamfer angle β*z*2 of the rack dovetailing turns into *d*2 when the pitch circle diameter is large, because the chamfer angle of the pinion cogs rises with the diameter.

Figure 11.15 shows the different chamfer angles of the pinion for different pitch circle diameters. Under the influence of a small engagement angle and assuming an 'installation angle' γ of the gear, the axial division *p of* the rack cogs is:

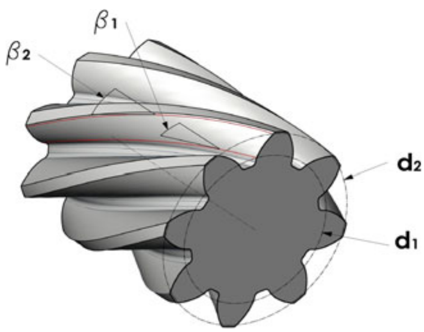
$$p1 = p_{t1} * cos(\gamma + \beta_{z2})/cos(\beta_{z1}) \quad (11.1)$$

The gear rack travel S_y per steering wheel rotation ('slope' or 'C factor') is, using the number of the pinion cogs *n*:

$$s_y = p_1 \ * \ n \quad (11.2)$$

As a result of the variation of the engagement angle (in the normal cut) of the rack cogs, the chamfer angle β*z* and the module *m* are non-constant (cf. Fig. 11.15).

Fig. 11.15 Different chamfer angles with different pitch circle diameters of the pinion



11.5.2.1 History of the Development

Variable gear ratios (VGR) are in use for sphere circulation gear since 1963. The development of the gear rack with variable dovetailing has lasted substantially longer. Although the first patents were submitted already in 1955, the actual development began with the patent of Henry Merritt in 1964. With Merritts implementation a spur gear pinion was mounted in 90° angles to the gear rack. This kind of variable gear ratio offered only a very small rise of the gear ratio of approx. 8 % in comparison to the basic gear ratio.

After different other patents for variable dovetailing and suitable advancements, Arthur Bishop introduced the modern variable gear ratio of a gear rack by means of his patented diagonal-interlocked evolving pinion. This kind of gear ratio was first applied within the scope of a mechanical gear for the Isuzu Piazza which came onto the Japanese market in 1981. The gear was produced in Japan by the gear manufacturer Jidosha Kiki Company (JKC). At this time, ZF also began with the use of variable gear ratios in the Opel Ascona, Ford Sierra, in the BMW 3 series (E30) and in the Fiat Daily 1985.

The initial areas of application were mostly limited to the reduction of the steering expenditure close to the steering stop unit. In the 90 s the variable gear ratio was used increasingly in the area of hydraulic steering. The first-time use of variable transmission ratios in connection with electrically powered steering column systems occurred in 2001 from TRW in the Fiat Stilo and from Koyo-Seiko with the Saturn Vue.

11.5.3 Applications

Gear racks with variable gear ratio became popular to master the conflicts in the gear ratio layout and to open new possibilities. Although the first applications were focussed on supplying enough gear rack force without exceeding too much influence on the manoeuvrability, later solutions concentrated upon the need for improved dynamic qualities and ergonomic advantages.

11.5.3.1 Applications for the Steering System

The first applications of the variable gear ratio concentrated primarily upon the mechanical advantage for the steering system. The whole ratio of a VGR steering system drops when the front wheel angle rises towards the stop unit. This phenomenon is a result of the actual steering arm radius rL which decreases by rotation of the front wheels from the straight ahead position to the full steer-angle. The gear ratio at increasing steer-angles can be reduced by the variable implementation while retaining a high whole steering ratio.

Mechanical gear—diminished torque

A basic challenge for the engineers of steering systems was to find a compromise for the steering ratio in the centre area of mechanical gears. It should limit the increase of the steering loads when the greatest steer-angle was approached and reduce the number of wheel turns to a satisfactory level. This task is complicated by rising axle and steering loads, caused by the chassis geometry.

Reducing the gear ratio for partially offsetting the higher loads during parking allows maintaining a higher gear ratio in the central position. This leads to a lower number of steering wheel rotations in comparison to a CGR system. Using a VGR in classical mechanical gears became less common during the past years. The latest application of a rack with variable dovetailing for a mechanical gear was found in the Smart, the MRR Roadster. The gear ratio course of the mechanical steering in this vehicle is shown in Fig. 11.16.

In contrast to hydraulic and gear rack-supported electromechanical gears, mechanical gears can be build smaller, however, on average they suffer higher pinion moments and load cycles. This results in increased forces in the dovetailing which can lead to wear problems. In this case, special attention needs to be paid to the cog sag and the contact tensions in the dovetailing.

Power-assisted gear—reduced power consumption

Power-assisted CGR systems often do not have enough power to move the rack into the stop position with the required steering angular velocity. High weight of the front axle, high steering loads or limited power of the power-assistance can be blamed.

A limited power supply can also be ascertained for some column- and pinion-supported EP systems. These systems are integrated into most vehicles with smaller front axle loads, when space and costs are crucial.

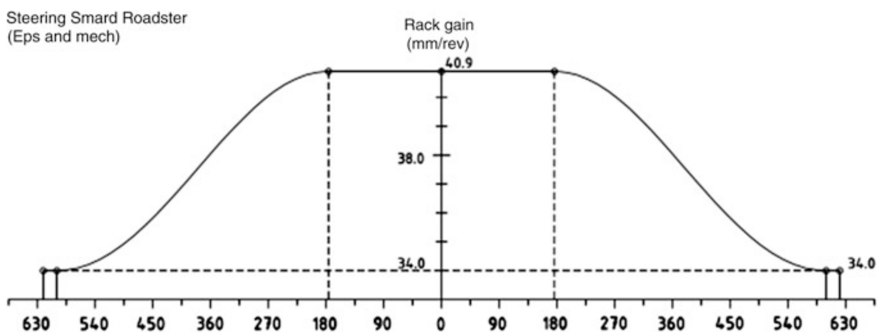


Fig. 11.16 Gear ratio course of the mechanical gear in the Smart Roadster. (*Source* Daimler)

The kind of variable dovetailing which is used for column- and pinion-supported EP systems resembles the gear ratio used in manual gears. They correspond to each other by a gear ratio that is lower farther away from the centre. The same favourably reduced pinion torque is achieved by this measure as in the manual gear. The loads on the dovetailing are, as expected, still higher in this kind of gears than in other models, because the whole driver and power-assist torque acts on the dovetailing of rack and pinion.

One purpose of the design of hydraulic or electric-mechanical rack systems is to limit the rack in its movement to the stop unit position. This reduces the power consumption and keeps it within the specified effective output of the power-assist system.

In all cases, the demands to the variable gear ratio can be plotted as a function of the steering geometry. The power and torque consumption of the power-assist system is thereby limited.

11.5.3.2 Driver Application

Power-assisted systems together with a VGR simplify car driving by suitable tuning. Advantages like easier parking, less high-speed nervousness or improved dynamic response of the vehicle are some of the characteristic features.

Ergonomics

In contrast to the reduction of the gear ratio with increasing steer-angle discussed above, a rise of the gear ratio is also conceivable, if the power-assistance is sufficient. The popularity of power-assisted systems allowed using the mechanical VGR for increasing the rack gain beyond the ratio desired at the central position towards the stop unit. The number of wheel turns, end stop to end stop, could be significantly reduced by that. This facilitates low-speed manoeuvres and parking. At the same time, there is less response to low corrections at the wheel during high-speed straight driving.

The change of the gear ratio increase usually began at a pinion or wheel angle between 30° and 90° and ended at a highest gear ratio at the angular range of 180°–270°. The driver can hardly discern this change of the rise, but its benefit is that the number of whole wheel turns, end stop to end stop, can be reduced to three or less. This prevented untimely tiring when driving in the city or on bendy roads, and it significantly promoted safe driving by the better manoeuvrability of the vehicle in such situations.

The first time this kind of variable dovetailing was applied was in the Ford Sierra in 1985. The use of this kind of gear spread only slowly during the 1990s, but it is common among higher-class vehicles of the 21st century. This solution also prevails among SUVs, because these cars have a rather high rolling axis, so that a direct, concentric gear ratio could produce instabilities (cf. Baxter and Heathershaw 2002).

Till the end of the 90s, the number of suitable processes to manufacture high-precision VGRs rose, so that these systems are now commercially accessible to all manufacturers of gears.

Vehicle dynamics

The development of the CGR had progressed till the beginning of the 21st century, and further benefits could be used. The economic VGR was made purely with the driver's needs in mind. The demand for a much higher rack gain within a shorter travel of the rack was rising. This enabled improving the vehicle's dynamic driveability.

The introduction of the Active Front Steering (AFS) by BMW in the series 5 of 2003 has shown that a change of the steering ratio improves the yaw rate response at low to medium speed clearly, even though the costs are high. The AFS system uses a mechatronical steer-angle superposing system to provide a VGR depending on speed. The angle-sensitive VGR can be called a not quite comparable, but cheaper alternative, according to Heathershaw (2004).

A relation between speed and the limits of the corresponding steer-angle change was first examined in the middle of the 1950s by Arthur E. Bishop at a conventional vehicle. On this basis, taking into account the typically generated lateral acceleration of 0.2–0.5 g, an angle-sensitive VGR could be made that had almost approached the characteristic curve of the yaw rate found for vehicles with a speed-sensitive VGR system. Since such a system is not speed-sensitive, it is very important that the gear ratio in the 'straight ahead' position is accordingly indirect, so that the high-speed steering feel is as expected.

The purpose of this measure is to make the change of the steering ratio discernible by the driver. This approach deviates from the principles of an 'ergonomic' VGR. A rack gain of 25–50 % over the medium VGR and, typically, between 90° and 120° of the first wheel turn is achieved by this principle. The quite large changes of the rack gain produce a high curvature of the rack cogs and their flanks, putting high demands to the accuracy of their shape.

The described kind of VGR was first used by Daimler in 2008, in the SL, SLK and CLC series within the scope of the Daimler "Direct Steering". Proper implementations of the gear ratio, the whole gear ratio and the variable dovetailing are shown in Figs. 11.17, 11.18 and 11.19.

11.5.3.3 Special Applications

VGR gears can be used as well for combined requirements, for special gear ratio purposes or for the correction of asymmetries in the steering geometry.

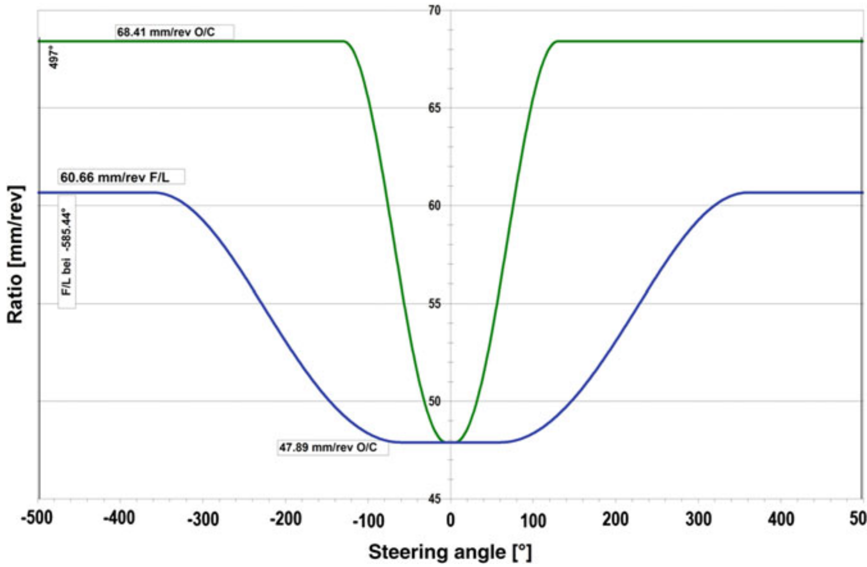


Fig. 11.17 Gear ratio of a standard VGR steering (*blue*) in comparison to Daimler's 'direct steering' (*green*) in the Mercedes 2009 ML class. (*Source* Daimler)

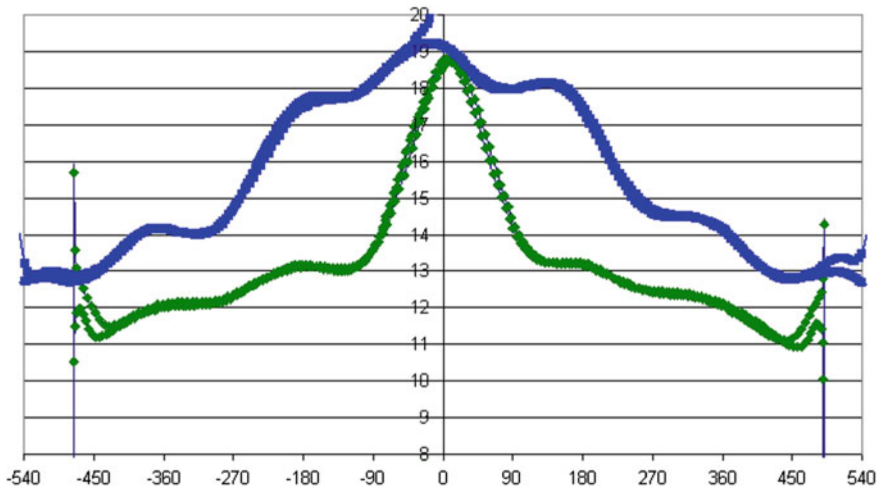


Fig. 11.18 Whole steering ratio of a standard steering with VGR (*blue*) in comparison to Daimler 'direct steering' (*green*) in the Mercedes 2009 ML class. (*Source* Daimler)

Combined applications

There are applications which demand a combination of driver- and system-related requirements for a VGR. The introduction of a driver-related VGR can be used for

Fig. 11.19 Bishop VGR—dovetailing of the Daimler 'direct steering' in the Mercedes 2009 ML class. (*Source* Daimler)

a rack gain between central area and highest steer-angle (stop unit position). As discussed before, this produces a too high power consumption of the power-assist system. An overloading of the power-assistance can then only be remedied by reducing the gain until the end stop position.

To meet these sometimes opposing requirements, some manufacturing processes enable two or more gear ratio changes within a dovetailing. The restrictions to construction and production of a variable dovetailing set the highest and lowest gear ratio level for a dovetailing. The highest gear ratio is selected for the area beyond the centre. The lowest ratio is either in the centre position or in the area of the end stop position (or in both of these areas, if the ratios are equal).

An example for three ratios in one VGR is the M-curve VGR of the Opel Corsa of 2006. Its name is derived from the M-like shape of the gear ratio curve. Renamed the 'Progressively variable steering', the M-curve VGR was used for the sports models of the Opel Corsa 2006. Proper M-curves of the steering ratio and gear ratio are shown in Figs. 11.20 and 11.21. In addition, Fig. 11.20 compares the characteristic curves of a CGR and a VGR.

Asymmetrical and special steering ratio purposes

The VGR can be used to solve problems related to the steering geometry as well. One particular challenge is the occasional deviation of the steering ratio from the

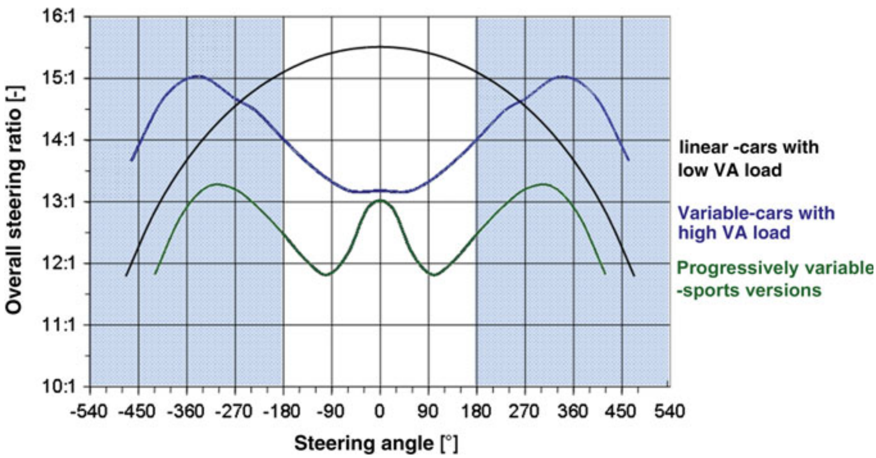


Fig. 11.20 Steering ratios in the Opel Corsa: CGR for petrol car, VGR for diesel and 'M-curve' VGR for 'progressively variable steering'. (*Source* IKA-RWTH)

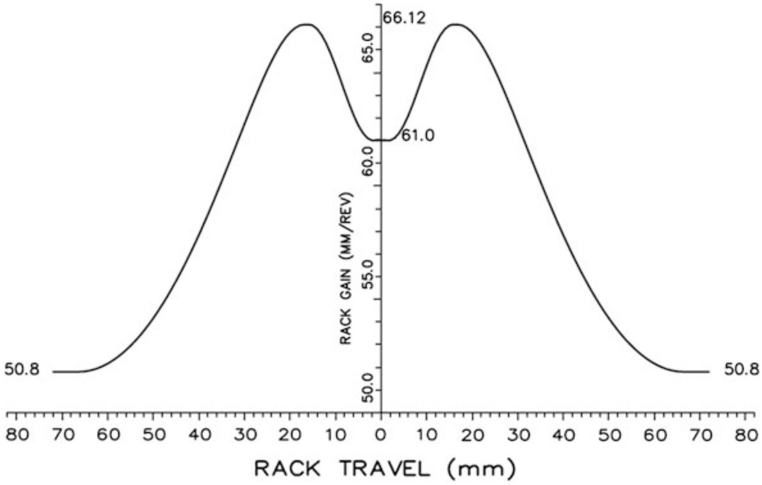


Fig. 11.21 Gear ratio of the Opel Corsa M-curve VGR for the progressively variable steering. (*Source* IKA-RWTH)

desired curve. A VGR that adapts to a nominal curve can be developed based on the central gear ratio and the steering geometry, within the mechanical limits of the system.

Some steering geometries are distinctly asymmetric, despite the use of a CGR or a symmetrical VGR. This produces an asymmetrical steering ratio curve. In this case, an asymmetrical VGR is used to offset the asymmetries, so that a quite symmetrical steering response can be achieved.

11.5.4 Technical Limits

A variable steering ratio is achieved by changing the standard engagement angle of the engaging rack and pinion cogs. The theoretical least and peak gear ratios are defined for a specific pinion module when the least and peak values of the engagement angle are set in the normal cut as upper and lower limits.

A conventional CGR version has a dovetailing optimised for contact engagement, stability, friction, tooth root stress and firmness. This task is more complicated for a VGR, because a satisfactory design has to be found for any normal engagement angle used in the VGR's pinion geometry.

11.5.4.1 Size of the Rack Gain

The absolute limits for a rack gain are defined by the production process. Nevertheless, some limits are also due to the product development which defines the 'constraints'. Early developments intended a least engagement angle of only 14°. These low angles have two shortcomings, though: One is a trend to wedge under load, i.e. the dovetailing of the pinion wedges together with the cogs of the rack, and this can cause excessive friction and high wear. The other is the higher accuracy of the cog flank surfaces required when the engagement angle is reduced, to achieve the same accuracy in vertical engaging. Concerning the rack gain and the cog engagement, the vertical portion has to be controllable in any case of deviation, because vertical deviations of the surfaces have a considerable impact. The horizontal component contributing to the rack gain is less important. Reducing the engagement angle entails higher requirements for accuracy but limits a further gain of the rack ratio. As a result, a lower limit of the normal engagement angle of 20° is used in reality.

The upper limit of the engagement angle is defined by the gear application. The helical pinion transfers forces on the cogs of the rack which can be divided into axial, transversal and vertical vectors. Accordingly, the general purpose in designing a rack-and-pinion steering is to maximise the axial component of this force, in particular for CGRs. Figure 11.22 compares pinion force vectors with low and high dovetailing engagement angles.

Variable dovetailing always sees deviations occurring at the cogs of the rack with regard to these force vectors. One tries to minimise the vertical force vector when higher pinion loads shall be transferred. A high vertical force component will also see high forces acting on the rack support and the rack yoke, provoking high friction loads. Due to these considerations, rack yoke contact surfaces with low friction coefficients are used for rack-and-pinion systems with high vertical force components (see Sect. 11.4.3.2).

11.5.4.2 Contact Lines

The engagement angle also influences the length of the contact lines between the pinion and the rack cogs, a measure of the dovetailing quality. This is described by

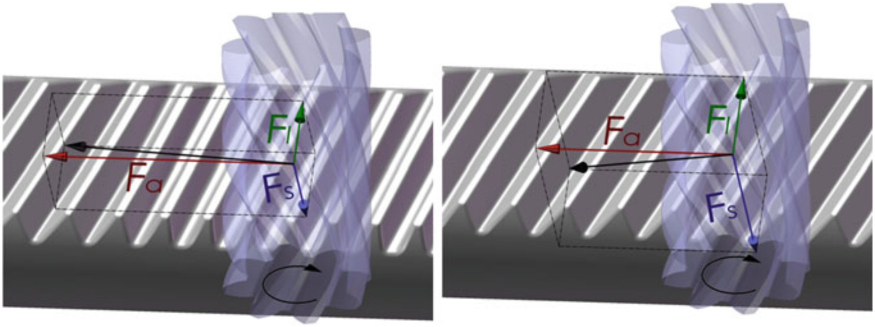


Fig. 11.22 Pinion force vectors with low and high dovetailing engagement angle

the overall engagement εγ, the total of tread engagement εβ and jump engagement εα, which results from the helical dovetailing. The whole engagement supplies the average number of cogs in contact. The theory of dovetailing calculation for involute gears is documented in DIN 3960.

A typical tread engagement decreases quite quickly when the normal engagement angle rises, while the jump engagement rises only minimally.

Different full engagements are used as standards for different kinds of steering systems. Concerning VGR design, it is ensured that these standards are observed.

Hence, a bigger gain of the gear ratio uses a bigger dovetailing area than a constant gear ratio, to maintain adequate full engagement. Implementing the desired rack gain may require compensating the reduced tread engagement by a bigger jump engagement (due to a bigger dovetailing width), to receive a satisfactory function.

11.5.4.3 Stability and Noise (NVH)

The power between pinion and rack dovetailing is transmitted along the contact lines. Two or more cogs are usually in contact. If the pinion is turned and the rack is driven, the contact lines move along the rack cogs. As long as there is always at least one contact force vector of a contact line acting on each side of the rack dovetailing's axis of rotation, the rack is prevented from rolling. In this case, the active torques of the contact force vectors have the same amplitude and opposite directions. Figure 11.23 shows how the contact lines of adjoining cogs move when the pinion is rotating. This rotation shifts the contact lines and their resultant forces.

Pinion geometry, engagement angle, chamfer angle of the pinion cogs and the distance of the rack dovetailing to the axis of rotation can cause conditions in which all the contact forces generate parallel torques around the rack's axis of rotation. The rack is then in an unstable state and inclines to slight turns around its

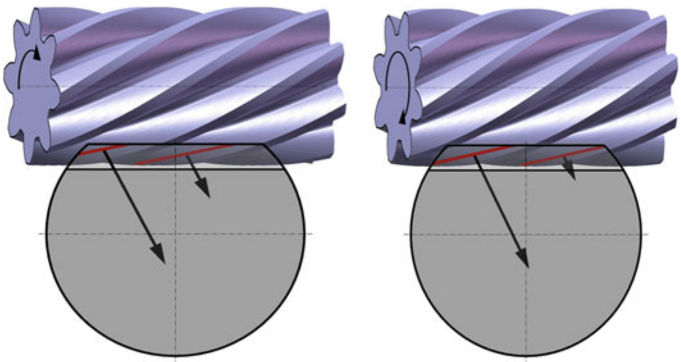


Fig. 11.23 Contact line movement of adjoining cogs when the pinion rotates

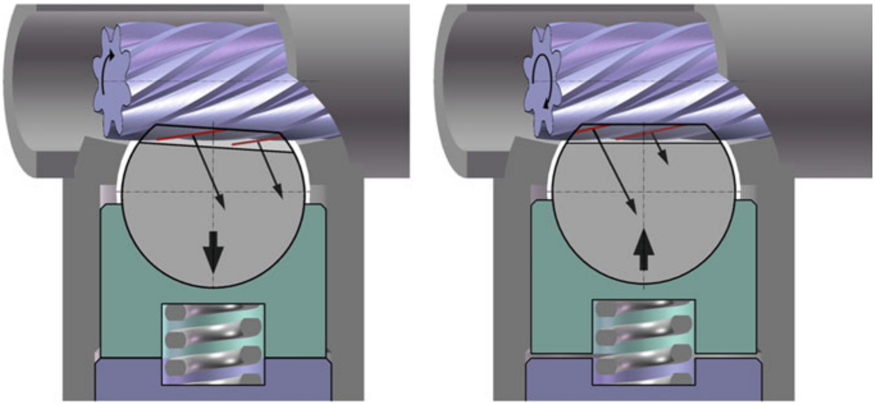


Fig. 11.24 Rotation of the rack by unstable contact line vectors and a shifting rack yoke (*left*), stable contact line vectors (*right*) upon full pinion contact and rack yoke clearance

axis of rotation. The rack is supported in the gear case by a rack yoke pressing its dovetailing against the pinion dovetailing. As was discussed in Sect. 11.4.3.2, the rack yoke clearance enables the offset of the dovetailing tolerance and permits a slight rolling of the rack under high loads.

Figure 11.24 (left) shows the field line vectors that cause the instability and thus the turning of the rack. The spring-loaded rack yoke is pressed against the adjuster bolt attached in the gear case. When the pinion turns on, the next cog is engaged and an offset torque is created around the rack axis (see Fig. 11.24 on the right). The rack turns back into the initial position of the full engagement, restoring the rack yoke clearance. Except for all the other forces that are acting on the rack in this case, the movements described in this unstable state can contribute to a noise in the gear [sic! Why "except" (mit Ausnahme)?]. The change of the torques in the engagement kinematics can also produce transmission errors in the rack gain and changes in the axial rack force.

A short engagement distance of rack cogs and pinion or rack cogs with a higher axis of rotation are indicators for an unstable dovetailing engagement.

Figure 11.25 shows that the sliding friction between the surfaces of the rack and pinion dovetailing affects the angle of the contact force vector. The pinion torque (on the driver's side) produces a greater vertical component of the contact force vector (shown in blue), while the rack force from the tie rods generates a lesser vertical component (shown in green). The result reveals a higher risk of unstable engagement and, hence, expectable noise from excitations of the chassis. The power flow from the road, as for example upon road kickback, should be considered more critically than the power transmission from the driver.

Using a VGR means that the engagement geometry has to change. Therefore, it is ensured that minima and maxima of the gear ratio curve do not make unstable dovetailing engagement more likely.

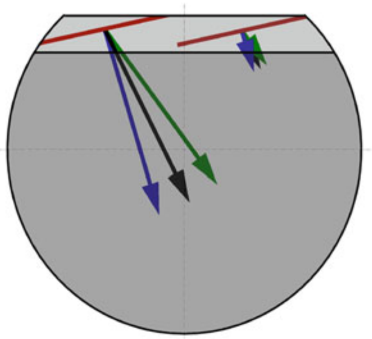


Fig. 11.25 Effect of the friction on the contact line vector forces (unstable situation during recoils [*green*])

11.5.5 Manufacturing Processes

Since VGR racks were introduced in 1981, many different processes have been used in serial production. The main requirement for a suitable standard process is to make a dovetailing with three-dimensionally curved, non-prismatic cogs. Vertical and transversal curvatures should be possible. High curvatures of surfaces or 3D shapes require more accuracy and quality of the dovetailing.

The first process, introduced by JKC, Japan, in 1981, was a combination of hot forging, deburring and cold stamping. Although this process was used for lots with few pieces for more than two decades, it suffered a significant decarbonisation of the cog surfaces as a result of the hot forging, and that interfered with the wear-resistance. Low service life of the tools and the risk of imprecise adjustment of the parts during calibration added to the disadvantages of this production mode.

Only three processes for making variable dovetailing are still in use.

11.5.5.1 Rotary Swaging

Rotary swaging was developed by a collaboration of ZF Friedrichshafen and Heinrich Schmid AG in early 1980 and implemented in production after 1985. A circular movement of the upper cog tools generates a gradual reshaping of the material. This circling is limited to a swinging parallel to the rack axis, to produce the rack dovetailing. The basic material is submitted to a special soft-annealing process to achieve the desired reshaping without fissures. At the same time, the blank is covered with a graphite lubricant to extend the service life of the tool.

This kind of production leads to excessive displaced material gathering in burrs on the flanks of the dovetailing, that need to be removed in a subsequent step. If a humpback rack (D-form rack) is made, material is machined away before reshaping, to avoid any distortion of the rack during the swaging process. The cold solidification of the rack's surface is exploited to compensate the low firmness of the raw material.

The described process can be used to make humpback racks or Y racks. Figure 11.26 compares the cross sections of both rack backs as an example.

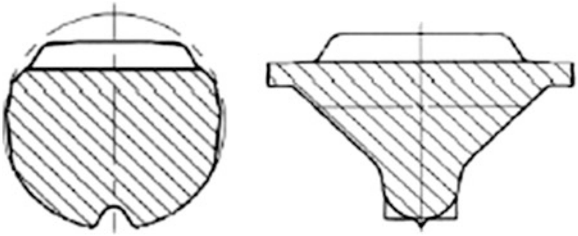


Fig. 11.26 Typical swagged rack cross sections. (*Source* Heinrich Schmid AG)

An offset of the swinging of the upper cog tool has to be considered during the construction of the tool. This limits the highest curvature of the cog flanks that can be achieved with this tool, a curvature needed nowadays for the dynamic implementation of a VGR.

11.5.5.2 Semi-hot Forging

To handle the decarbonisation problem, semi-hot forging was introduced by James N. Kirby Ltd. and Bishop in 1982, though calibrating and cold stamping were necessary parts of this process, too.

Bishop introduced a semi-hot forging process for Y racks in 1984; it did not need calibration any more. In the following years, this approach was further pursued, until at last a new process became available in 1994. The first VGR rack forged by this process, without burrs, was used in the mechanical steering of the Fiat Punto series.

A single forging process is sufficient to produce the VGR dovetailing at a temperature of 700–800 °C. This manufacturing method now allows achieving a high curvature of cogs and flanks with the required accuracy. In comparison to other processes, a good surface structure is achieved, internal tension is avoided and decarbonisation of the cog flanks is prevented. This process, which requires no further treatment steps in the cog area, provides a high precision shaping of the dovetailing, enabling the highest possible gear ratio range and the fastest possible rack gain.

Bishop developed this process, and the first variable dovetailing at a humpback rack was forged in 2005. The new process allowed producing a bigger variety of cross section shapes to widen the dovetailing, larger cross sections (in comparison to the shaft area) and all kinds of D, Y or V rack backs in the dovetailing area. Figure 11.27 presents typical semi-hot forged rack cross sections.

As described in Sect. 11.5.4.2, a significant rack gain often requires bigger and broader cogs to achieve a contact engagement which is favourable for the application. Bishop's semi-hot forging process enables the making of racks with dovetailing widths of up to 115 % of the rack diameter.

Before the actual forging, the blank is heated, and after forging, it is cooled down to produce the desired material properties. An exact monitoring of the heating and cooling is essential. All common chemical compositions and states of

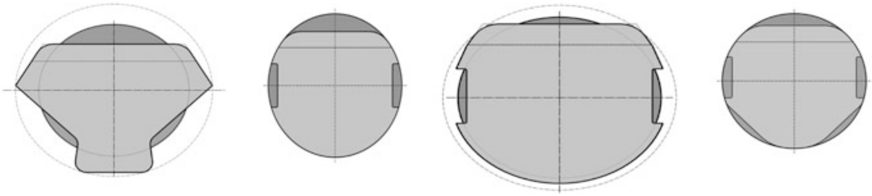


Fig. 11.27 Typical semi-hot forged rack cross sections. (*Source* Bishop Steering Technology Limited)

Fig. 11.28 Examples of D and Y 'ActivRak™' racks

delivery can be forged out of steel now, a fact which is very beneficial with regard to the currently more frequent EP systems. Figure 11.28 shows examples of recent D and Y rack cross sections with variable dovetailing.

11.5.5.3 Pipe Reshaping

A different approach to making racks was pursued in Japan during the mid-1980s. A reshaped pipe was taken as a blank to develop a lightweight version of a rack. The material in the middle of conventional 'solid' racks, near the rack axis, contributes to the rack's weight, but little to its bending strength. Hence, this material is in many cases removed by deep drilling to reduce the weight, at least in the non-dovetailing area of the rack. Making racks from pipes is therefore an elegant solution, weight and efficiency of use being the prime motivations.

In contrast to rotary swaging or semi-hot forging, as described above, pipe reshaping is achieved by inserting spikes on the inside. The pipe is preformed to increase the wall thickness in the cog area at some local places. Then the pipe is fixed between stamps, the desired cog form is contained in one half of the stamp. Several cold spikes (up to 16) are pushed successively through the pipe, to press the material gradually into the cog tools. Figure 11.29 shows a cutaway view with the pipe rack in the dovetailing area.

Fig. 11.29 Typical pipe rack cross section, here of a Mazda 6

The material necessary for pipe reshaping contains less carbon than most other rack applications that commonly have a content of 0.30 %. The pipe is fully hardened and tempered after shaping and treating, so that it receives the necessary bending strength. However, the surface resistance is lower than that which competitive processes and materials might achieve.

The pipe reshaping process was first used for CGR gears in hydraulically driven power steerings. But VGR dovetailing can also be made by this method. However, the cog width is then much lower in comparison to the forging processes, because the rack material the cogs are made from has to be pressed from the inside of the pipe outward into the cog mould. Accordingly, bigger pipe diameters have to be selected in order to achieve comparable widths and accuracy of the cog shape.

11.6 Requirements of a Mechanical Rack Gear

This chapter discusses the general requirements of function and firmness of mechanical rack gears, proven by bench tests. Before the actual test, the gear is submitted to a startup process, to anticipate any potential sagging effects and to support the test results.

Tests checking essential functional aspects of the gear are already carried out as soon as the gear is assembled in the production line. More extensive tests are performed at typical, randomly selected parts from a batch of gears (auditing). Firmness, wear and environmental tests on the other hand, are part of the development programme. After the release for series production, they will only be repeated at defined time intervals (e.g., once per year) as accompanying tests.

11.6.1 Functional Requirements of Gears

All function tests are carried out at a gear without tie rods in built-in position at room temperature (20° ± 5°). Some vehicle manufacturers, upon defining the functional requirements, make a distinction between the comfort area, with a steering wheel angle of ±180° or a rack path of ±25 mm around the central rack position, and the remaining dovetailing area.

11.6.1.1 Rack Yoke Clearance

Although the limitation of the highest rack yoke clearance (e.g., 100 μm max.) is in fact an indirect functional requirement, it is crucial for other functional requirements or the friction and noise response of a rack-and-pinion gear (see also Sects. 11.4.3.2 and 11.5.4.3). The widest clearance is usually defined at the central position of the rack and measured by a suitable drilling in the adjuster bolt or the yoke cover. A linear receiver records the relative movement of the rack yoke against the gear case. The reference point for the rack yoke clearance is set by driving the rack yoke into touch with the cover, a deflection torque of 5–10 Nm without lateral forces around the longitudinal axle of the rack is applied. The unloaded rack yoke then rebounds, and its relative movement is recorded by the linear receiver, resulting in the value of the rack yoke clearance, see Fig. 11.30. This clearance can also be recorded as a function of the rack's position. If the values are very small, then a rising friction at the rack is likely. If, by contrast, the upper limit value of the clearance is surpassed, there is a risk of steering rattle in the car.

11.6.1.2 Steering Pinion Torque

To measure the steering pinion torque, a rotation without tension or free travel and with constant rev is applied at the input shaft. There are no loading elements at the

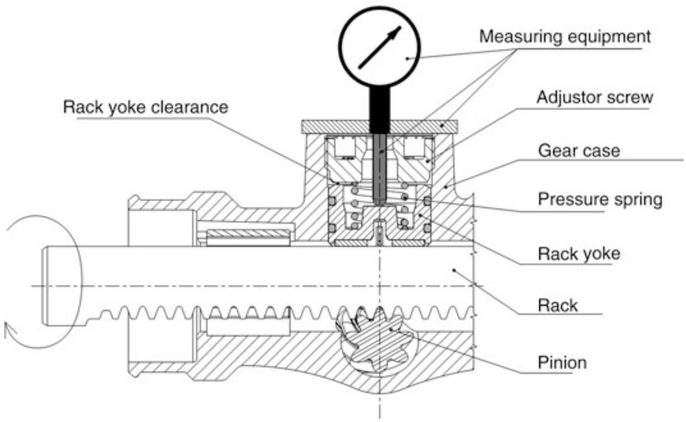


Fig. 11.30 Identifying the rack yoke clearance by applying a deflection torque without lateral forces to the rack

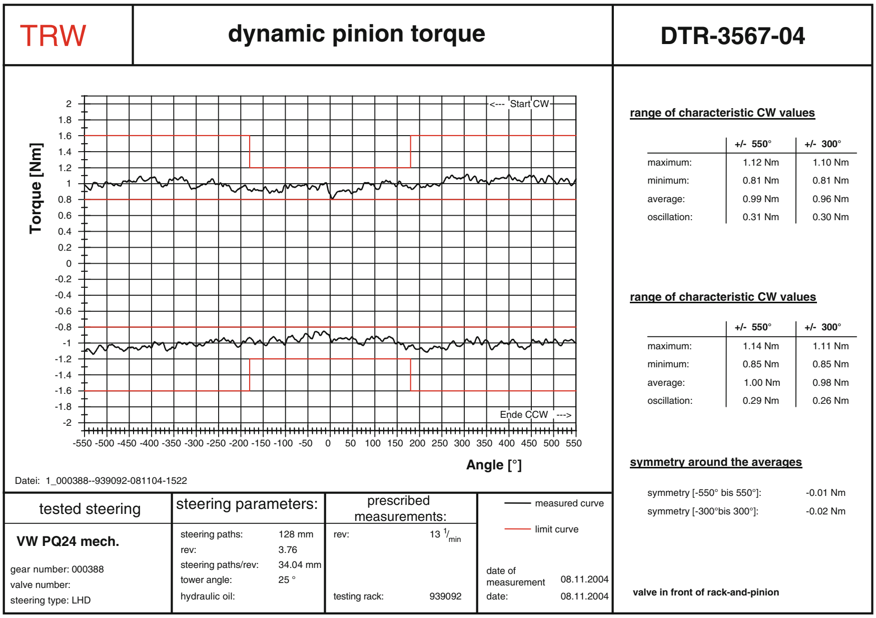


Fig. 11.31 Pinion torque of a mechanical rack-and-pinion steering

rack, so that it moves freely. The rotary angle and the torque at the input shaft are recorded over a measuring cycle.

As the minimal and maximal values of the rotary torque and its oscillation over the stroke are predefined, the test is counted as failed if these values are surpassed. A typical value of the pinion torque is in the range of 0.8–2.0 Nm, commonly achieved when the pinion has a rev of 15 rpm. Figure 11.31 shows the result of a measurement of the pinion torque, moving left or right and remaining inside the specified range. The highest permissible pinion torque that the specification of this steering permits is accordingly marked in the image and should be less than 1.2 Nm in the comfort area while staying below 1.6 Nm beyond the comfort area.

11.6.1.3 Rack Displacement Force

A linear drive is attached to the rack strainlessly and without free travel. There are no additional loads at the input shaft, so that the shaft moves freely when the rack is shifted with a steady displacement speed of 5–10 mm/s, according to specification. Rack forces of 150–350 N are permissible. The rack travel and the rack force is recorded over the measuring cycle. If the highest measurement value of the displacement force exceeds the defined limit value, the test is considered failed. If demanded, the oscillation of the displacement force is evaluated as well.

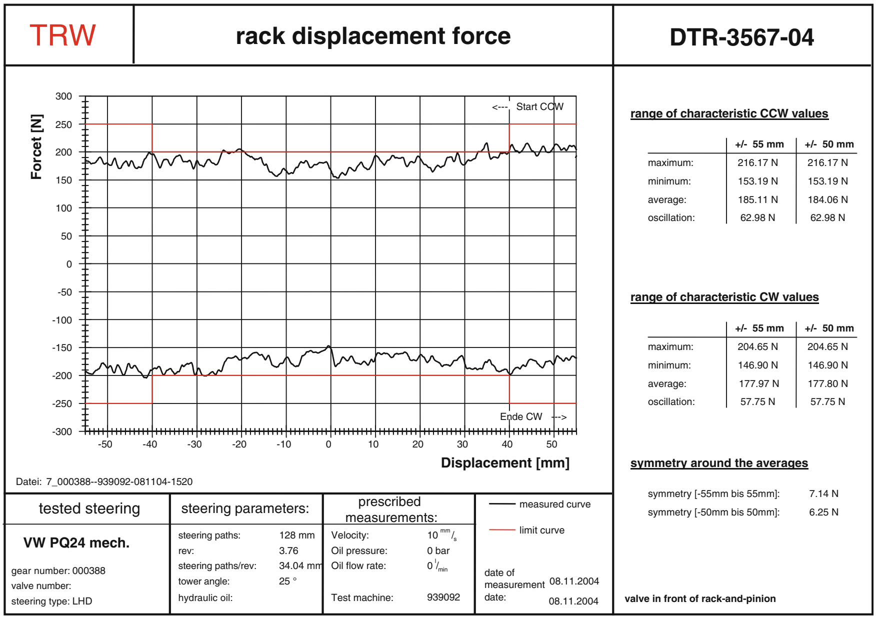


Fig. 11.32 Rack displacement force of a mechanical rack-and-pinion steering

The rack displacement force of a mechanical rack-and-pinion steering is applied over the rack path in Fig. 11.32. In this case there were values of up to 200 N permitted for the central area of the rack; 250 N may be achieved in the outer dovetailing area. The rack displacement forces of mechanical gears are usually a little higher than those of rack-supported power-assisted gears. The reason is that all the rack forces of mechanical gears are directly transferred over the pinion/rack link. High input steering torques generate high forces in the dovetailing, the major part of which is acting on the rack yoke. The high pre-tension of the rack yoke demanded thereby and a gliding foil corresponding to the forces influence the rack displacement force.

11.6.1.4 Efficiency

Some vehicle manufacturers require information about the efficiency of the gear, distinguishing direct and indirect efficiency. For the direct gear efficiency, one observes the conversion from the rotating input shaft into the translation of the rack, i.e. the ratio of rack force to pinion torque achieved as the pinion is moving. For indirect efficiency, also called reset efficiency, the movement transferred by the chassis on the rack is examined, i.e. the quotient of accessible pinion torque and rack force at the moving rack is computed. The direct efficiency established from the rack shift of a typical mechanical rack-and-pinion steering is shown in Fig. 11.33.

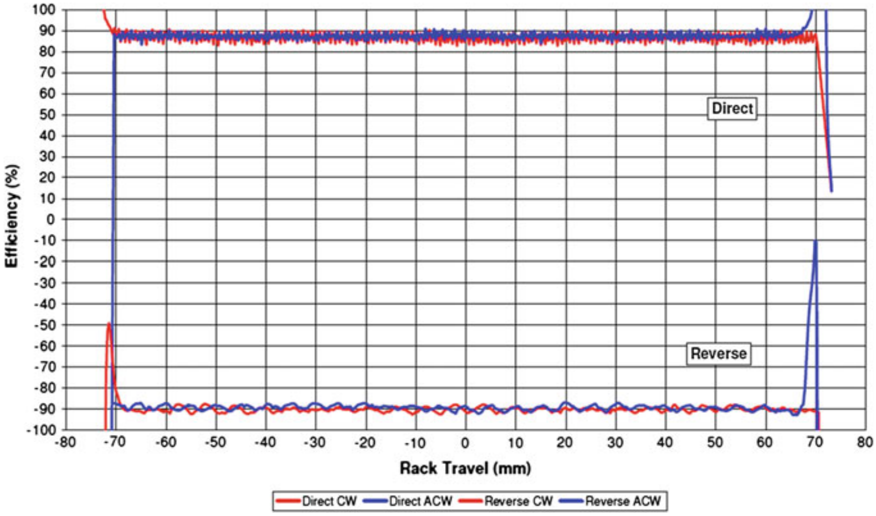


Fig. 11.33 Direct and indirect efficiency of a mechanical rack-and-pinion gear

The result confirms the standard values for the direct efficiency of a mechanical rack-and-pinion gear which, according to the load, are found in the area of 85–92 %. The indirect efficiency of the mechanical gear is often a little lower; this depends on the dovetailing design.

11.6.1.5 Noise (NVH) Requirements

The general Noise, Vibration, Harshness (NVH) requirement of a mechanical rack-and-pinion gear is the 'absence' of any annoying rattling noise. Hence, steps to avoid noise are considered already during the design of the gear. Calculations and simulations support these considerations. As soon as the first prototypes are available, hardware examinations on the test bench follow. The gear is fixed on the test bench and realistically loaded. Responses according to frequency and excitation amplitude occur at the bearing points of the rack which are then evaluated for their practical relevance. If there is any need for improvement in the area of the rack yoke, remedial actions as discussed in Sect. 11.4.3.2 may be examined and implemented as, for example, the change of rack yoke clearance and spring stiffness or the insertion of O rings. A successful tuning is confirmed in the vehicle test.

11.6.2 Strength Requirements of the Gear

Examination of the static and dynamic strength is decisive for the strength requirement.

11.6.2.1 Static Strength

The static strength of a mechanical rack-and-pinion gear is confirmed chiefly by the 'fracture torque test' and the 'Charpy impact test'.

Fracture Torque Test

The focus of this test is the check of the rack's static strength, pinion and case. The parts are tested until failing by applying a torque to the input shaft both ways, without any lateral force, while the rack is blocked. The test is passed if at least one defined torque level was achieved before the torque drops and the part in the steering fails. The requirements are usually in the range of 250 Nm.

Charpy Impact Test

The impact is directed onto the gear by a tie rod above the ball pivots of the outer joint. The position of the tie rod corresponds to the built-in location at full deflection. The rack is fully extended on the load side. The moment of inertia of steering column and wheel is applied by a substitute mass at the input shaft. The requirements to the gear are met if the full effectiveness was proven for each load layer and no part has fractured.

11.6.2.2 Dynamic Strength and Wear Test

Parking and wear tests are carried out within the scope of testing the dynamic strength.

Parking Test

The parking test is a bench test which applies high rack loads, to simulate the service life load of a gear within a short time. This test is an important part of the test programme for mechanical rack-and-pinion steering, because it yields statements on service life for almost all the parts. The input shaft is driven over the rack stroke with constant speed. A test load equivalent to the highest operation load of the steering opposes the rack's direction of movement. Equal operation loads are applied at both outer joints, right and left. At the end stop, the axial joint touches the gear case. The torque at the input shaft is raised to the prevail torque and kept at this value. Then the rotation of the input shaft and the direction of the load at the tie rods is reversed.

Wear Test

The wear test is performed on a test bench that simulates the service life load of the gear in a rather short time, because of the level of the initiated forces and torques. Primarily, parts like pinion, rack, bearings and the internal fouling of the gear are tested by abrasion.

The input shaft is driven against a test load at the rack, the speed should be constant over the rack stroke. The test load is applied to the outer joints of the tie rod and the input shaft of the steering. At the end stop (stroke end position), the axial joint touches the gear, enabling a gain of the torque. When the peak torque is reached, the rotation at the input shaft and thereby the direction of the rack force is reversed.

If after the wear test the functional values of the steering do not correspond to the given specification any more, the test is failed. A rack yoke clearance of 0.5 mm or higher also produces a fail result of the test.

11.6.3 Environmental Requirements of the Gear

Environmental requirements are met if the salt spray test and the dirt water test were passed in an environmental simulation.

11.6.3.1 Salt Spray Test

For the salt spray test, a steering is sprayed with different NaCl solutions, compliant with DIN ISO 9227. The following requirements apply:

- no base metal corrosion
- absolutely water-tight
- effective after the test.

11.6.3.2 Dirt Water Test

This test is performed at a completely assembled steering which is repeatedly driven against either end stop by turning the pinion. This test includes two operating modes, consisting of spraying the steering and a subsequent dry run. For spraying, the whole steering including the tie rods is sprayed with dirt water at zero pressure. The dry run heats the gear with hot air. After the test, the gear has to be absolutely sealed, as in the salt spray test.

11.7 Design Verifications and Product Validation of a Rack-and-Pinion Gear

The release procedure and the tests to be finished for release are parts of the specifications. The releases occur at important points of the development programme.

The entire release procedure is coordinated with the customer (OEM), before the gear supplier is selected. This release procedure becomes a compulsory part of the specifications. Only steerings that passed all the tests described in the test specifications can receive an internal product release from the steering manufacturer.

The purposes of these tests and the requirements of the respective samples are discussed in the following.

11.7.1 Concept Verification

The purpose of this test is to verify product features qualitatively (the product may be used without safety risk) and quantitatively (the demands of the specifications are kept). The concept verification (CV) phase is marked by loops in the development, finished when the design is ultimately fixed.

11.7.2 Design Verification

The purpose of the design verification (DV) tests is to corroborate product features of samples made with substitute tools. These tests confirm that the product design matches the demands for quality. The tests are carried out after the final definition of the design, no further loops will be passed.

11.7.3 Product Validation

The purpose of the product validation (PV) test is to ratify product features of samples in series production (industrial manufacturing). When the samples are made, the parameters of the later series production have to be observed precisely. This includes the manufacturing of parts, assembly, machines and tools used, procedure sequence and quality control.

The PV test (and the DV test) are conceived to check the product features concerning

- Static strength
- Dynamic strength
- Wear-resistance and
- Environmental resistance.

The size of the applied test loads depends on the operation loads expected for the product. The release programme is passed when all requirements of the relevant test standards were met within the defined scope of validation.

11.7.4 Accompanying Test

The purpose of the accompanying test is a random check of product features which cannot be monitored by the quality control during production. These features are commonly described in the generic product specification. The quality of the supplied parts and the stability of the assembly process are controlled by these tests. Hence, the samples have to be made compliant with the defined and released process parameters.

11.8 Hydraulic Steering Systems

The hydraulic power-assisted rack-and-pinion steering is an advancement of the manual steering into which additional elements were introduced to amplify the rack force. Figure 11.34 shows that it consists of a mechanical part, very similar to the mechanical rack-and-pinion steering described in Sects. 11.4 and 11.5, and a hydraulic part including the components for force generation (cylinder/piston), steering (valve) and transmission of the auxiliary energy (transfer lines and hydraulic connections).

The effective rack force of this steering model is generated by a superposition of mechanical and hydraulic force:

$$F_R = F_{mech} + F_{hydr} = \frac{M_H}{r_{Pi}} + \Delta p_C(M_H) \cdot A_C \tag{11.3}$$

11.8.1 Objectives

11.8.1.1 Reducing the Steering Forces

The main argument for the development of power steering was the reduction of the necessary steering forces, esp. when the vehicle is at rest, to improve the driver's

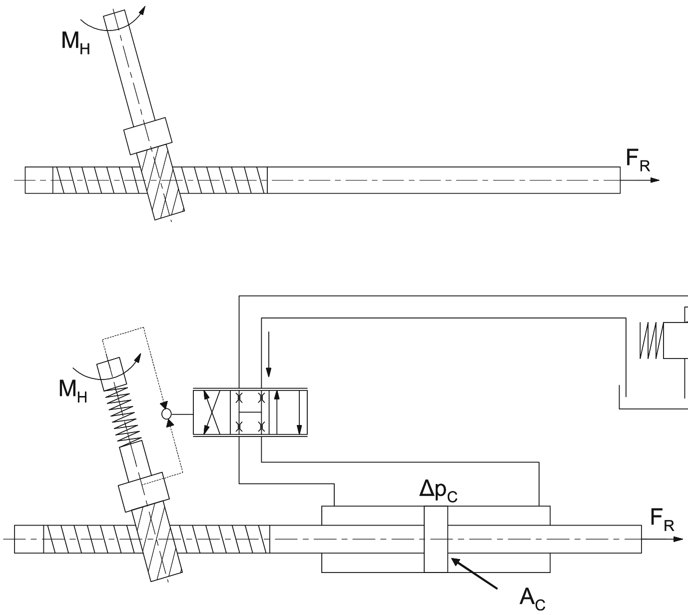


Fig. 11.34 Design comparison of mechanical and hydraulic steering

convenience. The demand for this discharge of the driver has grown with rising front axle loads and the resulting higher rack forces of the steering.

11.8.1.2 Reducing the Steering Ratio

The possibility to reduce the steering ratio was also exploited early on. For manual steerings, it has to be high, so that the steering forces are kept convenient for any driver and within the scope of the registration regulations (ECE regulation R79). Such a design often prevents fast changes of course at low speed, because the driver is not physically capable of applying the necessary wheel angles fast enough. The reduction of the steering ratio in power steerings improves the steering convenience and safety, because, for example, quick evasive manoeuvres in city traffic are better supported. The tendency to further reduce steering ratios or to employ VGRs that are dropping beyond the central position of the rack is ongoing and has become an important objective for power steering.

11.8.1.3 More damping of the steering system

Another purpose is to increase the damping of the steering system for various appearing vibrations and to suppress interferences initiated in the steering system by the road. One particular objective is to integrate the additional elements that some mechanical gears may need into the power steering, so that the add-on costs of the power steering are partially compensated.

11.8.1.4 More Freedom in the Design of the Chassis

A power steering is standard equipment in most vehicle classes, hence, the possibility to avoid compromises in chassis design in favour of lower steering forces is exploited as well. This concerns, e.g., the definition of the castor and the choice of the tyre thickness.

11.8.2 Necessary Changes in the Vehicle Opposed to Manual Steering

The package of the hydraulic steering has to accomodate some additional components by which the gear loses compactness (see Fig. 11.35). One is the steering valve on the input shaft, extending the least distance between rack axis and attachment of the gear to the intermediate steering shaft. Furthermore, the steering cylinder has to be arranged in the extension of the rack axis or parallel to it.

The auxiliary power supply, either a part of the secondary drive unit of the internal combustion engine or an electrically driven engine/pump aggregate, and its connection to the power steering by pipes and hoses, entails more profound changes.

On the other hand, compromises in the design of the front axle geometry in favour of low steering forces are not required any more.

11.8.3 Necessary Changes in the Gear Opposed to Manual Steering

The gear is designed for much higher operation loads. This is needed because of the generally higher steering forces resulting from the design of the front axle

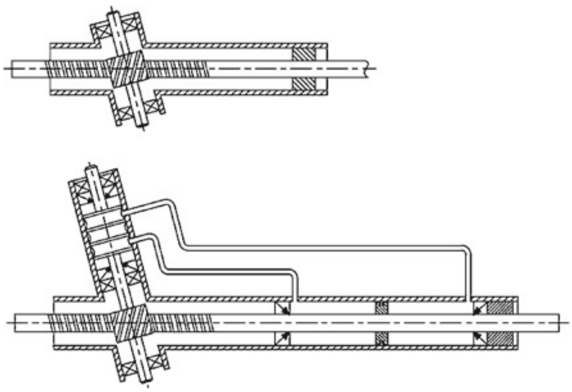


Fig. 11.35 Comparing packages of manual and hydraulic gears

geometry. Even more significant is a change of the driver's behaviour who steers much more often when the vehicle is at rest, i.e. when the rack forces are high, because any power steering will facilitate this without trouble. Yet some components of the gear, like dovetailing or pinion bearing, are less stressed by driving, because they are loaded only with lower torques applied by the driver. Hence, the design of the strength of these components is focussed on a few or a single peak load which may appear during abuse situations like a kerb impact or accidents in which even the highest possible power-assistance is insufficient and a major part of the rack force will transfer into the mechanical part of the steering.

11.8.4 Specific Features of Hydraulic Gears for On-board Use

No servicing is intended for the hydraulics of the steering system—apart from a regular check of the oil level within the scope of a general vehicle inspection, to avoid failure from minor leakage. This places high demands on the sealing system of the steering hydraulics.

Other than with a manual steering, the safety concept of the power-assisted steering system also has to consider a failure of the power-assistance. The formal requirements are met by a mechanical driving of the steering wheel to the front wheels and by a proof of low enough steering forces if the power-assist fails, compliant with EU regulation ECE R79. It is ensured as well that the driver can handle a vehicle with failed power-assist safely enough, even in real traffic situations, and that any failure is indicated early on.

11.9 Constructions and Components of Hydraulic Gears

The function of a power-assisted rack covers the following aspects:

- Solid or elastic attachment of the gear case to the vehicle (mainly at the subframe which takes up the front axle)
- Defined mounting points of the axial joints of both tie rods at the rack in a fixed distance
- Translation of the rack, parallel to the Y axis of the vehicle co-ordinate system (transverse axis)
- Defined gear ratio between rack translation and input shaft rotation
- Defined mounting point for the steering column at the input shaft of the gear, so that its position and the orientation of the input shaft allow the observation of the specifications for the attachment to the steering wheel (package, free running, uniformity of the gear ratio)

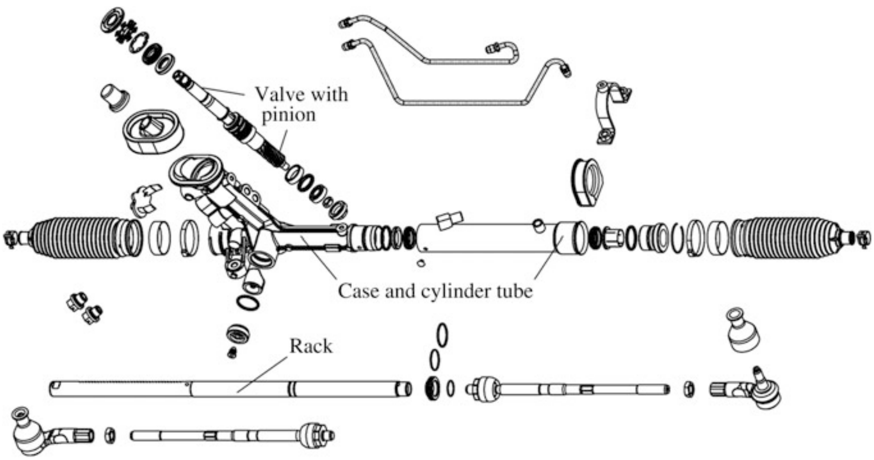


Fig. 11.36 Main components of the hydraulic rack-and-pinion steering

- Integration of the steering hydraulics which receive the wheel torque applied at the input shaft and its translation into an assisting force in the direction of the rack.

Figure 11.36 shows an exploded view of a typical gear, the main components are labelled.

Many components are similar to those of the mechanical gear: pinion, rack yoke, rack, bushing, case and gaiters. They were already discussed in Sect. 11.4. This chapter therefore only discusses the additional components of hydraulic power steerings, based on the general construction of a rack-and-pinion steering: steering cylinder, steering valve and connections.

11.9.1 Configurations

Various configurations were developed to adapt the gear to the package and front axle geometry of the car, differing chiefly by the arrangement of the steering cylinder.

11.9.1.1 End Tap

This configuration, shown in Fig. 11.37, is characterised by the continuous rack which is dovetailed on one half (mechanical part of the gear) and designed as a piston rod with piston on the other half (hydraulic part of the gear). The respective axial joints of the tie rods are joined at their ends. This is currently the most common version.

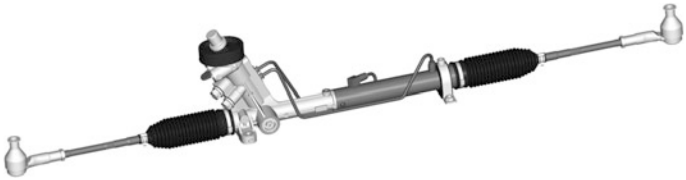


Fig. 11.37 Hydraulic rack-and-pinion steering with end tap

This configuration of the steering is compact and of low weight; it facilitates a stiff joining of the wheels, because the rack also joins both tie rods without further components and because the tie rods transfer the force into the gear without any offset into the rack axis. Left- and right-hand drive version can often be built by mirroring the steering at the middle of the rack, ultimately exchanging the mechanical and the hydraulic part. The rather short remaining length of the tie rods can be a shortcoming of this arrangement. It is defined by the distance of the steering arms at the wheels and by the total length of the rack. Big bending angles of the tie rod joints can result from deception and rebound of the front wheels, entailing a high lateral force as a part of the tie rod forces transferred to the rack; this has to be considered when designing their strength. On the other hand, large toe-in changes over the spring path of the front wheels can result, forcing compromises in the design of the chassis geometry.

However, a suitable design of the chassis can apply short tie rods favourably to reduce the steering ratio for high steer-angles even further.

11.9.1.2 Central Tap

This configuration, shown in Fig. 11.38, also consists of a mechanical and a hydraulic part which are connected by a continuous rack. The tie rods, however,

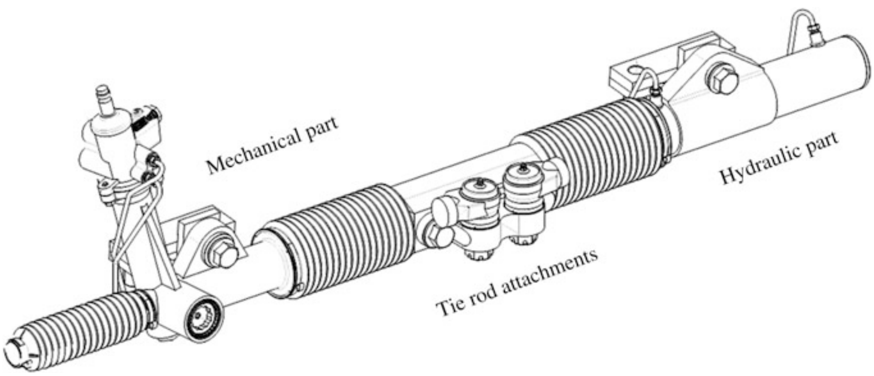


Fig. 11.38 Rack-and-pinion steering with central tap (tie rods not shown)

are mounted outside of the axis in the middle of the rack, between the mechanical and the hydraulic part.

The advantage of a possibly larger tie rod length is countered by higher construction expenditure and weight. In the 1980s and 1990s, when power steerings were offered as extra equipment for lower vehicle classes, this construction allowed installing a compact and light manual gear with one-sided tap (short gear) and a power-assisted gear with central tap in the same vehicle, without further adaptation of the mounting or the tie rods. This design is avoided in new constructions and was lately used only in some all-terrain vehicles, in which advantages of long tie rods for the chassis geometry are more apparent, on account of very long spring paths.

11.9.1.3 Parallel Auxiliary Cylinder

The parallel arrangement of the steering cylinder to the rack axis developed from the desire to use a power steering in a chassis that was originally made for a manual steering, without essential changes. A differential cylinder is connected to the steering case, so that the piston rod is parallel to the rack and coupled to its end by a connecting element. The cylinder is an independent unit, the steering case is modified chiefly to accept the steering valve. One shortcoming of this construction is the different effective piston surface in the two cylinder chambers, resulting from the fact that on one side of the piston, the piston rod covers a part of the piston's surface. If the same rack force is desired, different pressures have to be applied according to the steering direction, this needs to be observed during valve design. Because of the axial offset, the piston rod transfers bending torques into the rack, and the connecting element is charged with high loads, too. The weight is then higher than that of a gear with end tap, and efforts for assembly and costs are higher, too. The advantages are a possibly longer tie rod and a basically higher flexibility of the package, because the steering cylinder can be arranged in different positions inside the gear case. This configuration is not used any more in passenger cars if the whole platform is equipped with power steering.

Heavy-duty vehicles, though, feature prototypes of a similar configuration, shown in Fig. 11.39. They should be used if the still widespread rigid front axle

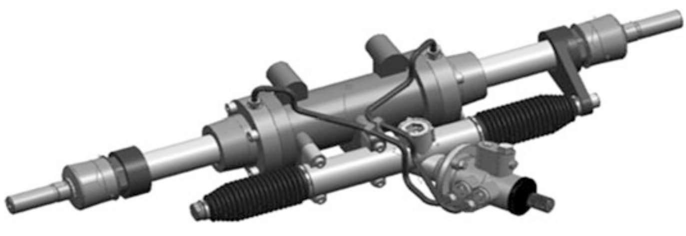


Fig. 11.39 Prototype of a parallel gearbox for heavy-duty vehicles

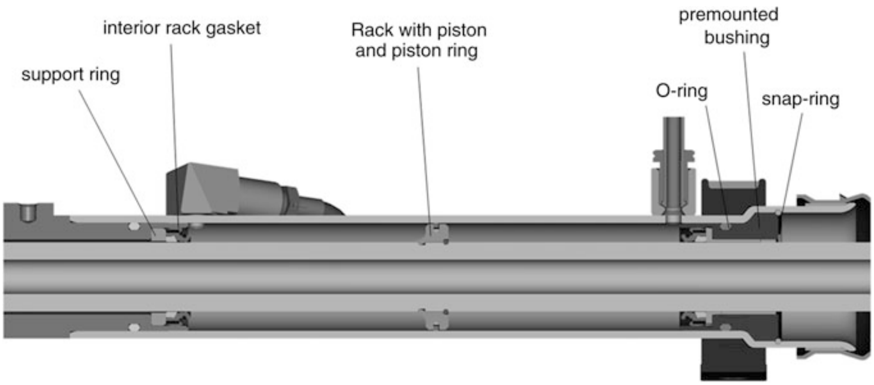


Fig. 11.40 Assembly pipe and rack

with recirculating ball-with-nut gear and steering linkage is substituted with an independent suspension with rack-and-pinion steering. However, the axial joints are then attached at both sides of the hydraulic cylinder, and the mechanical part of the steering including the rack is arranged in parallel.

This design enables quite long tie rods which are useful to achieve large turning angles of the front wheels (>55° for the inner wheel) while maintaining Ackermann's condition to a great extent. Both features are necessary to provide a small cornering circle even for a long wheelbase and to make is usable in practise. The reduction of the whole steering ratio can be limited until the full angle of the wheels is achieved. That is desirable for heavy-duty vehicles, so that enough power-assist is available over the whole steering path, while the gear is very small and light.

The configurations in the following chapters refer to steering systems in the preferred design, the end tap.

11.9.2 Cylinder

The cylinder of a power steering contains all the parts for supplying an assist power by hydraulic means. The parts are shown and labelled in Fig. 11.40.

In addition, the cylinder includes one or several outside mounting points for on-board mounting. In power steering, the bushing of the rack is placed into the cylinder area to establish a very wide distance between the two bearing points in the steering.

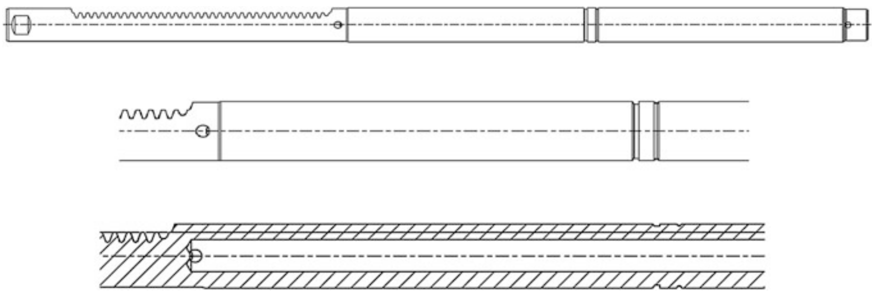


Fig. 11.41 Rack with concave drillings and piston groove

11.9.2.1 Rack in the Cylinder Area

The rack in the cylinder area has the following functions:

- Bearing the piston and the piston forces
- Sealing the cylinder space and the rack gaskets
- Bearing externally transferred longitudinal and lateral forces with low elasticity
- In most configurations: Air pressure compensation between both spaces, surrounded by the gaiters, when the rack is moving.

Racks are mostly made of corresponding round bar steel in one piece—two-part assemblies are seldom used for mechanical or hydraulic applications. To reduce weight and to permit the mentioned airflow, the rack is drilled hollow across the whole length or at least in the hydraulic area (see Fig. 11.41). When the mechanical section including the dovetailing is finished and a radial groove is introduced into the rack to fasten the piston, it is ground in the whole hydraulic area to achieve a sufficient surface finish for the rack seals, which may not be impaired by the further production process.

The same requirements apply to the layout of the rack strength as to mechanical steering systems. They result esp. from the fact that the rack is a critical part in safety issues that has to maintain a solid connection between the guided wheels and a controlled movement along their longitudinal axis under any circumstances. The area critical for the strength is generally near the dovetailing, towards the middle of the rack, hence, no particular requirements need to be observed for the hydraulic section of the rack. Yet, overall, the rack has to be laid out with more strength than a manual steering would provide. One reason is the wider length of the rack which allows for higher bending torques when lateral forces are introduced at their ends. Another is the higher potential rack forces, particularly in abuse situations like kerb impacts. They are caused by a combination of hydraulic peak support and extremely high steering wheel torques (they can amount to more than 100 Nm). Even in this case, the rack may not suffer plastic deformation that would affect its easy motion in the gear.

11.9.2.2 Pistons with Piston Ring

The piston is made from a rotary steel part, the piston ring consists of a PTFE compound (filled with bronze or graphite), produced by sintering or extrusion, then it is twisted and cut to length.

Together, they have the following functions:

- Hydraulic separation of the two cylinder spaces with least internal leakage
- Transmission of the hydraulic forces to the rack.

Figure 11.42 shows piston and piston ring in the mounted state, joined to the rack.

The sealing of the cylinder chambers is a primary task of the piston ring. The gap (approx. 0.1 mm) between the outer diameter of the piston and the inner diameter of the cylinder pipe is closed by the piston ring. It is located in a groove of the piston whose depth allows it to perform compensating movements. Since the outer diameter of the piston ring is wider than the cylinder pipe and there is the biasing force of the O-ring under the piston ring, a static contact pressure is generated to prevent the fluid from flowing over the piston ring when the pressure rises quickly (blow-by effect).

The actual sealing effect, however, results from a contact pressure, applied by the fluid pressure. The piston ring is pressed against the corresponding groove flank, and the O-ring exerts a radial force on the inner diameter of the piston ring. A required prerequisite for meeting this task is constant free access of the fluid to the O-ring. During rapid mutual pressurisation it may happen that the piston ring will touch the turning point of the groove flank and prevent direct access of the pressure medium to the O-ring. Then the pressing force for sealing is not produced, which can cause leaking. Therefore, grooves are made at both sides of the piston ring, to keep the hydraulic connection to the O-ring upright (Fig. 11.43).

The gap between piston ring and piston groove should allow the ring to move. If the ring is too wide, however, it will move during a change of the pressurised side (e.g., by reversing the steering direction) from one contact surface of the groove to the other. This first delays the pressure buildup in the cylinder chamber, then releases it with a steep gradient when the piston ring has achieved its final position and the sealing begins. This causes sound impulses in the hydraulic system, these are clearly audible in the vehicle and are unacceptable.

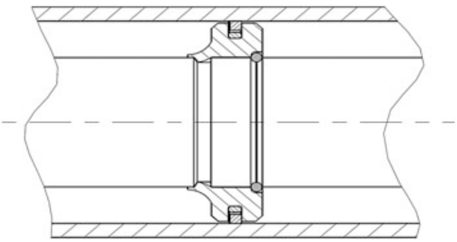


Fig. 11.42 Assembly rack with piston and piston ring

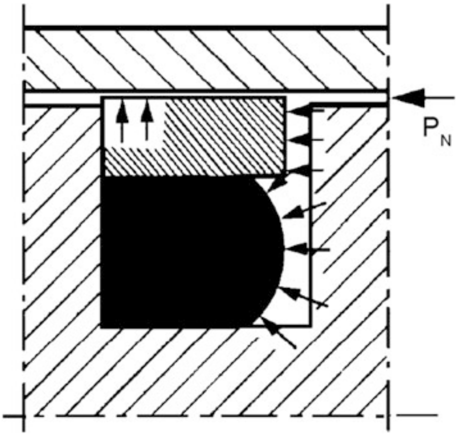


Fig. 11.43 Effect of the fluid pressure on piston ring and O-ring

The gap between the outer diameter of the piston and the inside diameter of the cylinder pipe prevents the piston from touching the cylinder wall, even if the rack bends under lateral force. But under lasting influence of high pressure and high temperature, the material of the piston ring is able to creep through a too wide gap (oil temperatures up to 120, 140 °C are possible in individual applications), so that the piston ring extrudes into the gap, loosing its sealing capabilities.

There are different ways to attach the piston to the rack. They have in common that material from the inner diameter of the piston is transformed by pressing into a radial groove on the rack. The piston disposes of a collar with low wall thickness at its inner diameter, either on one side or on both sides. This collar is radially deformed by pressing with a corresponding matrix, by rolling out or by swaging hammers into proper grooves on the rack, establishing a positive connection. If the piston has a collar only on one side, then the other side will be supported by a retaining ring which lies in a rack groove.

Basically, the piston has to be able to transfer the hydraulic peak forces into both directions without moving relative to the rack. Otherwise its motion would generate sound impulses in the hydraulic system, similar to those described for the piston ring.

11.9.2.3 Rack Gaskets and Sealing System of the Cylinder

The main function of the rack gaskets is to seal the hydraulic area of the cylinder pipe reliably, under any possible condition, to prevent leakage for the whole service life. The gaskets should further ensure that no air is sucked from the space within the bellows into the cylinder. This can happen during extreme steering movements at high rack speed, when low pressure is generated in a cylinder chamber, because the volumetric flow extracted by the pump is below the

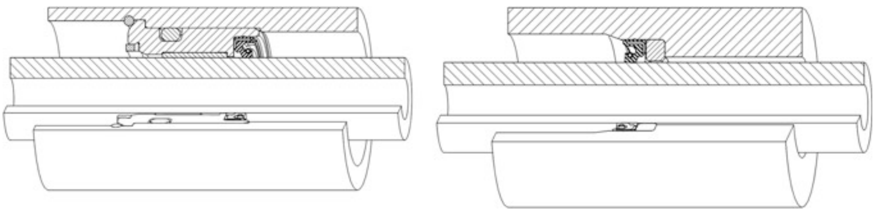


Fig. 11.44 External and internal installed rack seal

changing volume of the chamber. Another common and, for this effect, critical situation occurs when the vehicle is mounted and the front wheels are swivelled while the engine is switched off.

Figure 11.44 shows installed rack seals. The left figure displays a seal pressed into the rack jack, as it is inserted at the end of the hydraulic cylinder (external rack seal), the right figure shows a seal pressed into the steering case on the border between the mechanical and the hydraulic part of the gear (internal rack seal).

Configuration of a rack seal

The rack seal consists of a metal body which establishes the strength to resist pressures of 120 bar and more. This body is moulded with a rubber mixture in such a way that the sealing elements in the outer and inner diameter receive their shape.

The used rubber material has to establish the compatibility to the used hydraulic oil, it needs to be sufficiently strong at high temperatures and sufficiently flexible at low temperatures. At the mating face to the rack high abrasion resistance and low friction is also required, this avoids stick-slip effects and any resulting audible seal squeaking.

Interface at the external diameter

The seal at the external diameter is pressed into the case or jack. To establish sealing at this point, the press fit has to be configured for sufficient radial pre-tension of the mating face, the seal seat has to be machined with narrow tolerance margins and it needs a defined surface structure. A too rough surface could cause the outer seat of the seal to wear or shear off upon assembly. A too smooth surface could mean that in spite of correct size, the press fit of the seal could fail and be pulled out of its seat when low pressure is applied in the corresponding cylinder chamber. This occurs when the steering system is filled and in some operation modes. A large enough wedge-face supports the seal if the cylinder chamber is pressurised. A supporting plastic ring integrated into the seal often serves this purpose.

Interface in the inner diameter

At this interface relative movements between sealing lip and rack may occur. It is laid out so that it leaves only a thin oil film for lubrication behind when the rack moves out, and this oil film can be fully retaken when the rack moves in. The radial pretension is established by combining three effects, see Fig. 11.45.

First, some 50 % of the whole radial force is issued by the spiral spring integrated into the seal, when it is widened to the rack diameter during assembly.

Second, the perimeter of the sealing lip is widened during assembly, too, contributing some 40 % of the whole radial force as an elastomer tension.

Third, there is an elastomer bend section (approx. 10 % of the whole radial force, all data given for ambient temperature) which results from the bend (bend angle 7–12°) of the diaphragm surface when it is pushed onto the rack.

In addition to the static radial force, the pressure admission flow generates a radial force part on the sealing lip which rises proportionally to the pressure, because the sealing lip is not vertically oriented to the rack, but turned at an angle into the cylinder space. The diaphragm surface is distorted under rising pressure, so that in addition to the primary sealing lip, acting in the static state (0 bar pressure above atmospheric), the sealing lips on the base or the diaphragm surface on the rack are touching and supporting the sealing (see Fig. 11.46).

One particular difficulty in the layout of a seal is the required high flexibility of the mating face during radial movements of the rack. The bearing plan of a rack-and-pinion steering (guide about rack yoke and steering pinion) is the reason that the rack moves radially under lateral force, relative to the seal, and it rolls, for example, due to radial dovetailing forces between rack and steering pinion. The sealing lip has to follow these movements and make sure that a sufficient radial pretension is maintained around the whole perimeter of the rack—otherwise oil will leak out in such situations. This is most difficult in extreme cold start situations, because then the elastic mixture loses its flexibility. Further, one needs to

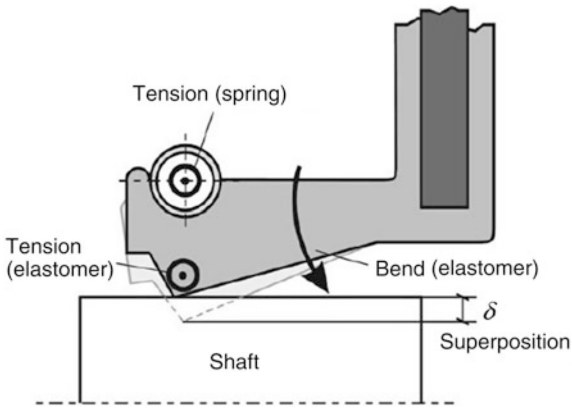


Fig. 11.45 Static preliminary tension of the rack seal on the rack

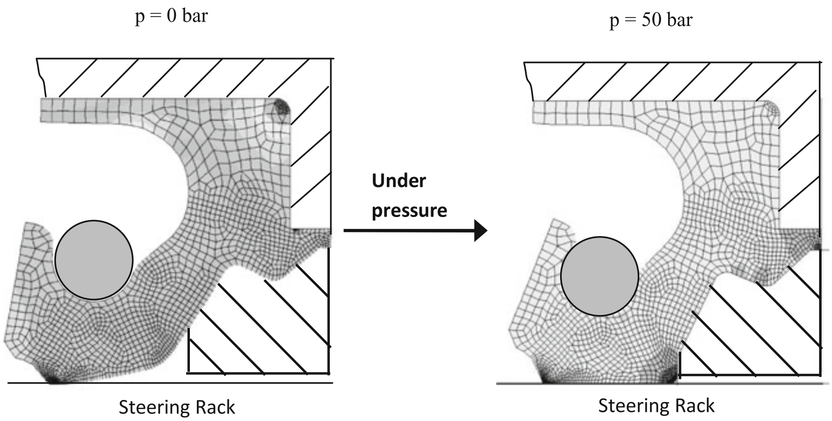


Fig. 11.46 Function of the rack seals

ensure that these movements entail no overstretching of the elastic body, because it will otherwise lose its mechanical strength and fail early under pressure (penetration wound).

Integration of the seal in the gear

The internal rack seal which separates the hydraulic from the mechanical section is used directly at an accordingly shaped place in the steering case.

The external rack seal is implemented into the rack jack. Then the outer diameter of the rack jack is sealed with an O-ring against the cylinder pipe.

In general, the maximum working pressure of a rack-and-pinion steering is limited by the rack seals. A very high pressure is aimed at, because then the required rack force can be generated with a small piston surface, allowing for the layout of a low volumetric pump flow, which is favourable with regard to energy consumption and thermal load of the steering system, beside advantages for package and weight. Above a limit of approx. 140–150 bar, however, the risk of steering failure due to leakage rises for conventional seals within the intended service life.

11.9.2.4 Cylinder Pipe with Connections

The cylinder pipe has many tasks:

- Surround the cylinder space with sufficient pressure resistance
- Supply hydraulic connections to link the cylinder chambers and the transfer lines to the steering valve
- Bear and seal the rack jack

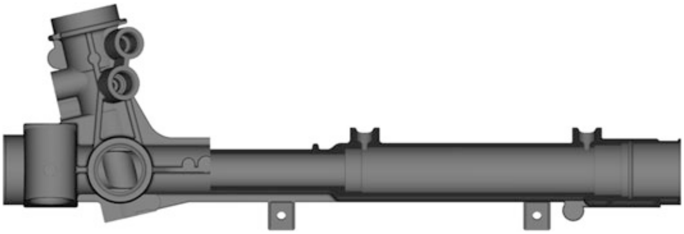


Fig. 11.47 Gear case as an integral case

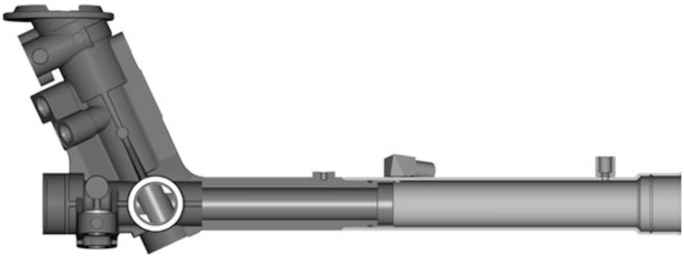


Fig. 11.48 Gear case with pipe/case connection

- Supply mounting points for the gear
- Bear other individual mounting points, for example, for a heat shield
- Join the seal bellows.

Two configurations are common, shown in Figs. 11.47 and 11.48.

First is the cylinder pipe, an integral part of the steering case, made of die-cast aluminium (integral case). The mounting points of the gear, hydraulic connections and other mounting points can be integrated into the mould. Like the valve area, the cylinder area has to be fully machined.

Second is a separate pipe, most commonly a steel pipe, pressed onto a short gear case containing input shaft and valve. The inside of the pipe receives its inside diameter and the required surface finish without further treatment by a process called calibrating, it consists of pushing a steel sphere with defined diameter through the pipe. For the hydraulic connections, however, the cylinder pipe has to be drilled first, and then corresponding connecting pieces have to be welded to the pipe. Mounting elements are welded at the pipe, too, bearing the risk of welding distortion of the pipe, which would not be acceptable in the cylinder area. In general, the combination of gear case and pressed steel pipe is cheaper than the integral case. The latter also puts high demands to the available die-casting technology, limiting the scope of potential manufacturers. However, depending on the number and complexity of the parts to be welded at the steel pipe, the integral case can also be cheaper, in addition to the lower weight of this configuration.

11.9.3 Rotary Disk Valve with Input Shaft and Steering Pinion

The steering valve is, from the technological point of view, the core of the hydraulic power steering. Today it is always a rotary disk valve and has the following tasks:

- Maintain a mechanical through-put from the input shaft to the steering pinion
- Connect both cylinder chambers with the inlet and runback of the hydraulic supply
- Record the wheel torque introduced by the driver and identify the course
- Adjust the hydraulic pressure depending on the acting wheel torque and direct the pressure into the cylinder chamber corresponding to the course.

The steering valve is connected to the driver by the steering column and the steering wheel, and esp. the level of the current power assist, i.e. the pressure directed from the steering valve into a cylinder chamber is clearly felt. Therefore, the steering valve has to build up the driver's support in a predictable, repeatable and steady way, with low hysteresis and without lag. This means that the requirements for precision of the steering valve parts are high.

The parts of the gear are shown and labelled in Figs. 11.49 and 11.50, both assembled and in an exploded view.

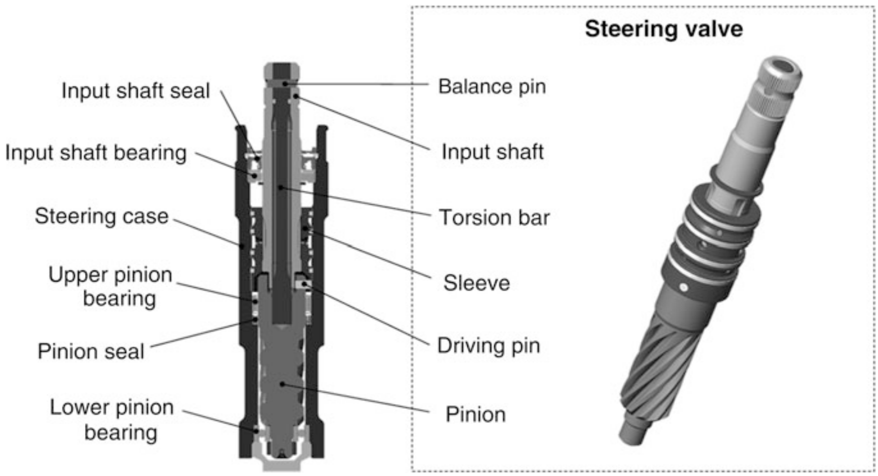


Fig. 11.49 Steering valve ZSB with naming of the parts

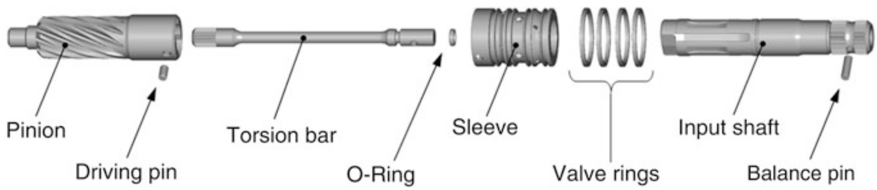


Fig. 11.50 Parts of a steering valve in explosion representation

11.9.3.1 Steering Pinion

The steering pinion is very similar to the version in mechanical gearboxes. Its support however, is right above and below the dovetailing area in the case, so that reaction forces are not transmitted from the dovetailing into the steering valve. The mechanical coupling of the input shaft is maintained both by a torsion bar whose end is pressed into the body of the steering pinion, and by end stop surfaces for the input shaft, that limit the peak torsion of the torsion bar, which represents a mechanical overload fuse.

Furthermore, the steering pinion disposes of a mating face for a shaft-lip type seal, to seal the valve chamber arranged right above or below the upper steering pinion support, so that the support is either inside the hydraulic chamber and runs wet or is outside and runs dry.

11.9.3.2 Torsion Bar and Overload Fuse

The torsion bar connects input shaft and steering pinion axially, so that they have a defined position to each other. It also causes a relative twist between input shaft and steering pinion as a function of the steering wheel torque, which is used to control the power assist and its direction. Its layout has to maintain an exactly defined torsion spring stiffness which is primarily determined by diameter and length of the choke. It has an influence on the required steering wheel torques (more stiffness = more steering forces) but also a profound impact on the steering feel and the feedback of the steering—these facts are discussed in more detail in Sect. 11.10. Within the scope of the greatest possible angle difference between input shaft and steering pinion, defined by their respective stop surfaces, the torsion bar has to be permanently rigid. Endangered cross sections are near the choke and near the O ring groove.

The torsion bar is pressed into a drilling at one end of the steering pinion. At the other end it is connected with the input shaft usually by piercing both crosswise and securing them with a pin. The valve is mounted into a hydraulic balancing stand which models the surroundings of the gear near the steering valve. The steering pinion is fixed onto the mounted sleeve, and the operation of the valve is simulated by moving the input shaft. Then the valve characteristics for a clockwise and a counterclockwise torsion are recorded. Finally, the input shaft is moved into a position in which the valve indicator is symmetrical for both rotational directions: the hydraulic centre. That is, a twist out of this position around a certain angle, clockwise or counterclockwise, will produce the same pressure. In this position, the input shaft is connected with the unloaded and tension-free torsion bar, e.g., by drilling and pinning. By this process, called the balancing of the valve, it is ensured that pressure is not driven into one cylinder chamber without wheel torque and that the pressure builds up, according to the wheel torque, symmetrically into either rotational direction.

11.9.3.3 Sleeve with Valve Rings

The sleeve is one of two parts representing the function of the steering valve (see Fig. 11.51). It is solidly connected with the steering pinion in the rotational direction, however, it can follow an angular or axial offset between the surrounding input shaft and the steering pinion within certain limits. On the outside, the sleeve has three radial grooves that create three chambers, separated by the valve rings, once they are installed into the case. The middle groove is connected with the inlet of the hydraulic supply by drillings in the case, the upper and lower groove are connected with one of the cylinder chambers.

The valve rings are almost identical with the already described piston rings in terms of function and resulting requirements, material and configuration.

Inside, the sleeve is cylindrical. Axial grooves are implemented at a certain distance from the end of the sleeve which have sharp, right-angled edges at the rim. They adopt the valve function in their interaction with the grooves and control edges of the input shaft (see Sect. 11.9.3.4). The remaining cylindrical sections of the inner diameter without grooves, and the corresponding surfaces of the input

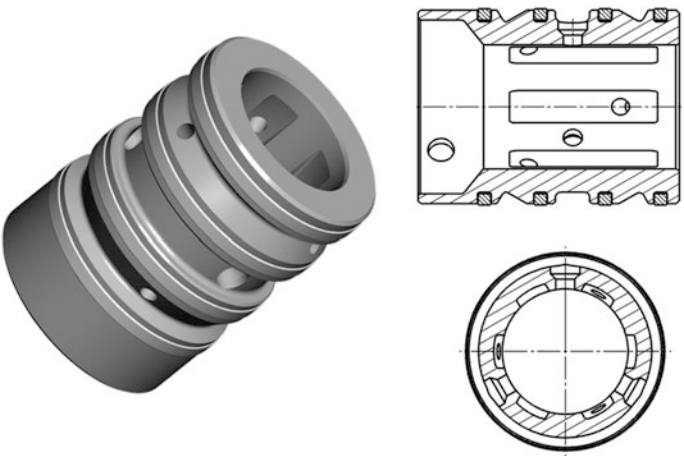


Fig. 11.51 Sleeve with valve rings

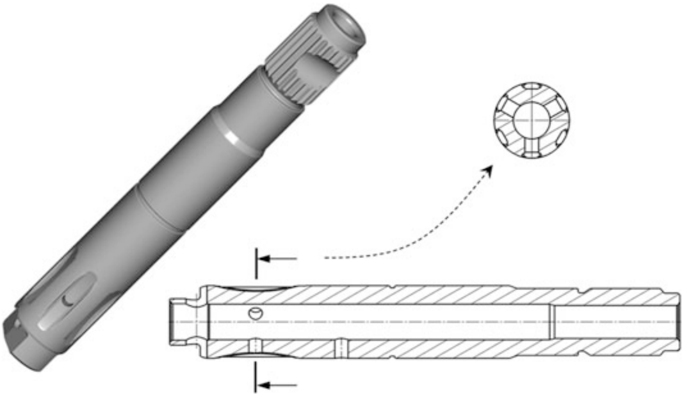


Fig. 11.52 Input shaft

shaft sealing this valve area by a gap, are as narrow as possible. Hence, high demands are made to the treatment of the cylindrical inner diameter, both with respect to the sizes with very low tolerance margins and with respect to the surface finish.

11.9.3.4 Input Shaft

The input shaft is the second part of the steering valve. It has the purely mechanical function of transferring the wheel torques, either from the pinning into the torsion bar or from the end stop surfaces into the steering pinion, it is shown in Fig. 11.52.

The inside of the input shaft is drilled hollow. The hollow cavity of the version shown here is connected by lateral drillings to runback grooves and the valve chamber beyond the sleeve, whose section in turn is connected by a case drilling to the runback of the hydraulic supply. On the outside, the hollow cavity is sealed by an O ring between input shaft and torsion bar.

In the actual valve section, the input shaft is the counterpart of the sleeve with a high-precision cylindrical outer diameter and axial grooves. Their edges, called control edges, are provided with an exact structure, either by sanding several facets or by stamping a certain form of the control edges. The curve of the power assist over the wheel torque is determined by this structure, as described in Sect. 11.10.1. The contour near the sleeve and the sleeve itself seal the high-pressure area by narrow gaps, as described above. At its upper end, the input shaft is supported to take up the lateral forces initiated by the steering column. A radial oil seal seals it against its surroundings (Fig. 11.53).

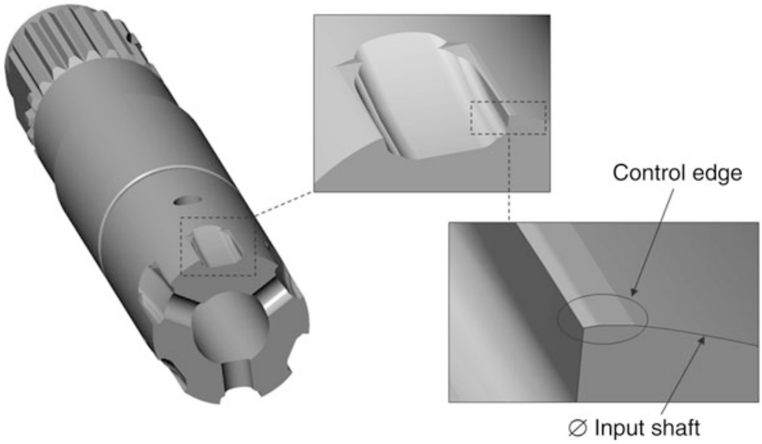


Fig. 11.53 Cut view of an input shaft with control edge

11.9.3.5 Configuration Forms

Many versions of the described valve configuration were developed to solve special problems, mostly for packaging. They are not widespread, because these versions cause add-on costs or negative concomitants like more friction, and they are not required if the car is properly designed. Alternative valve designs (star valve, coil valve) have been fully displaced by the rotary disk valve as well.

However, to kinds of valve housings are still distinguished, see Fig. 11.54.

The easier form, the cartridge valve, has a valve housing which is a part of the complete steering case. In this version, the lower pinion bearing is fixed. The outer

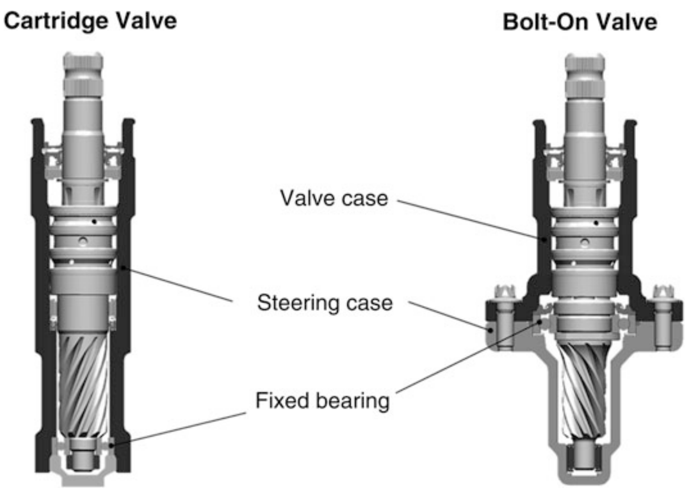


Fig. 11.54 Cartridge valve (on the *left*) and bolt-on valve (on the *right*)

diameter of the pinion is limited by the inner diameter of the pinion seal which in turn is given by the outer diameter of the sleeve and the space required for the seal. The widest diameter of the pinion determines the biggest possible steering ratio (rack stroke per pinion rotation).

The alternative form, the bolt-on valve, is completely preassembled in its own valve housing and screwed on the steering case. The fixed bearing is the upper pinion bearing, attached between valve and steering case. The size of the pinion is not restricted by the sleeve in this configuration. A higher assembly effort and a greater number of individual parts are the setbacks.

11.9.4 Other Parts of a Hydraulic Gear

Other parts which have to be added to a hydraulic gear shall be mentioned briefly, for the sake of completeness.

11.9.4.1 Transfer Pipelines

The task of the transfer pipelines is to hydraulically connect the valve section with the cylinder section individually for each cylinder chamber. They are usually made of low-diameter steel pipe and provided with different kinds of hydraulic connections at their ends, depending on the requirements of space and assembly.

The cross sections have to be selected in such a way that, despite quick steering movements and accordingly high volumetric flows in the transfer pipelines, there will be no high resistance to flow, esp. not when the fluid is cold and more viscous.

11.9.4.2 Bellow Vent Pipe

In contrast to the mechanical gears which admit a free air exchange between the compressed and the expanded bellow when the rack moves, the sealed cylinder area of the hydraulic steering has to be circumvented. Solutions establishing the exchange by an external connection of the bellows by means of a pipe have almost fallen out of use. The costs for the bellows rise, because of the required pipe connection (the bellow is no longer axially symmetric), the external pipe has shortcomings in the package and the air exchange system is prone to leakage during operation, potentially entailing long-term corrosion on the rack and subsequent oil leakage of the gear.

More often, the hollow drilling of the rack is used to enable the aerial exchange between both sides of the cylinder. This is made possible by a lateral drilling before the dovetailing and either another lateral drilling at the end of the rack or venting grooves introduced into the axial joint.

Table 11.5 Some characteristics of hydraulic rack gears

Vehicle class	Compact cars	Compact category	Middle class	Van/SUV	Light utility vehicle
Front axle load (kg)	950	1,150	1,500	1,800	2,100
Maximum rack force (N)	7,000	8,500	11,000	12,500	15,000
Rack diameter (mm)	24	26	28	30	32
Piston diameter (mm)	40	42	46	48	52
Maximum working pressure (bar)	90–120				
Volumetric flow (l/min)	6	7.5	9	10.5	12
Rack stroke (mm)	±65 to ±85				
Gear ratio	40–60 mm/rotations (sports car also 75 mm/rotations)				
Temperature	−40 to +120 ˚C (partially also +140 ˚C)				

11.9.5 Typical Characteristics of Hydraulic Gears

The layout of rack gears for passenger cars is always specific to the respective car platform. Specific features of the drive concept, the layout of the package or the definition of the chassis affect the requirements for the respective gear significantly. Therefore, there are major differences between the gears even for vehicles in the same class. This also concerns their characteristics.

Table 11.5 gives an overview of the representative characteristics of gears in the different vehicle classes.

11.10 Functionality of the Steering Hydraulics

11.10.1 Steering Valve: Principle of the Cutback

According to the terminology for hydraulic parts, the steering valve is a mechanically operated 4/3-way proportional valve with an open centre. This means that a continuous volumetric flow is transported by the hydraulic supply through the valve. The resistance to flow is least in neutral position. With increasing deviation, the resistance to flow of the valve increases steadily, and the flow pressure rises. At the same time, depending on the rotational direction, one chamber of the steering cylinder is connected to the inlet and the high-pressure levels there, the other one is connected with the runback and the low-pressure level prevalent there. The active principle of the steering valve is the specific cutback of a continuous volumetric flow.

As shown in Fig. 11.55, the configuration of the steering valve corresponds to that of a Wheatstone bridge. A pressure difference is generated by changing the resistances to flow of the bridges B1 and B4 or B2 and B3 in pairs. They are

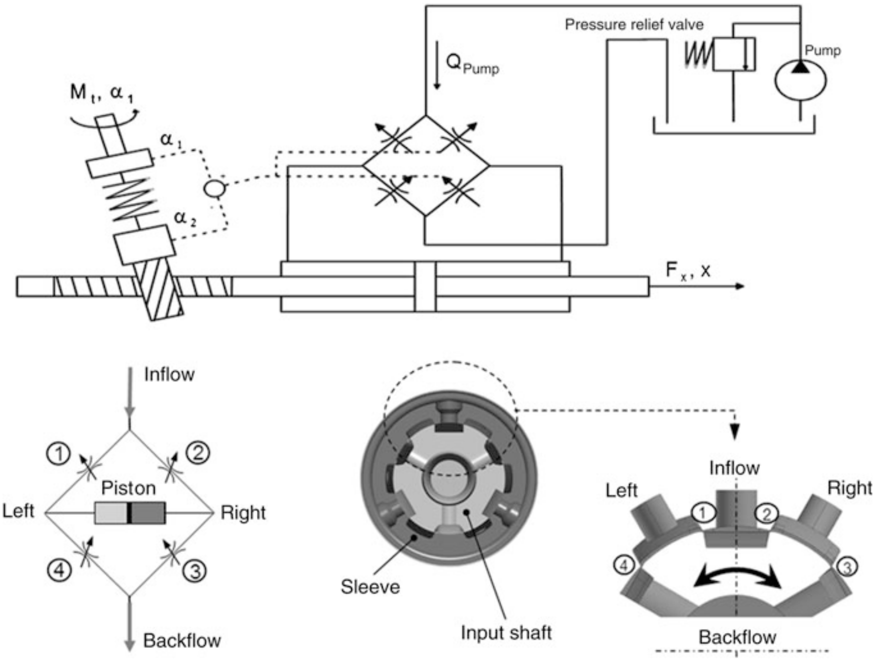


Fig. 11.55 Diagram of the steering hydraulics

constantly passed by the hydraulics fluid and directed into both chambers of the steering cylinder. In real valves, there are several Wheatstone bridges in parallel, three of them are shown in the example. Only one bridge, i.e. a 120° segment of the valve cross section, is examined to understand the active mechanism in the valve.

The flow path of the hydraulic liquid in the valve is as follows: The fluid streams into the central ring channel on the outside of the sleeve. Drillings pass it on to the space between sleeve and input shaft, more precisely, to a place where the input shaft has an axial groove on its outer diameter while the sleeve has none on the inner diameter.

The fluid continues into a space where the sleeve has an axial groove on its inner diameter but the input shaft has none on the outer diameter (note: In the image below on the right, the hydraulic spaces are dark while sleeve and input shaft are bright). A sufficiently wide gap remains in the neutral position of the valve between both areas (bridge B1 and B3), because of the negative overlap of the valve, so that the fluid can flow over with little resistance. The volumetric flows splits symmetrically between both possible directions of flow. Each of the grooves in the inner diameter of the sleeve is again connected by drillings to one of the ring channels in the outer diameter of the sleeve: the groove that is left in the image is connected to the upper ring channel and the right groove to the lower ring channel. The two ring channels are connected via the transfer pipelines with one cylinder chamber each: the upper ring channel with the cylinder chamber which supports a steering movement to the left and the lower ring channel with the opposite chamber.

Next, the hydraulic liquid flows again into an area with a groove in the input shaft and without groove in the sleeve (bridge B2 and B4), again, there is little resistance in the neutral position because of the negative overlap.

In the form shown here, the grooves in the input shaft are connected by drillings to their hollow internal parts, so that the fluid is conducted to there. It flows axially to the input shaft into an area which is not covered by the sleeve, leaves by lateral drillings in the input shaft and returns by a drilling in the case and a connected runback into the tank. The state of the not actuated valve (central position) is shown on top in Fig. 11.56.

If the input shaft is torsioned relative to the sleeve, as shown in the figure, the cross sections change where the fluid is streaming from a groove in the input shaft to a groove in the sleeve, or vice versa. For example, if there is a counterclockwise torsion of the input shaft, the cross section of the bridge B1 grows, so that the fluids passes easier into the groove of the sleeve which is connected to the cylinder chamber supporting a steering movement to the left. The cross sections of the bridges B2 and B4 shrink, so that the respective volumetric flow passing over both bridges is hemmed in the same measure (symmetrical flow distribution). There is a corresponding back pressure in the space limited by these bridges, it is equal to the cylinder chamber pressure *pA*. The bridge B3 then receives a wider cross section, so that the volumetric flow across this bridge can freely pass away into the tank. In this area, the pressure level is low and corresponds approximately to the tank pressure, and thus also to the pressure in the cylinder chamber B (*pB*).

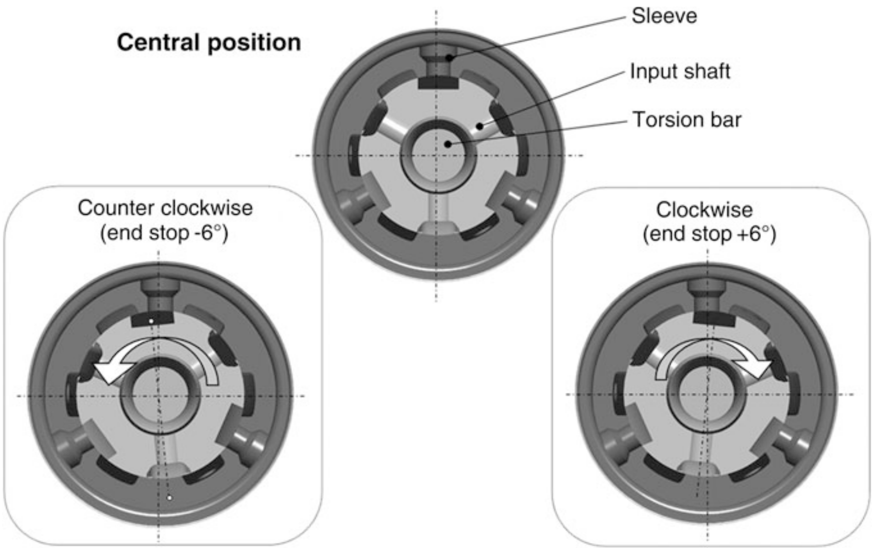


Fig. 11.56 Cutaway view of the valve in the actuated state

The form of the control edges in the input shaft essentially determines the runoff cross section of the respective bridges as a function of the relative position of input shaft and sleeve, describing the valve curve.

The orifice formula serves well to compute the approximate pressure drop as a function of the runoff cross section:

$$Q_1 = B_1 \cdot \sqrt{pP - pA} \tag{11.4}$$

$$Q_2 = B_2 \cdot \sqrt{pA - pR} \text{ mit } pR \approx 0 \tag{11.5}$$

$$Q_3 = B_3 \cdot \sqrt{pP - pB} \tag{11.6}$$

$$Q_4 = B_4 \cdot \sqrt{pB - pR} \text{ mit } pR \approx 0 \tag{11.7}$$

$$B_i = CD_i \cdot A_i \cdot \sqrt{\frac{2}{\rho}} \tag{11.8}$$

Symmetry of the control edges:

$$B_2 = B_3 \tag{11.9}$$

and

$$B_1 = B_4 \tag{11.10}$$

Distribution of the volumetric flows:

$$Q_P = Q_1 + Q_3 = Q_2 + Q_4 = Q_R \tag{11.11}$$

The Eqs. 11.9 and 11.10 result in:

$$Q_1 = Q_3 \tag{11.12}$$

and

$$Q_2 = Q_4 \tag{11.13}$$

$$Q_2 = Q_1 - QA(\dot{x}r) \tag{11.14}$$

With static rack:

$$\dot{x}_r = 0 : Q_2 = Q_1 \tag{11.15}$$

The Eqs. 11.11, 11.12, 11.13 and 11.15 yield:

$$Q_1 = Q_2 = Q_3 = Q_4 = \frac{1}{2} QP \tag{11.16}$$

Effective cylinder pressure:

$$\Delta p = pA - pB \tag{11.17}$$

From Eq. 11.4

$$pP = \frac{Q_1^2}{B_1^2} + pA \tag{11.18}$$

From Eq. 11.6:

$$pP = \frac{Q_3^2}{B_3^2} + pB \tag{11.19}$$

With Eq. 11.17:

$$\frac{Q_1^2}{B_1^2} + \Delta P + pB = \frac{Q_3^2}{B_3^2} + pB \tag{11.20}$$

Equations 11.9 and 11.16 yield for a static rack:

$$\Delta p = \frac{1}{4} Q_P^2 \cdot \left[\frac{1}{B_2^2} - \frac{1}{B_1^2} \right] \tag{11.21}$$

Substituting Eq. 11.8 yields:

$$\Delta p = \frac{\rho}{8 \cdot CD} Q_P^2 \cdot \left[\frac{1}{A_2^2} - \frac{1}{A_1^2} \right] \tag{11.22}$$

The valve curve is given as the pressure difference over the wheel torque, see Fig. 11.57. The relationship between wheel torque and relative torsion angle of input shaft and sleeve is given by the stiffness of the torsion bar. The outmost line which is recorded for increasing actuation of the valve, i.e. for rising pressure

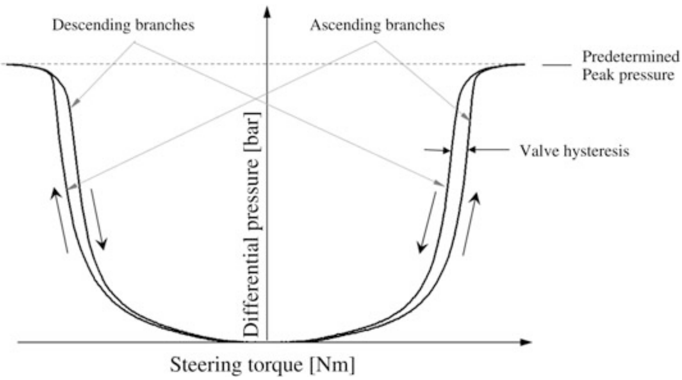


Fig. 11.57 Example of a valve curve

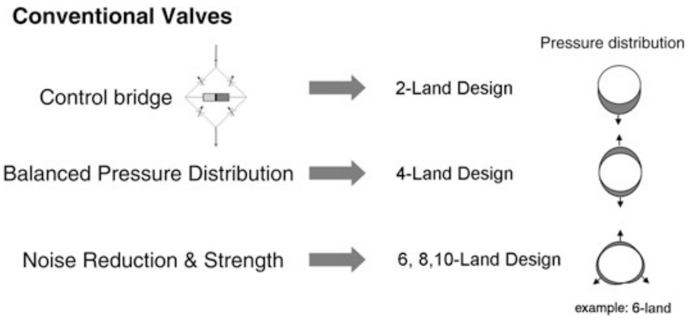


Fig. 11.58 Number of the bridges and distribution of the pressure zones in the sleeve

(ascending branch) is authoritative for the examination of a measured valve curve. Friction in the input shaft moving relative to the sleeve and to the case produces lower wheel torques on the descending branch at the respective pressure. The difference between the ascending and the descending branch is the valve hysteresis.

Real valves have three or four bridges arranged in parallel, so that the volumetric flow splits evenly between them. On the one hand, this requires more precision in manufacturing the valve parts because the cross sections at each bridge have to be small to achieve the desired total cross section, and all bridges have to be precisely synchronised. On the other hand, there are two advantages:

- Segments with high flow pressure and low return pressure are alternating in the inside of the sleeve. As mentioned before, the gap should be narrowest between inner diameter of the sleeve and outer diameter of the input shaft to provide for least leakage, however, nothing may stick. The maximum deformation of the sleeve is less if it occurs in many, but small segments all around the valve, rather than in a few large segments (see Fig. 11.58). A higher number of parallel bridges allows designing a lower gap width, in particular for high-pressure valves.
- A higher number of control edges enlarges the hydraulically moistened perimeter. This enlargement improves the pressure energy drop. A lesser chance for cavitation is the result, valve hisses are suppressed.

The accuracy by which a desired valve curve may be achieved is strongly dependent on precise treatment of input shaft and sleeve.

11.10.2 External Influence on the Valve Indicator

The equations derived from the orifice formula to define the pressure difference in the piston over the actuation angle of the valve show the following dependence (assuming a constant flow coefficient):

- The pressure rises quadratically with the volumetric flow.
- The pressure rises quadratically with the shrinking cross section of the gap at the control edges.
- The pressure rises linearly with the density of the fluid.

De facto, the friction of the fluid in narrow cross sections has an influence as well, but it is not significant for driving and shall not be discussed here. One effect occurs at high viscosity of the fluid that increases the pressure difference in the piston with rising viscosity. In reality, this high viscosity is only present after a cold start, in particular when mineral oil is used as a hydraulic fluid. Their influence is well perceptible in the car (lower wheel torques), but short-lived, on account of the quick heating up of hydraulic power steering.

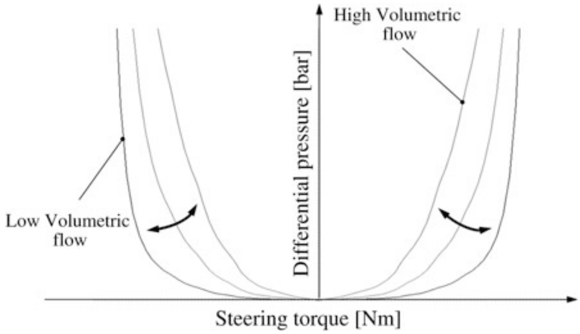
However, the density of the fluid varies only slightly with temperature and between different fluids during operation, so that there is no significant effect.

The valve curve is designed so that for a volumetric flow, which is assumed to be constant and transported by the pump, a given valve curve (pressure difference in the piston over wheel torque) is generated by adjusting the gap at the control edges as a function of the position of the valve, that belongs to the respective wheel torque. According to the treatment process (sanding or stamping the control edges of the input shaft), the accessible structures are limited. A change in size of 1 μm would already alter the valve curve at the control edges measurably, therefore, the fine tuning of the control edges is carried out after a preliminary tuning in a simulation, applying an iterative process of making and testing.

Observing the given tolerance margins for the valve curve in serial production is not accomplished by monitoring the shape of the parts and by controlling it in the manufacturing process—already the required measurement of a corresponding number of parts is not feasible with the needed accuracy. At least, the valve characteristic of a mating of sleeve and input shaft should be measured during balancement and checked for obervance of the tolerance margins before they are attached to each other. For some scattering widths, a hydraulic measurement of the parts against a reference part can make sense, so that input shafts or sleeves or both are classified, and then favourable matings are selected for balancement and mounting.

Installation in the gear changes the valve curve against its state during balancement. This is partially due to more friction at the input shaft (seal and support), partially also to minor tensions esp. between input shaft and sleeve that result from tolerance margins of the parts and the steering case. These tensions entail more friction and slightly change the relative position of the control edges of the input shaft to the grooves of the sleeve. As already mentioned, variations of 1 μm have already a significant influence on the valve curve there, so that changes can result which, if they are systematic, may be corrected by corresponding corrections of the control edges and the balancement of the valve. However, a stiff valve assembly with little free travel and high rigidity is favourable, to lower the influence of outside factors affecting the valve.

Fig. 11.59 Influence of the volumetric flow on the valve characteristic



Other than the interfering influence which has to be considered to keep the given valve indicator, there are also influences which can be used for a specific adaptation of the valve characteristic. It can be efficiently altered, for example, by a variation of the volumetric flow, based on a given shape of the control edge, see Fig. 11.59.

This effect becomes usable in pumps with variable volumetric flow, for example, to generate a speed dependence of the required wheel torques, providing low torques at low speed and high torques at high speed, to improve the control of the steering manoeuvres. In turn, a pump powered by the internal combustion engine has to transport a sufficient volumetric flow even at idling speed, to avoid an uncomfortable increase of the wheel torques at this operation point which is often used for parking manoeuvres.

11.10.3 Effects of Steering Movements: Volumetric Flow Splitting

So far, only the static case, without any movement of the rack, was considered in the examination of the valve indicator. If the rack is moved, the distribution of the volumetric flows in the Wheatstone bridge changes. A volumetric flow is taken from a place at the growing chamber (for example, between the bridges B1 and B4) and added at another place by the shrinking chamber (for example, between the bridges B2 and B3). These volumetric flows to and from the cylinder depend on the speed and surface of the piston.

Figure 11.55 shows an example: a movement of the rack towards the arrow at the speed x and against the load Fr. The valve is actuated according to the introduced load in such a way that the cross sections of the bridges B_1 and B_4 are widened and those of the bridges B_2 and B_3 are shrunk, so that $p_A \approx p_P$ applies on the high-pressure side and $p_B \approx p_R$ on the low-pressure side, i.e. the bridges B_2 and B_3 delimit the high-pressure area from the low-pressure area. Due to the symmetry of the control edges, the same volumetric flow $Q_2 = Q_3$ is flowing. The hydraulic cylinder takes a volumetric flow Q_A from the volumetric flow Q_1, so that $Q_2 = Q_1 - Q_A$ with Q_A depending on the rack speed x.

Equations 11.11 and 11.14 imply:

$$Q_2 = Q_3 = \frac{1}{2}(Q_P - Q_A(\dot{x}_r)) \tag{11.23}$$

$$Q_1 = Q_4 = \frac{1}{2}(Q_P + Q_A(\dot{x}_r)) \tag{11.24}$$

By analogy with the Eqs. 11.21 and 11.22, Δp is:

$$\Delta p = \frac{1}{4} \cdot \left[\frac{(Q_P - Q_A(\dot{x}_r))^2}{B_2^2} - \frac{(Q_P + Q_A(\dot{x}_r))^2}{B_1^2}\right] \tag{11.25}$$

$$\Delta p = \frac{\rho}{8 \cdot cD} \cdot \frac{(Q_P - Q_A(\dot{x}_r))^2}{A_2^2} - \frac{(Q_P + Q_A(\dot{x}_r))^2}{A_1^2} \tag{11.26}$$

This means that for increasing rack speed, the volumetric flow passing the two closed bridges will drop. To maintain the same power assist, their cross sections have to be reduced, the valve has to be actuated further, meaning more manual power. If, finally, the complete volumetric flow is absorbed by the steering cylinder, no more volumetric flow is flowing through the bridges, there is no more pressure drop there, whatever the cross section of the bridges. In this case the power assist suddenly fails ('Catch-the-pump').

Steering systems with variable pump (e.g., EHPS) enable a recording of the steering wheel rate, which the volumetric flow of the pump may follow within the scope of its efficiency. This helps to compensate the described effect. Esp. EHPS systems can raise the issued volumetric flow by lowering the system pressure, so that a sudden failure of the power assist can be avoided. The power assist will only drop proportionally to the rising steering speed when the limit of the system performance is achieved.

11.10.4 Valve Noises: Hiss

The principle of the valve demands rather high volumetric flows through narrow gaps. This is accompanied by a perceptible pressure drop in the fluid, entailing cavitation, if there is an overflow into the low-pressure area. HF hissing sounds develop, which are emitted as an airborne sound and as a structure-borne sound by the input shaft of the gear into the steering column. Whether airborne sound enters the passenger compartment depends on the installation position of the gear. The shape of the steering column determines whether structure-borne sound advances to the wheel and is emitted there as airborne sound.

The focus of gear development is on preventing the origin of noise. Attempts are made to achieve at least a very steady curve of the flow speed by corresponding design of the control edges. The given valve curve should be achieved by a shape

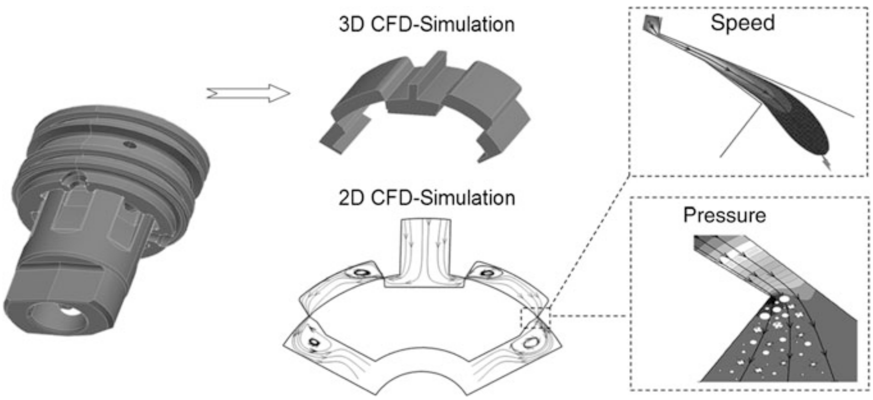


Fig. 11.60 Fluid flow in the actuated valve—CFD simulation

of the control edge that generates the desired back pressure without zones of very high flow speed. The corresponding layout is supported by CFD simulations.

Figure 11.60 shows an example of the given 3D-geometry and the calculation results in the form of flow speed and local pressure profiles.

Another means to reduce hissing noises is a higher return pressure, e.g., the specific use of throttles in the tank connection of the valve or in the runback. The effect is a generally higher pressure level in the whole gearbox, reducing the likelihood for cavitation even at consistently high flow speed. On the other hand, hydraulic losses are a drawback of this measure, the volumetric flow has to be transported by the pump against a higher pressure level. In the end, this increases the fuel consumption of the vehicle and the thermal load of the whole hydraulic power steering, so that additionally, an engine oil cooler may have to be used, increasing the weight and the costs of the system.

To handle this problem, variable throttles were developed for the runback, these are controlled over the flow pressure. The effect of the throttle is thus limited to those situations in which cavitation can occur in the valve. This function can be integrated into the steering valve. This valve design is described in Sect. 11.11.4.

However, a complete elimination of the cavitation is not feasible. Therefore, a second approach is to keep the remaining noise away from the driver and the passengers.

11.10.5 Internal Leakage

Internal leakage denotes the volumetric flow which flows through the valve when it is fully actuated, i.e. closed. This happens only at certain pressures, most of the time at the peak system pressure. This volumetric flow cannot be used to do work

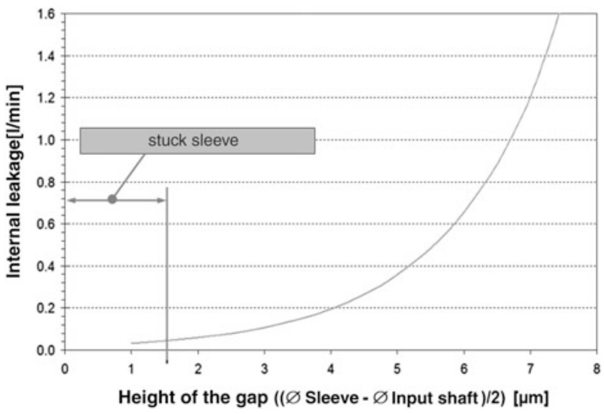


Fig. 11.61 Effect of the gap heights in the valve on internal leakage

in the cylinder, it is therefore a loss to be avoided. However, in contrast to the external leakage, internal leakage does not affect the function of the gear but only its efficiency.

Internal leakage occurs at different places of the gear:

- At the control edges which leave a residual gap between input shaft and sleeve even in the completely actuated state to avoid a contact of both parts and thus the risk of sticking. Therefore the bridges cannot be completely closed, entailing a permanent overflow by the valve from the high-pressure to the low-pressure side.
- At the cylindrical mating faces of input shaft and sleeve that should prevent an axial escape of the fluid from the high-pressure area of the valve. A small gap has to remain again to avoid contact of both parts so that the valve would be stuck. Figure 11.61 shows the internal leakage as a function of the height of this gap.
- At the sealing rings between sleeve and valve housings which should prevent an overflow of the fluid from a ring channel of the sleeve with high pressure into a ring channel with low pressure or into the runback space of the valve housing. With corresponding layout and correct assembly this leakage can to be assumed to be zero.
- In the cylinder over the piston ring or the mounting of the piston by overflow of the fluid from the high-pressure side of the cylinder to the low-pressure side. This leakage path can also be neglected if layout and assembly are correct.

Especially high-pressure steering systems with pumps feeding only a little flow to conserve energy, such as electrically powered hydraulic steering, high internal leakage affects the efficiency of the steering system. Corresponding optimisation helps to lower the internal leakage below 0.3 l/min at 100 bars.

11.10.6 Modelling (Position Control Circuit)

The regulation-technical aspects of a hydraulic power steering can be modelled as a position control circuit. A steering wheel angle is applied by the driver and transferred to the input shaft. This steering wheel angle corresponds to a nominal position of the rack. The external rack forces are acting as disturbance variables, caused by righting moments at the wheels or drilling torques of the wheels when steering at rest.

If there is a deviation between nominal and actual position of the rack, the torsion bar is used to close the control circuit on two parallel paths. First, a force proportional to the deviation is mechanically transferred to the rack by means of the torsion bar and the steering pinion. The second path leads over the valve, guiding a certain pressure into the corresponding cylinder chamber to support the desired movement of the rack. In the quasi-stationary case, the level of the assist power is determined by the valve curve. If the rack force is sufficient, the rack moves towards its nominal position. Indeed, the valve, being a controller in this circuit, has no I portion, so that some deviation will always remain as a function of the applying external forces.

This explains that even a steering system follows control rules with corresponding natural frequencies and damping, in particular it has a stability limit. If that limit is crossed, vibrations occur in the steering system which are perceptible or audible. They are perceived by the driver as very disagreeable.

To precisely determine the stability limit, the dynamic behaviour of the steering system is modelled and complemented by the following parts and effects:

- masses, elasticities and damping of the different mechanically coupled subsystems
- non-linearities, such as the friction between quite moving parts
- capacities and inductances of the hydraulic system originating from stretching hoses under pressure and mass and flow speed of the fluid
- resistances to flow in the hydraulic system from throttles or orifices
- external factors, such as the control behaviour of the pump for varying pressures.

Hence, the parameter setting for the correct modelling of a steering system in the simulation is rather complicated and assumes intensive recording of the characteristics of the steering system in tests.

11.10.7 Damping: Instabilities

An essential factor for the stability reserve of the steering system is the gradient of the valve characteristic (pressure difference in the cylinder over wheel torque) which shows the amplification in the control circuit. In principle, it is desirable to lay out the valve characteristic in such a way that beyond a certain pressure, which is never achieved during driving but only in a standing vehicle, the wheel torque rises as little as possible. This enables a good feedback of the steering to the driver in the operating range and, yet, quite low wheel torques during parking. On the hand, this would require a very large gradient of the valve curve, which entails instabilities in the steering system. Hence, in reality, one aims for a high gradient that is sufficiently far from the stability limit.

To lift the stability limit even further, it is also possible to increase the damping in the steering system. In the easiest case, a resistance to flow is introduced into the hydraulic system, e.g., a throttle in the runback line. It generates a resistance that is dependent on the volumetric flow and takes energy from the system to lower the amplitudes for a non-stationary curve of the volumetric flow. The pressing of all seals at their mating faces, reinforced by the higher system pressure, entails additional friction and dampening. The effect of the throttle is the bigger, the higher the back pressure. However, the limits are narrow, because the throttle losses decrease the efficiency of the steering system, increase the energy consumption caused by the steering system and load it very strongly with heat.

There is also the possibility to throttle the volumetric flow streaming out of the cylinder chamber (see also Sects. 11.11.3 and 11.11.4). A damping proportional to the rack speed results. In addition, there is less kickback of the steering system, i.e. the gear transmits less external force impulses to the driver, caused, for example, by road bumps. Mind that narrow limits are set here as well, because quick steering movements generate much higher rack speeds than would occur in an unstable situation, so that a throttle setting yielding sufficient damping will often intolerably limit the efficiency of the steering system in evasive manoeuvres.

11.11 Additional Hydraulic Systems

Additional systems for hydraulic steering are used to further improve its properties, based on the standard parts of the gear, both by resolving opposing target situations and by shifting the limits of the system beyond those of the standard layout.

11.11.1 Centring

The standard valve layout provides a power assist even for small wheel torques. However, in the region which is relevant for straight driving with small course corrections, the function of the hydraulic steering should be very similar to a mechanical steering. The standard valve desing offers a driving of the input shaft to the steering pinion with very high stiffness, this is favourable for precise steering in the mentioned driving conditions. Nevertheless, this stiffness is defined by the torsion bar in the hydraulic steering. At the same time it has to permit a sufficient relative torsion between input shaft and steering pinion to close the corresponding gaps in the valve, without applying too high wheel torques. A fundamental disadvantage of the hydraulic steering develops. The layout of a hydraulic power steering aims at using a very stiff torsion bar without raising the wheel torques for cornering or parking too much.

To solve this conflict, a centring torsion bar is used, as depicted in Fig. 11.62, both disassembled and assembled. A coarsely rigid driving is produced by a pretense unit parallel to the torsion bar, until the wheel torque surpasses the centring torque applied by the pretense unit. Then the input shaft starts to torsion relative to the pinion, and the additional righting moment is applied by the torsion bar.

The depicted system is an example of such a parallel unit. It consists of two rings, three calottes are introduced at angles of 120° into one of the facing sides. The rings are arranged in such a way that the calottes lie on top of each other and that a sphere lying in between them establishes a positive contact between both rings.

If both rings are torsioned relative to each other, the sphere moves up a ramp in both opposite calottes and presses the two rings apart.

One ring is connected to the sleeve of the valve and to the steering pinion by grouting. The other ring disposes of axial grooves in its inner diameter. These grooves correspond to axial grooves which are introduced, in addition, into the input shaft. The positive contact is again established by spheres running in the grooves between ring and input shaft. Therefore the rotation of the input shaft is transferred to the ring, while the degree of freedom required for an axial movement relative to the shaft is granted.

A centring force is exercised by a spring between the input shaft and the attached ring. It is transferred by the spheres to the ring fastened on the sleeve. A relative torsion between input shaft and sleeve and the attached rings converts

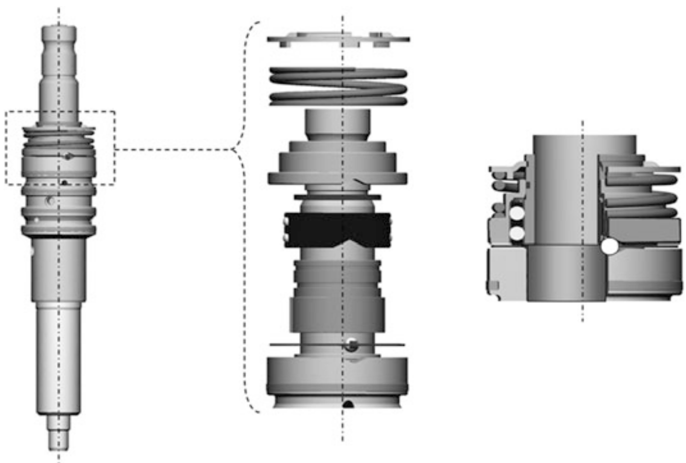


Fig. 11.62 Centring by means of a spring-loaded axial rotary clutch

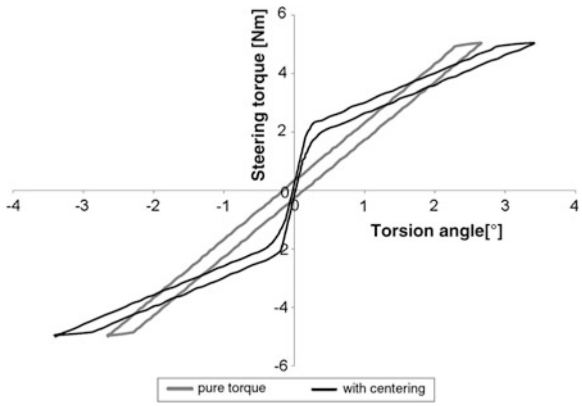


Fig. 11.63 Wheel torque over angle of torsion in the comparison

the axial spring tension into a righting moment which acts with the righting moment of the torsion bar. In contrast to the torsion bar, this mechanic may use the shape of the calottes (ramp angle) and the pretension of the spring to set a torque which is (almost) rigidly transferred from the input shaft to the steering pinion, without relative torsion and, hence, without actuation of the valve. This results in the desired centring, shown in Fig. 11.63 where it is compared to a steering valve which has only a conventional torsion bar.

In addition, there is the option to choose spring stiffness and shape of the calottes so that the effective valve stiffness and valve characteristic are even more controlled, for example, by a degressive curve of the ramp angle.

There is also the option to use this mechanism for a parameterised steering that may be speed-sensitive. This is discussed in Sect. 11.11.2.

An additional system for centring permits separating the operation ranges of the valve for straight driving and cornering within certain limits and to optimise them individually. This reduces conflicts and increases the degrees of freedom by tuning.

By proper choice of the centring torque, the operation point of the power assist is set in such a way that the steering during straight driving is very precise. Then the torsion bar stiffness can be set lower, so that the wheel torques rise less steeply to the parking range with increasing rack force than in a conventional system. This improves the steering comfort in such situations.

Centrings for steering valves exist in many configurations which are also more compact, easier and cheaper than the version discussed here. Their effect is similar to the described solution.

11.11.2 Speed Dependence

Conflicts in valve tuning result from the vehicle speed, too. At very low speed and in particular when parking, the steering should be very smooth to provide the best steering comfort, i.e. to offer high power assist for low wheel torques. At high speed, the support has to be much lower to maintain sufficient steering precision and to avoid accidental jerking around of the steering. Yet the full power assist should be available to the driver at need, for example, to keep control of the steering if a front wheel has left the road and rolls on a soft shoulder. In the layout of the vehicles, high-speed safety dominates esp. in Europe.

A speed-sensitive influencing of the valve characteristic grants a solution of this conflict and more parking comfort.

There are two basic technical concepts: Setting the volumetric flow in the steering valve allows, as described, a variation of the wheel torque. This principle is used only in systems which supply the gear according to demand by a variable pump.

There are other systems that vary the applying wheel torque beyond a certain relative torsion between input shaft and pinion, they set the effective stiffness of the torsion bar, in other words, they adapt to the respective driving situation: A low stiffness if high support is desired, a high stiffness if a driving with low elasticity is important.

A conventional torsion bar with very low stiffness is complemented with a parallel additional system in the construction, supplying an additional righting moment that can be controlled from the outside.

A central part of such systems is a mechanism that transform the relative torsion of the valve into a linear movement along the valve axis, as discussed in Sect. 11.11.1 for the centring. The ring linked with the input shaft of the additional system is here a hydraulic piston, the reaction piston, with a piston ring dividing the space behind the valve drilling of the case into two chambers.

A spring sets a pretension of the system and a rigid driving at low wheel torques, as in the centring system. Beyond this torque, the valve is actuated. The torsion bar is not very stiff, so that the mechanically generated righting moment rises only little. The actuation of the valve raises the flow pressure. This pressure is diminished by a driven magnetic valve and fed into the chamber between the reaction piston and the input oil seal. The axial force generated by the pressure on the reaction piston generates the main part of the righting moment in the valve.

The magnetic valve creates a fixed ratio between flow pressure and reaction pressure as a function of electric power. This allows to set different righting moments by modifying the flow through the magnetic valve to a specific flow pressure.

Finally, an excess-pressure valve in the reaction chamber ensures that the wheel torques do not rise too high if the full power assist is required in extreme situations while the magnetic valve permits high reaction pressures at the same time. This is also a part of the safety concept which adjusts the characteristic curve at the highest reaction effect if the additional system has failed. Then the high-speed response of the steering can be set while the driver is able to achieve the maximum pressure without applying too high manual torques.

Figures 11.64 and 11.65 show this system and the possible variation of the valve curve.

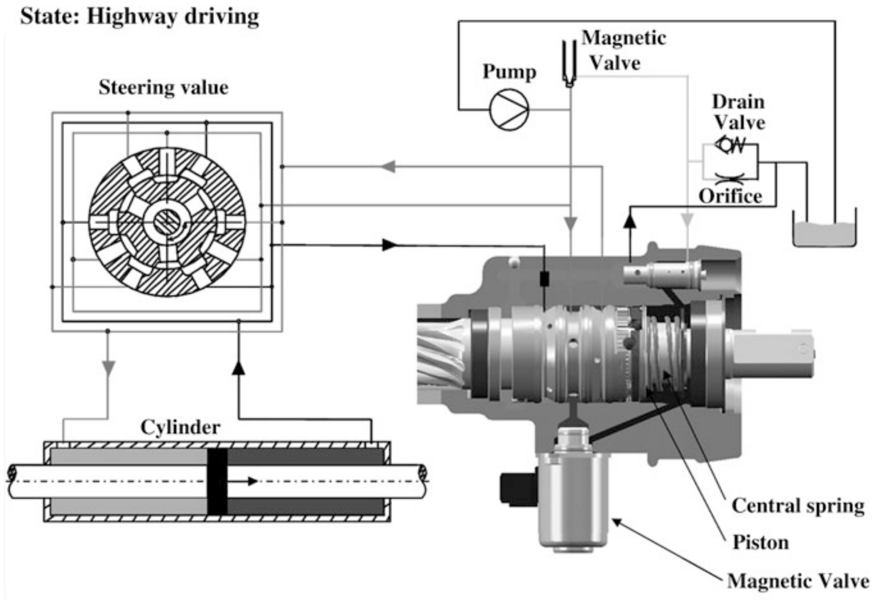


Fig. 11.64 Representation of a system with hydraulic reaction

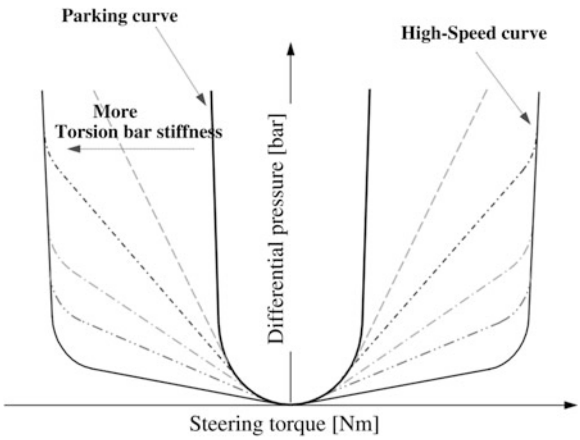


Fig. 11.65 Valve characteristic of a system with hydraulic reaction

It can be seen that the valve characteristic can be divided into three areas. Around the middle position, the pretension of the centring spring acts, so that no power assist builds up. At higher wheel torques, the cylinder pressure over the wheel torque rises about linearly, its gradient depends on the electric power of the magnetic valve. The characteristic curve with the lowest wheel torques (often called parking characteristic [no translation found]) uses only a low reaction pressure to prevent the reaction piston from lifting. For the characteristic curves with a more level gradient, a steep increase of the gradient is then observed beyond a certain wheel torque. A further rise of the reaction pressure is prevented by the excess-pressure valve in the reaction chamber.

This design provides the most freedom in valve layout by tuning the shape of the control edge at the valve, the torsion bar stiffness, the shape of the calotte, stiffness and pretension of the centring spring, the excess-pressure valve in the reaction chamber and the ratio of reaction pressure to flow pressure. This is opposed by a considerable additional expenditure of parts for the valve and the whole steering system, in particular the magnetic valve, its electric connection and an ECU, to supply it with power according to the given parameters, such as the driving speed.

More internal leakage occurs, because the volumetric flow in the reaction chamber is taken from the inlet. This additional leakage rises with the reaction pressure and has to be considered during the layout of the hydraulic pump.

A more simple system falls back on the same mechanics, but pressurises the piston from the other side. Here the space between reaction piston and input oil seal is directly connected with the runback. The exhausting fluid from the valve, though, is still dammed up by an electrically adjustable throttle valve, before it can flow away into the steering runback. The centring spring is powerful in this system. The reaction piston generates an opposing force to the centring force of the spring, as a function of the back pressure caused by the electrically adjustable throttle valve.

The variation of the valve curve is achieved mainly by a change of its dead band. The setting for high speed does not dam up the runback, and the full centring force of the spring acts. For parking, the centring force of the spring is almost completely eliminated. However, the slope of the valve characteristic is also determined by the torsion bar stiffness and, in addition, by the form of the calottes. It cannot be varied.

This system is a little cheaper because it does not fall back on an adjustable pressure converter but only on an adjustable throttle. Its disadvantage is the accumulation of the whole volumetric flow in the runback, resulting in higher losses and more heating of the system in driving situations when only low wheel torques are aimed at.

11.11.3 Damping Valves

This additional system has the purpose to reduce the steering kickback when force impulses are initiated from the outside. This happens, essentially by integrating a steering damper into the rack-and-pinion steering, as was occasionally used in steering systems with a ball-recirculating gear with nut.

The cylinder and the rack with piston assume the supply of the power assist and the function of the identical parts of a steering damper. To receive a speed-sensitive damping, the volumetric flow leaving the cylinder chamber has to be throttled, so that a force develops against the direction of movement. However,

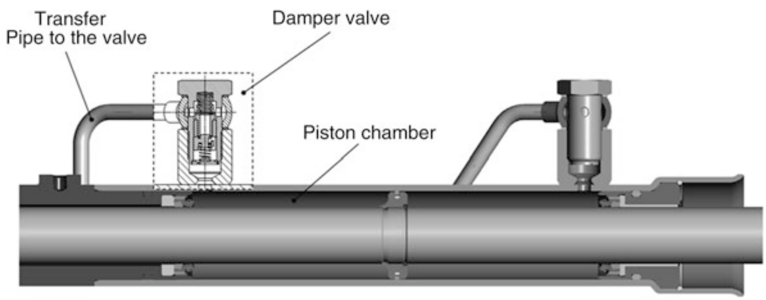


Fig. 11.66 Cylinder with hydraulic connections and damping valve

the streaming volumetric flow may not be throttled to avoid cavitation and aeration in the cylinder chamber.

The following conflict results: An effective damping of external force impulses requires a cutback which would be so powerful that a desired quick steering movement could not achieve the required rack speed even when the peak pressure was issued into a cylinder chamber. Thus it would not support evasive manoeuvres accordingly. This has to be counteracted by limiting the pressure difference from the throttle with an excess-pressure valve.

Hence, a damping valve like in Fig. 11.66 is attached directly to the hydraulic connections of the cylinder and combines the function of three hydraulic parts:

- The check valve that allows fluids to stream unhindered
- The throttle which builds up a pressure dependent on the volumetric flow as the fluid leaves
- The excess-pressure valve which limits the peak back pressure caused by the throttle.

Figure 11.67 shows one example of a characteristic curve for a damping valve passed in damping direction.

11.11.4 Steering Valves with Damping Qualities

Proper design enables integrating the functions of the damping valves into the steering valve to a great extent. Figure 11.68 shows the configuration of such a valve.

The purpose is to activate the throttle function only at need, i.e. when cornering, when the strongest kickback occurs.

At the same time such steering valves offer a solution for hydraulic instabilities in the steering system. The conflicts between damping in the steering system and efficiency and load, mentioned in Sect. 11.10.7, can be resolved this way. The throttles are activated only when instabilities can occur. The usual valve curves

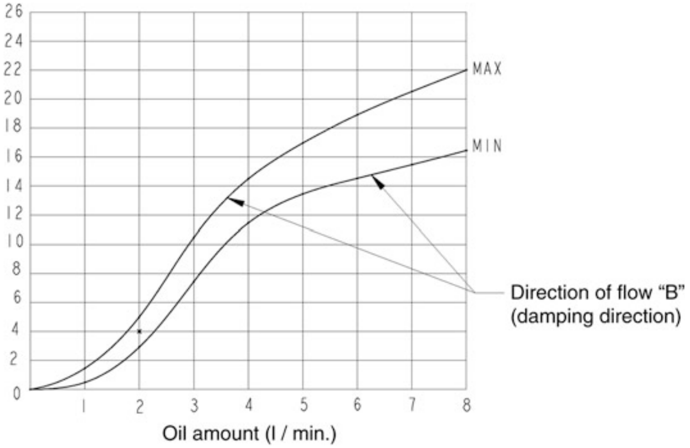


Fig. 11.67 Characteristic curve of a damping valve

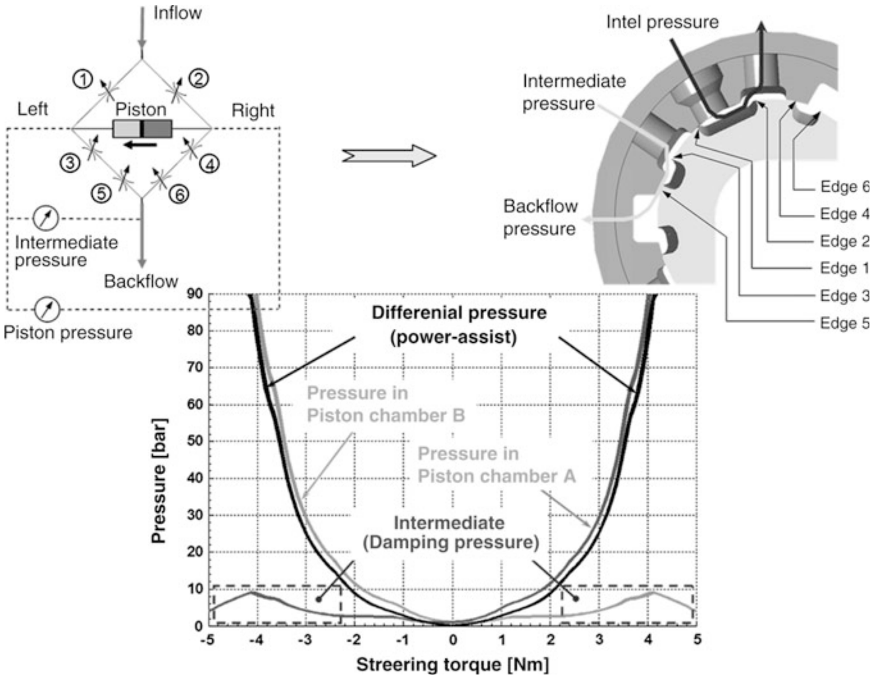


Fig. 11.68 Steering valve with defined damping by additional control edges

have a flat gradient around their neutral position, so that there is a sufficient stability reserve, even without additional measures. The biggest part of driving takes place here. Passing into the parking range, at high pressure, makes a gentle introduction of the throttle desirable.

This function can be integrated into the steering valve by an additional control edge pair, passed by the fluid before it leaves the valve and streams into the runback.

In the neutral position of the valve, these control edges leave a wide gap for the discharge of the fluid. If the valve is operated, the control edges on the high-pressure side close like those of a conventional valve, while those on the low-pressure open even further. On the low-pressure side, the additional control edges reduce the cross section of the gap if the valve is strongly actuated. The discharging fluid is throttled. The throttle is not activated on the high-pressure side.

Therefore a slight counterpressure against the high working pressure in the accordingly pressurised cylinder chamber is generated in the opposite chamber. Its level depends on the applying volumetric flow. Since oil is squeezed out of the attached chamber as the rack moves, the volumetric flow rises proportionally to the rack speed, so that a speed-sensitive damping results from an instationary movement of the rack, effectively reducing the vibrations. The intensity of the throttle effect is controlled by the layout of the additional control edges.

11.11.5 Pressure Limitation in the Rack-and-Pinion End Position

An undesirable effect of hydraulic gears is the peak system pressure applying as the steering is held at the end position. The steering valve is fully modulated then and allows only a small volumetric flow to pass, while the pump works against the peak pressure and conducts almost the entire transported volumetric flow through the excess-pressure valve back into the tank.

This causes a mechanical load of the steering system by acting forces and pressures. It also causes a very quick heating of the hydraulic fluid. Especially in pumps driven by the internal combustion engine, the permissible maximum temperature of the fluid can be passed within less than a minute if very high engine speeds apply at the same time. Adjusting and maintaining this operation mode is ultimately an abuse of the car, but it cannot be excluded for the whole range of end customers, and is difficult to prove if the customer complains about a failing system.

Another problem develops because the driver suddenly imposes a high load for the pump when the steering is already at the end position, the valve is in neutral position (no applying wheel torque) and the driver steers towards the end position. If the engine is idling, the control of the idling speed cannot always compensate this sudden additional load fast enough—there is the risk that the engine shuts down as a result of the steering movement. This can be avoided by raising the idling speed. But this significantly increases the standard consumption of the vehicle, on account of the idle running phases which are frequent in the test cycle.

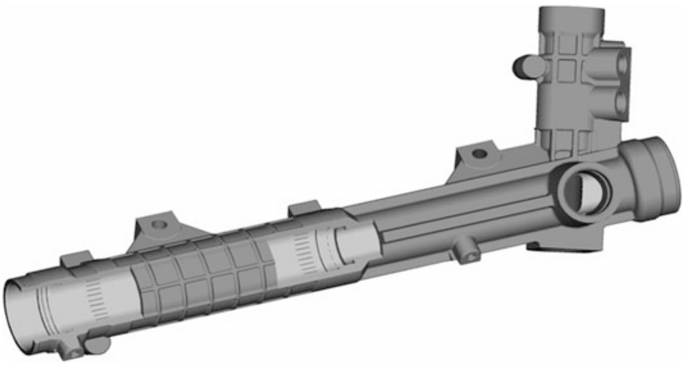


Fig. 11.69 Cylinder with overflow grooves in the piston end positions

A practical solution of these problems is a pressure limitation at the end position. This is made by inserting overflow grooves into the cylinder near the end positions of the piston. They enable a leakage oil flow from the high-pressure side of the cylinder to the low-pressure side if the piston crosses them, see Fig. 11.69.

Depth and number of the grooves are set in such a way that all the volumetric flow given off by the pump is passing through them with the desired rest pressure. A very low rest pressure is aimed at. It is limited, nevertheless, by the need to provide a sufficient hydraulic rack force for steering the opposite way, away from the stop unit, so that the wheels are moving without additional wheel torques from the driver (manual steer). The required force is lower than that needed for steering towards the stop unit, because of the weight righting of the steering.

The thermal load of the steering system is significantly lowered by a pressure limitation at the end stop, so that additional engine oil coolers can be renounced in most cases. Part of the add-on costs for this system is thus compensated. Besides less loss at the end position, this is also an effect of better thermal absorption, because the pump does not conduct the hot hydraulic fluid directly to the tank, but through the pipes to the gear, then through the valve and both cylinder chambers and then through the runback to the tank. It can emit heat this way.

Another advantage is that noises from steering at the stop unit are reduced by the lower pressure. To be precise: mechanical noise from touching the end position and hydraulic noise which can be caused, otherwise, by the high pressure gradient. If this noise occurs in gears without pressure limitation with an annoying intensity it has to be eliminated. To do so, an additional elastic element between the end position in the gear and the axial joint of the tie rod is inserted.

A limiting factor for the application of such a system is the layout of the chassis structure. The righting moments applied by the axle at the full steer-angle are often very low, especially in vehicles with front-wheel drive. There is a risk that the remaining support is not sufficient to steer away from the end position at need. That is perceived by the driver as a subjectively unpleasant ‘sticking’ of the steering.

11.12 Ball-Circulation Gears with Nuts/Utility Vehicle Steering Systems

11.12.1 Fields of Application

The ball-recirculating gear with nut is the traditional form of steering which converts a rotation of the steering wheel into a swivel of the steering arm which is transferred by the track and tie rods to the wheels.

This transmission by leverage is also the main advantage of their modern applications: vehicles whose guided axle is rigid are almost always equipped with ball-recirculating gears with nut. By a corresponding layout of the steering structure they permit mounting the gear to the frame and following the complicated spatial movements of the rigid axle with little kinematic repercussions on the steering system. This applies in particular for axles which are attached only by leaf springs.

However, a rack-and-pinion steering has to be connected solidly with the body of a rigid axle. This produces conflicts with package and steering kinematics and demands for a very loadable and highly flexible attachment of the steering column. The joints in the steering column have to cover a wide range of angles. Their length offset should have a very high stroke, so that the joints are able to follow any possible relative movement between the axle with gear and the frame or even that of a driver's cab that is elastically supported on the frame.

Nowadays, vehicles with rigid front axles are mainly found in two categories:

- cross-country vehicles destined for off-road service that should admit extremely high axle crossing
- utility vehicles of the middle and heavy class with permissible front axle loads of more than 2 t.

In typical configurations, either the gear is mounted to the body frame on the driver's side, the corresponding front wheel is connected by a track rod and both front wheels are connected by the tie rod (shown in Fig. 11.70), or the gear is placed in the middle of the car and both front wheels are coupled directly by tie rods.

No sufficiently large rack-and-pinion gears are available for coaches with independent suspension in the front axle. A steering quadrangle is usually used for applications in these and for the rare use of ball-recirculating gears with nut in passenger cars. One more track rod is sometimes used as a connecting element in coaches, on account of the driver being seated far in front of the front axle.

All arrangements have in common that they contain many elements which introduce friction and elasticities or even free travel into the steering system. They affect in general the steering feel and precise steering. Lots of leverage elements also indicate an expensive steering system, so that the ball-recirculating gear with nut was displaced by the rack-and-pinion steering wherever it is available as a feasible alternative.

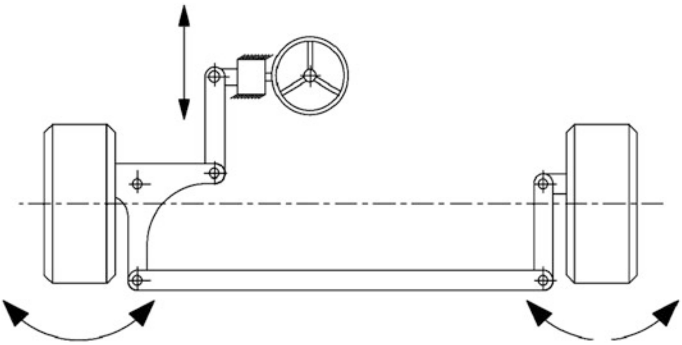


Fig. 11.70 Connection between gear and front wheels in utility vehicles with rigid front axle

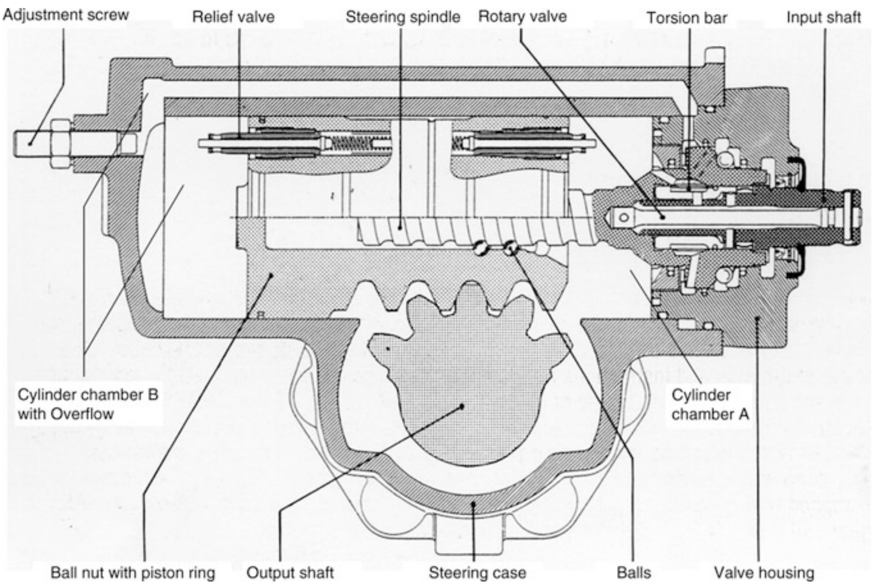


Fig. 11.71 Ball-recirculating gear with nut in a cutaway view

11.12.2 Configuration of Ball-Recirculating Gears with Nut

Ball-recirculating gears with nut are produced only in an integrated module, so that the mechanical part of the gear and the hydraulic part are gathered in a common case. This module is also called integral steering or block steering.

The ball-recirculating gear with nut, of which Fig. 11.71 shows a sample cut-away view, can be separated into the following subassemblies:

- Valve with valve housing and input shaft
- Steering shaft, linked with the valve

- Ball nut and piston, linked with the steering shaft by balls
- Steering shaft with dovetailing segment.

The input shaft, linked with the steering column, and the valve are similar to shaft and valve of a hydraulic rack-and-pinion steering. It is another rotary disk valve with input shaft, sleeve and torsion bar. However, the valve is designed for much higher volumetric flows and higher working pressures, compliant with the purpose of the gear, so that, for example, the circulatory cross sections are wider and the walls of the sleeve are thicker.

There the spindle follows instead of the pinion, it is connected to the steering nut. A rotation of the spindle is converted into a translation of the ball nut and the positive connection between them is established by recirculating balls, so that the friction is low. The opposite end of the ball nut is a piston in a cylindrical part of the steering case. The cylinder chamber, located there, is connected by a channel in the case with the corresponding ring groove of the sleeve, while the other ring groove is connected directly with the whole remaining cylinder space. Waste-gate valves are in the ball nut, they are operated if the piston approaches the end position. They open a transfer canal, conducting fluid from the high-pressure side to the low-pressure side with a defined rest pressure (cf. Sect. 11.11.5). This avoids unrequired loads of the whole steering system when the steering is kept at the end position.

The movement of the ball nut is transferred by a dovetailing to the steering shaft and then produces a swivel of the tightly pressed-on steering arm at the outside of the case. The dovetailing has a constant or a variable gear ratio, according to the configuration of the gear.

11.12.3 Comparison Between Ball-Recirculating Gear with Nut and Rack-and-Pinion Steering

Resemblances between a ball-recirculating gear with nut and a rack-and-pinion steering are only found in the steering valve which is made in the same manner.

All together, the ball-recirculating gear with nut is much more robust and suitable for applications with hard conditions or that have to merge high reliability with a long service life, as, for example, utility vehicles, which are usually laid out for a road performance of 1 million km.

One reason for the high robustness is the sealing system with only one radial high-pressure oil seal at the steering shaft. The shaft is only rotating there and, hence, easier to be sealed than a rack with two high-pressure seals against which it is linearly moving. This allows for peak working pressures of 185 bars, soon to become 200 bars.

A ball-recirculating gear with nut offers a better damping against external impacts as well, chiefly because of the mechanical efficiency of the ball-recirculating gear with nut. It is higher for movements applied by the input shaft than for repercussions of the ball nut, so that these are damped by more friction. Finally, the ball-recirculating gear with nut is more compact and, hence, suggests to cover a wide spectrum of vehicle applications with few standard products. Then any required individual adaptations are carried out in the design of the steering linkage and the support for the steering. This way, the financial benefits of mass manufacturing may be exploited even for specified applications which are built only in small numbers, as is common with some utility vehicle models.

This is opposed by drawbacks of the whole steering system. Including the required steering linkages, a steering system with ball-recirculating gear with nut is much heavier and, at least when build in great numbers, more expensive than a rack-and-pinion steering. The huge number of transmission elements between steering column and wheels and their elasticities and frictions at the joints affect the steering feel—a rack-and-pinion steering permits a more immediate perception of events at the wheels. These factors are important for passenger cars where the advantages of the ball-recirculating gear with nut are limited. That is why rack-and-pinion steerings are almost always used in passenger cars.

11.12.4 Technical Data and Parameters

See (Table 11.6).

11.12.5 Additional Systems

On account of the equal configuration of the steering valve, the respective systems for centring the torsion bar (Sect. 11.11.1) and speed-sensitive control of the valve characteristic (Sect. 11.11.2) are also found in ball-recirculating gears with nut. They are somewhat similar to the configurations for rack-and-pinion steering.

Valve designs that introduce specific damping into the gear (Sect. 11.11.4) can be transferred to ball-recirculating gears with nut as well. In utility vehicle applications, they are more important than in passenger cars, on account of the high power assist and the resulting high power of the valve.

Damping valves (Sect. 11.11.3) are not required, due to the mentioned damping qualities of the ball-recirculating gear with nut against external impacts. A pressure limitation at the end stop (Sect. 11.11.5) is already included in the standard layout. One special additional system found in gears for heavy utility vehicles with several guided axles is an option to connect an auxiliary cylinder.

Instead of designing large special gears for these vehicles, which are usually produced in quite small numbers, the required additional steering force is introduced by coupling an auxiliary cylinder to the steering linkage near the second guided axle, that is required anyhow. This cylinder is supplied over the steering valve, since the valve housing has hydraulic connections linked to the corresponding ring channels

Table 11.6 Some parameters of hydraulic ball-circulation gears with nut

Vehicle class	Cross-country vehicles	Light utility vehicle	Moderately heavy utility vehicle	Heavy utility vehicle	Construction vehicles
Front axle load (t)	1.8	2.5	5.5	7.5	9.5
Maximum source torque (Nm)	1,200	1,700	5,000	6,500	8,500
Maximum working pressure (bar)	120		715–185		
Volumetric flow (l/min)	6	8	12	16	25
Swivelling angle output shaft	±90° to ±100°				
Gear ratio	Approx. 16:1 to approx. 26:1				
Temperature	−40 to +120 °C				

of the valve sleeve. This permits using the same gear that is found in cars with one guided axle, except for this small modification, and keeping the load of individual parts very low by distributing the power supply according to demand.

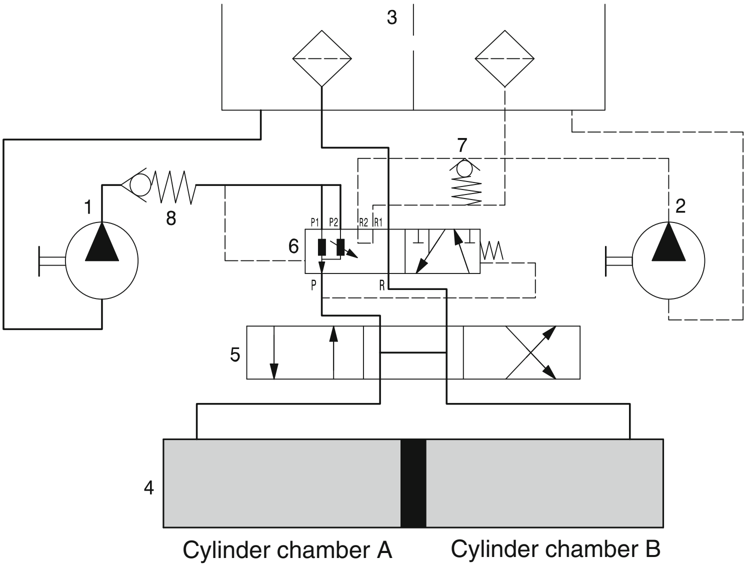
11.12.6 Dual-Circuit Steering

Heavy utility vehicles with very high front axle loads or two front axles may fail to observe the registration regulations for operation at power assist failure when the steering is used mechanically exclusively. The dual-circuit steering, whose schematic hydraulic diagram is shown in Fig. 11.72, maintains a sufficient volumetric flow of the steering oil supply even when one steering circuit has failed.

Normally only the pump powered by the internal combustion engine (1) is connected with the gear (4/5) by the dual-circuit transfer valve (6). A slight cutback of the volumetric flow in this valve helps to install a control pressure which actuates the valve against a spring into the displayed position. The runback leads over this valve into the tank as well. The tank has two separate chambers which are interconnected above the least level.

The auxiliary pump (2) is mechanically connected to the live axle, usually at the output shaft of the drive gearbox. This helps to ensure that the required volumetric flow can be provided even when the internal combustion engine has failed, as long as the vehicle moves. Normally, the transported volumetric flow is conducted by the dual-circuit transfer valve back into the tank (3). A slight cutback is present here as well. The developing back pressure is monitored to warn the driver if the second control circuit fails (he or she would not notice otherwise) and to request stopping the vehicle.

If the primary control circuit fails, there is no control pressure at the dual-circuit valve. Then the spring adjusts the valve in the second switch position. Now the gear is fed by the second control circuit, a leak loss over the main pump is prevented by the check valve (8). The driver is warned and asked to stop the vehicle.



1 Main pump (internal combustion engine)
2 Auxiliary pump (gearbox output)
3 Tank with two chambers
4 Steering cylinder
5 Steering valve
6 dual-circuit transfer valve
7 Excess-pressure valve auxiliary circuit
8 Check valve of the main pump

Fig. 11.72 Scheme of a dual-circuit steering

The gear itself and all its parts are considered sufficiently failsafe, so that no further redundancy is necessary.

11.13 Requirements for a Hydraulic Gear

The general requirements for function and strength of hydraulic gears are, in the end, an extension of the requirements for mechanical rack-and-pinion gears with specific tests of the steering hydraulics.

Hence, only the additional requirements shall be discussed here.

11.13.1 Functional Requirements

The function tests are complemented by measurements of the valve characteristic and the internal leakage. Both were discussed in Sect. 11.10.

11.13.2 Strength Requirements

11.13.2.1 Static Strength: Burst Pressure Test

The static strength of the hydraulic system is tested by introducing a volumetric flow into the gear when the steering valve is closed. The flow is controlled in such a way that the pressure rises steadily. The gear has to remain tight, up to the peak working pressure including a safety reserve, and it has to stay fully functional. In addition, the burst pressure, i.e. the point at which a part of the steering hydraulics fails, has to lie above a certain limit value, say, at least three times the peak working pressure.

11.13.2.2 Dynamic Strength: Pressure Pulsation

The dynamic strength of the steering hydraulics is tested by generating pressure pulses in the gear. This may be done either by actuating the steering valve when the rack is blocked or by introducing a varying rack force. A high number of pressure pulses is desired, hence, the excitation should be high-frequency (5 Hz or more). The gear has to be fully operatable after this test.

11.13.3 Environmental Requirements: Cold Start Test

An important quality of hydraulic gearboxes to be checked is the sealing under cold circumstances. As described, seals lose their flexibility when it is very cold, and they may not be able to sufficiently follow movements of the rack, the steering pinion or the input shaft any more. A cold start simulation test to check whether hydraulic oil is leaking under these conditions.

Down to a temperature of −20 ˚C, this must not happen at all. At lower temperatures, most tests run down to −40 ˚C, some millilitres of leakage are tolerable, according to the purpose of the vehicle.

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