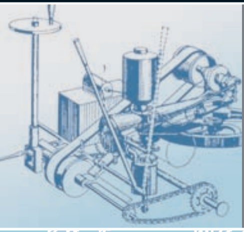
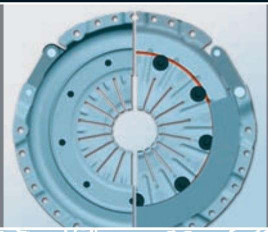
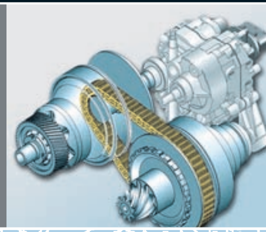
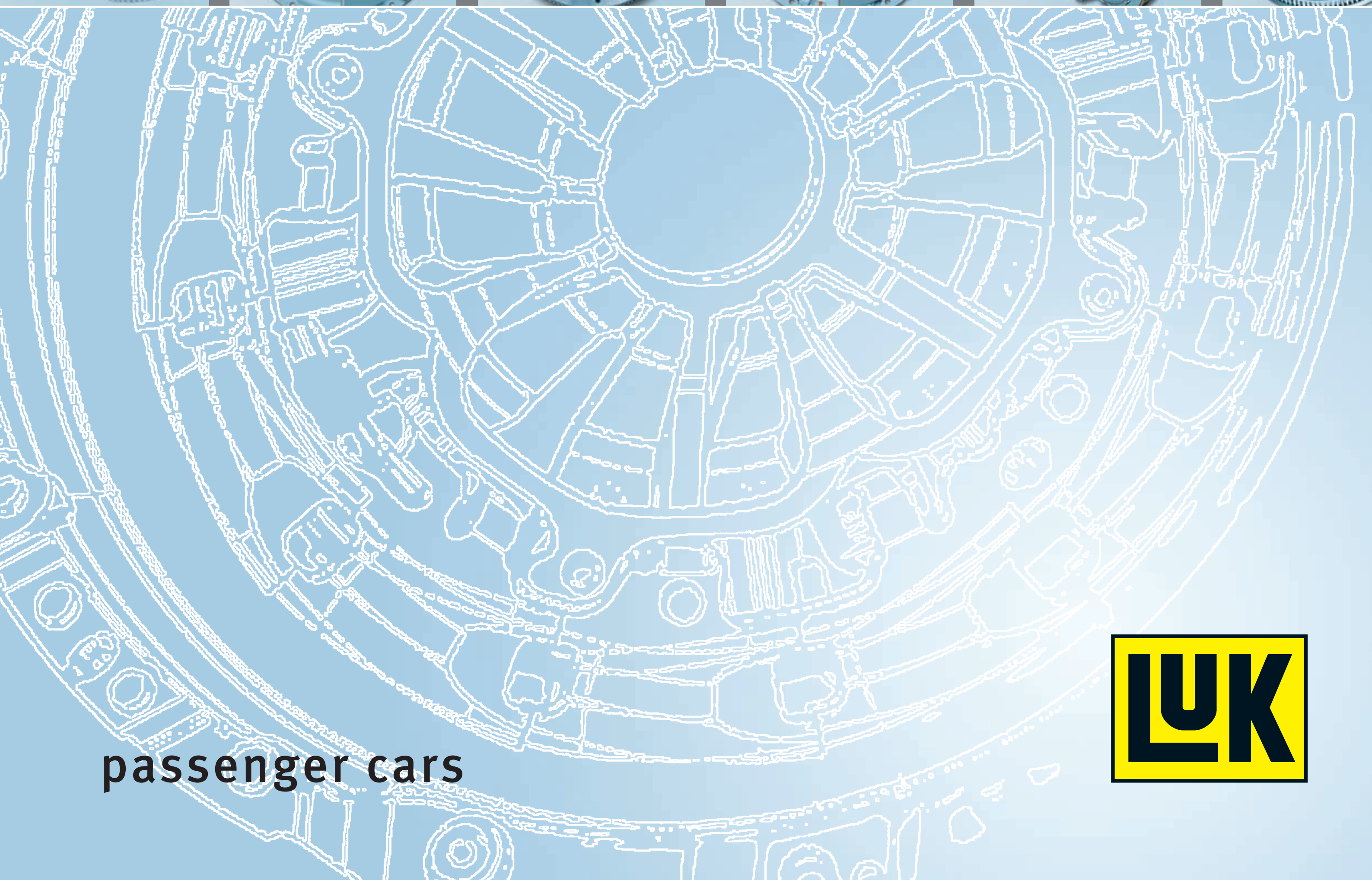
LuK Clutch Course

An introduction to clutch technology for passenger cars









passenger cars

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A history of clutch technology

In the course of over 100 years of automotive history, nearly all components have undergone enormous technological developments. Reliability, production costs and service-friendliness as well as, more recently, environmental safety, have been and continue to be the criteria demanding new and better solutions from automotive engineers. The basic designs are usually known early on, but only the availability of new materials and processing procedures makes their realisation feasible.

It was not until the end of the first decade of this century that the internal combustion engine surpassed the competing steam and electricity-based automotive drive concepts on a large scale. In 1902, a petrol-engined vehicle for the first time broke the overall speed record; up to then, electric and steam-powered vehicles had set the standards, and proponents of the three drive concepts continued to compete for the absolute speed record throughout the first decade.

Steam and electric drives have a decisive advantage over "motorised vehicles with liquid fuels", as they used to be called. Thanks to the almost ideal torque band, they required neither clutches nor transmissions, and thus were easier to operate, had fewer malfunctions and were easier to service. As an internal combustion engine only delivers its output at engine speed, there must be a division between engine and transmission. The speed-dependent drive principle of the petrol engine necessitates a mechanical aid for starting, as sufficient output (torque) is only available after certain engine speeds have been attained.

Besides the function of a starting clutch, however, that of a dividing clutch is equally important, for it allows load-free gear changing while driving. Because of the complexity of the related problems, many smaller vehicles in the early years of automotive design did not have a starting clutch; the motor car had to be pushed into motion.

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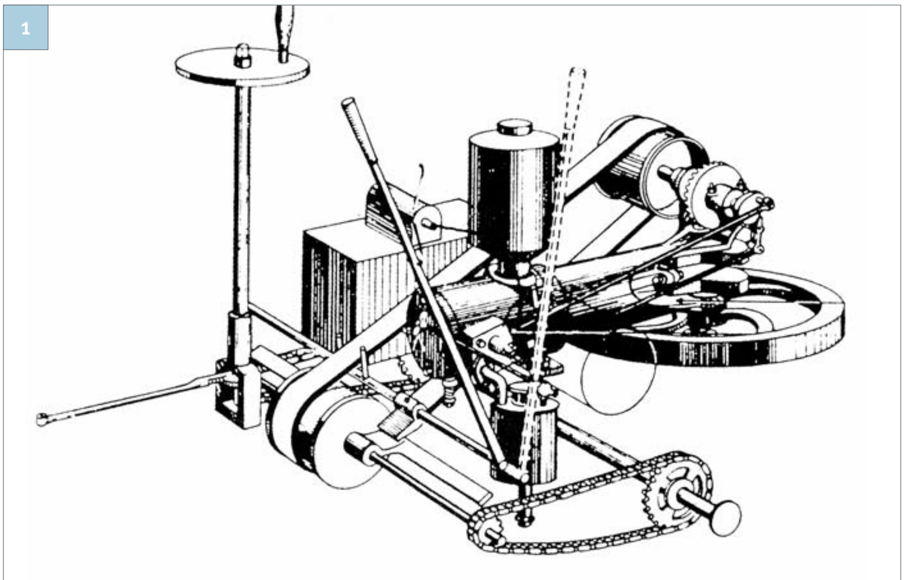

Transmission belt clutch from the Benz patented motor car of 1886

The operating principles of the first clutches originated in the mechanised factories of early modern industry. By analogy with the transmission belts used there, flat leather belts were now introduced into motor cars. When tensioned by a roller, the belt transmitted the drive output of the engine's belt pulley to the drive gears, and when loosened, it slipped through – i. e. disengaged. As this procedure caused the leather belts to wear out fast, a new tactic was adopted of installing an idler pulley of the same size beside the drive belt pulley. By moving a lever, the transmission belt could be guided from the idler pulley on to the drive pulley. The motor car patented by Benz in 1886, which Bertha Benz used to make the first long-distance journey in the history of motor vehicles – from Mannheim to Pforzheim – already operated according to this clutch concept.

The disadvantages of a belt drive, such as low efficiency, high susceptibility to wear and inadequate running characteristics especially under rainy conditions, on one hand and the necessity of variable-speed transmissions for the gradually increasing engine outputs on the other hand, induced engineers to seek better alternatives to transmission clutches.

The results were a wide variety of clutch types – including the forerunners of our present-day clutches – all based on the principle of the friction clutch. Here the disc is located on the end of the crankshaft and is joined by a second, stationary disc. When the two make contact, friction is produced and the secondary disc is set in motion. As the clamping load is increased, the driving disc carries along the driven disc with increasing speed until power transmission is reached, and both discs have the same rotational speed. In the period up to full engagement, the main driving energy is converted into heat as the discs slide across one another. This arrangement meets the two chief demands – on the one hand gradual and gentle engagement, so that, when driving off, the engine is not cut off and does not jerk with the drive train, and on the other hand loss-free power transmission with the clutch engaged.

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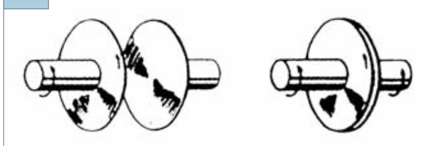

The basic principle of the friction clutch: The driven disc is pressed onto the driving disc until the frictional connection is made

The clutch is actuated via the foot pedal, which pulls back the cone carrier via a release lever against the spring force and thus disengages the clutch.

The basic form of this design principle was already used in 1889, in the steel wheel cars from Daimler, which had a cone/bevel friction clutch. Here a freely moving frictional cone located on the engine shaft and firmly connected to the clutch shaft via the clutch housing engages in the conically machined out flywheel. A spring presses the cone into the flywheel recess so that pressure on the foot pedal will pull the cone back against the spring pressure via the freely movable clutch release sleeve, thereby interrupting the power transmission. Camel hair belts originally functioned as friction linings on the cone surface, but were soon replaced by leather belts. The latter were soaked in castor oil as a protective measure against moisture, grease and oil.

The advantages – self-adjusting, no strain on the drive or transmission shaft – were, however, out-weighed by the disadvantages: on the one hand, the friction lining wore out fast and replacement was complicated; hence one switched to designs with spring-loaded pins or leaf springs under the leather lining. Secondly, the flywheel and clutch cone were very large, so that, owing to its high moment of inertia, the clutch part came to rest much more slowly than was required after the release for gear changing (transmissions were not yet synchronised). To remedy this problem, around 1910 an additional clutch brake or transmission brake was installed which had to be actuated via a second foot pedal – usually in conjunction with the clutch pedal and located together with the latter on a common pedal shaft.

The habit of many drivers of allowing the clutch to slip instead of changing gears when regulating the vehicle speed, heated the flywheel more than it did the friction cone, which was thermally insulated by the leather lining. After a spell of rugged driving, the cone could engage more deeply in the flywheel as it had been expanded by the heat – leaving it jammed tight when it cooled down.

By the end of the First World War, metallic friction linings were becoming increasingly popular. Previously, one had experimented with other solutions: For example, the "Neue Automobil-Gesellschaft (NAG)" constructed a clutch containing a camel-hair

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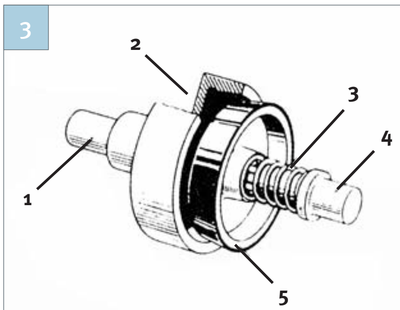

1 = From engine
2 = Engine flywheel with hollow cone
3 = Coil spring
4 = To transmission
5 = Driver cone with leather lining

Design of the cone/bevel friction clutch dominant throughout the 1920s

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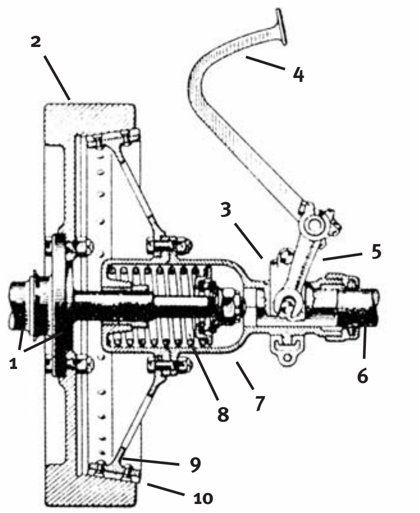

1 = Crankshaft flange
2 = Flywheel
3 = Release sleeve
4 = Clutch pedal
5 = Release lever
6 = Clutch shaft
7 = Clutch housing
8 = Clutch spring
9 = Clutch cone
10 = Clutch lining

Cross section of a cone clutch showing the typical components: clutch cone and correspondingly turned-out flywheel lined cone, stamped from sheet metal and equipped with fan-like blades for cooling, which engaged in a two-part, leather-lined ring screwed into the flywheel. The two-part construction allowed the ring to be easily removed, simplifying maintenance and reducing the frequency of jamming.

The Daimler engine corporation developed an open friction clutch with a bare aluminium cone. For a soft release, oil had to be dripped onto the frictional surfaces at regular intervals.

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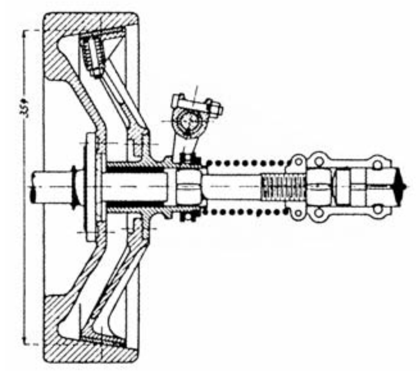

Cone clutch with spring-compressed leather lining

Cone clutches continued to dominate throughout the 1920s thanks to their simplicity. Metallic clutches with cylindrical friction surfaces did not win acceptance because of their poor operational characteristics. Only the spring band clutch, a derivative of the cylindrical clutch that had been installed in Mercedes cars by Daimler since the turn of the century, was able to persevere until the First World War thanks to an ingenious design.

In the spring band clutch, a sturdy, spiral-shaped spring band, which received the drum-shaped end of the transmission shaft, was fitted in a recess of the flywheel.

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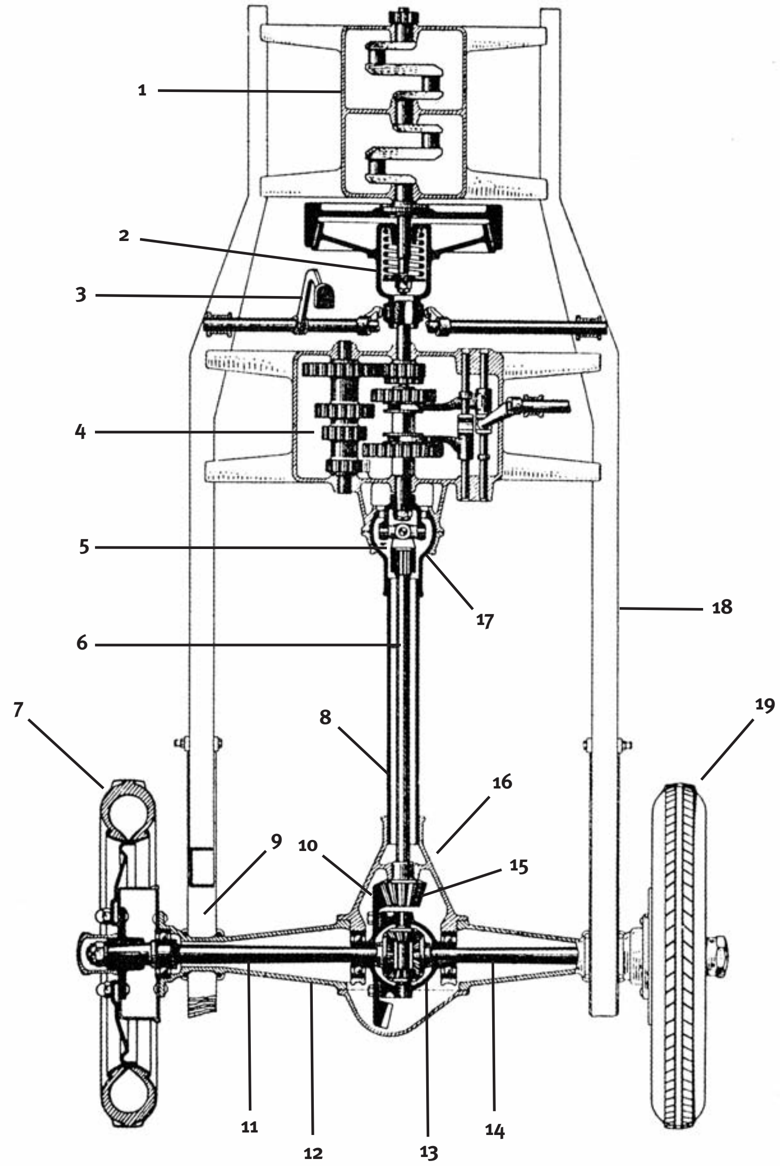
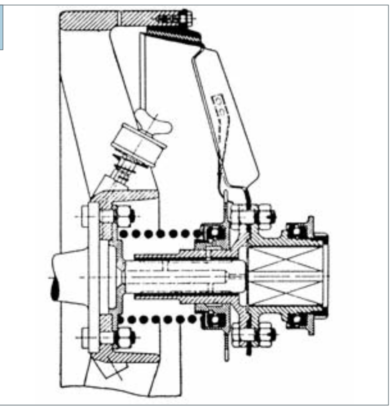


1 = Engine
2 = Clutch
3 = Clutch pedal
4 = Variable-speed transmission
5 = Universal joint
6 = Propshaft
7 = Rear left wheel
8 = Propshaft tube
9 = Spring
10 = Differential ring gear
11 = Left rear axle shaft
12 = Rear-axle housing
13 = Differential
14 = Right rear axle shaft
15 = Differential bevel pinion
16 = Differential housing
17 = Torque ball for absorbing rear-axle thrust and transferring it to chassis frame
18 = Chassis frame
19 = Rear right wheel

View of a chassis with a cone clutch. During disengagement, the clutch brake ensured quick speed reduction of the large mass with a cone clutch

7



NAG clutch with two-part hollow cone ring, which greatlysimplified maintenance

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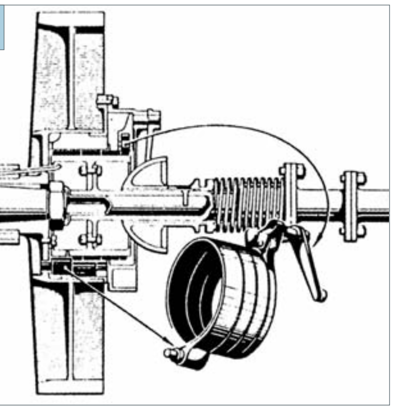

The Daimler spring band clutch, which, owing to its ingeniously simple design, was produced up to the First World War

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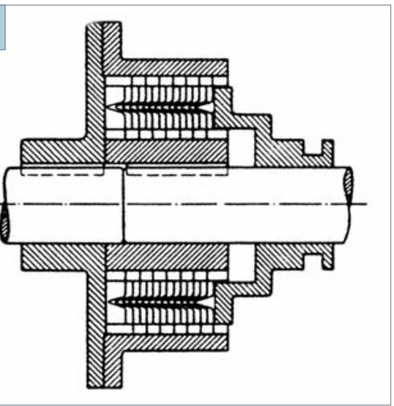

Professor Hele-Shaw from England was the first to experiment with multi-plate clutches

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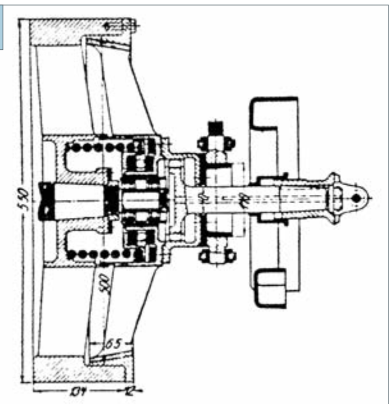

Cone clutch of the Daimler Engine Corporation, with aluminium cone

One end of the coil spring was connected to the flywheel, while the other was fastened to the cover of the spring housing. The actuation of the clutch pedal tensioned the spring band, which then coiled itself (self-reinforcing) more and more firmly around the drum, driving the transmission shaft – and engaging the clutch. The compression of the springs required only slight force and effected a gentle engagement of the clutch.

At about the same time that the Daimler corporation were developing their spring band clutch, Professor Hele-Shaw from England was already experimenting with a multi-plate clutch that can be regarded as the forerunner of today's conventional single-disc dry clutch. Multi-plate clutches, named “Weston clutches” after the first large-scale producer, had a decisive advantage over the cone friction clutch: much greater friction surface area with a lower space requirement and constant engagement.

In the case of the multi-plate clutch, the flywheel is connected to a drum-shaped housing that has grooves on the inside corresponding to the shape of the outer edge of the plate, allowing it to turn with the crankshaft or flywheel and at the same time to move longitudinally. An identical number of discs with matching inner recesses are centred on a hub connected to the clutch shaft. The discs can move longitudinally along the clutch shaft on the hub. During installation, inner and outer clutch plates are alternately combined to form a plate packet, so that a driving and a driven disc always follow one another.

The plate pairs formed in this fashion, originally with a bronze disc always turning against a steel one, were pressed together by a pressure plate under the force of a clutch spring. In this way, all clutch plates were constantly engaged.

This gradual increase of frictional effect enabled the multi-plate clutch to engage very gently.

As the spring pressure eased off, the plates disengaged again, in part supported by the spring-loaded strips bent out from the plane of the plate. By varying the number of plate pairs, a basic clutch type could be adjusted to eachengine output.

Multi-plate clutches operated either immersed in oil/petroleum or dry, in which case, however, special, riveted friction linings were used.

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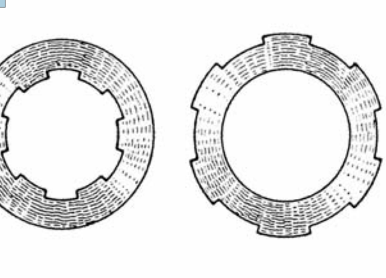

Plate pair from a multi-plate clutch: left, the inner clutch plate, right the outer plate

The greatest drawback of the multi-plate clutch was certainly the drag effect, especially in the oil bath, causing only partial disengagement, and thus making gear changing difficult.

By 1904, De Dion & Bouton had introduced the single-plate clutch principle, which because of the initially inadequate materials only came into wide spread use in the US during the 1920s – largely on demand from the supply industry, who towards the end of that decade granted licences to European manufacturers. Within a few years, the single-plate had superseded cone and multi-plate clutches.

While De Dion & Bouton still lubricated the friction surfaces of their multi-plate clutches with graphite, clutch technology greatly advanced with the advent of Ferodo-asbestos linings, which were used from about 1920 to the present day, when they were replaced by asbestos-free linings. The advantages of the single-plate dry clutch were clear: the low mass of the clutch plate allowed it to come to rest more quickly when released, making shifting much easier – farewell to transmission brakes.

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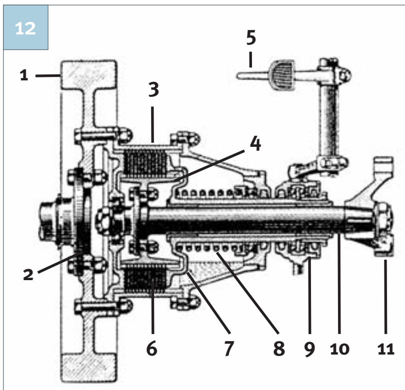
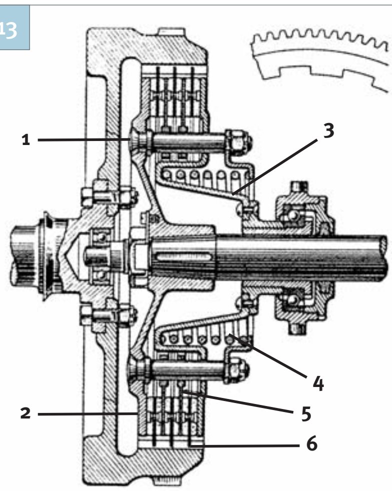


1 = Flywheel
2 = Crankshaft flange
3 = Clutch hub
4 = Clutch housing
5 = Clutch pedal
6 = Clutch plates
7 = Pressure plate
8 = Clutch spring
9 = Release sleeve
10 = Clutch shaft
11 = Propshaft flange

Oil-immersed multi-plate clutch

13



1 = Guide pin
2 = Drive disc for clutch shaft
3 = Thrust plate with release sleeve
4 = Clutch spring
5 = Inner clutch plates
6 = Outer clutch plates with clutch lining

Multi-plate dry clutch with riveted lining

The initial design of the single-plate dry clutch was relatively complicated. The clutch housing was flanged onto the flywheel, and the clutch cover screwed into the housing. This cover held lug levers which were pressed inwards by springs and which transmitted pressure from an intermediate disc via the friction plate and hence the power transmission from the flywheel. The friction disc was connected to the connecting or transmission shaft by a driver. The clutch was engaged and disengaged by a slip-ring disc that moved a cone back and forth. The sides of the cone accordingly actuated the spring-pressured lug levers, which stressed or released, i. e. engaged/disengaged, the intermediate disc. As the cone rotated about the slip-ring disc at rest, lubrication was required at regular intervals.

The coil spring clutch, in which the clamping load is produced by coil springs, was able to gain acceptance. At first, experiments were made with centrally arranged springs, but only the version with several smaller coil or clutch springs

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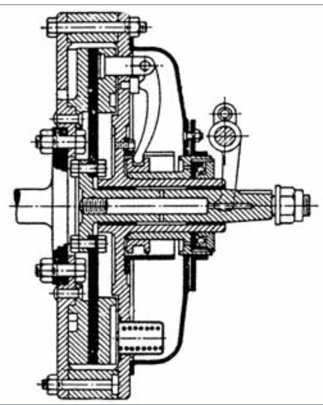

De Dion & Bouton were the first to recognise that single-plate clutches would be the way of the future

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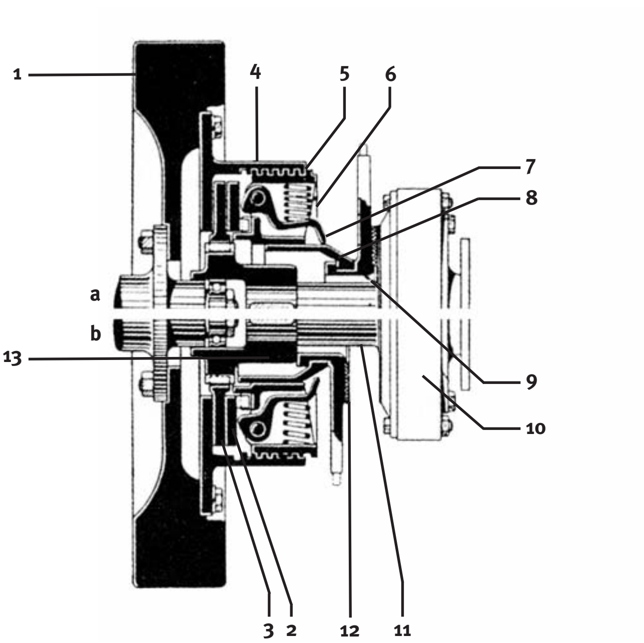
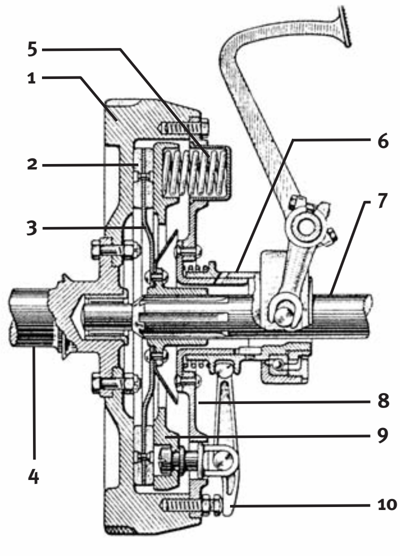


1 = Flywheel
2 = Intermediate disc
3 = Friction disc
4 = Clutch housing
5 = Clutch cover
6 = Spring
7 = Lug lever
8 = Cone
9 = Slip ring disc
10 = Spring joint
11 = Connection shaft
12 = Leather lining
13 = Drive disc

Initial design of the coil spring clutch with clutch springs perpendicular to the central axis

16



1 = Flywheel
2 = Clutch lining
3 = Clutch plate
4 = Crank shaft
5 = Clutch springs
6 = Release sleeve
7 = Clutch shaft
8 = Clutch cover
9 = Pressure plate
10 = Operating levers

This form of coil spring clutch, with the clutch springs arranged parallel to the central axis, predominated through the 1960s

distributed along the outer edge of the clutch housing entered large-scale production. The levers compress the coil springs via a release bearing that moves freely on the clutch shaft, releasing the pressure plate and thus disengaging. The clamping load could be varied by using different spring packages but had the crucial disadvantage that, as the engine speed increased, the coil springs located outside on the pressure plate were pressed further outwards against the spring housings by centrifugal force. The friction arising between the spring and the housing then caused the clamp load characteristics to change. As the engine speed increased, the clutch became progressively heavier. In addition to this, the bearings for the release levers were constantly under strain, making them susceptible to wear, and the spring housings, especially when gear changing at high engine speeds, quickly wore through.

To overcome these systematic drawbacks, the diaphragm spring clutch was developed, created in the research laboratories of General Motors in 1936 and entering volume production in the US in the late 1930s. In Europe, it became especially familiar from the American GMC military trucks used after the Second World War, and starting in the mid-1950s it was used on an individual basis by European manufacturers. The Porsche 356, the Goggomobil, the BMW 700 and DKW Munga were the first German-made vehicles to be so equipped. The clutch entered volume production in 1965 with the Opel Rekord.

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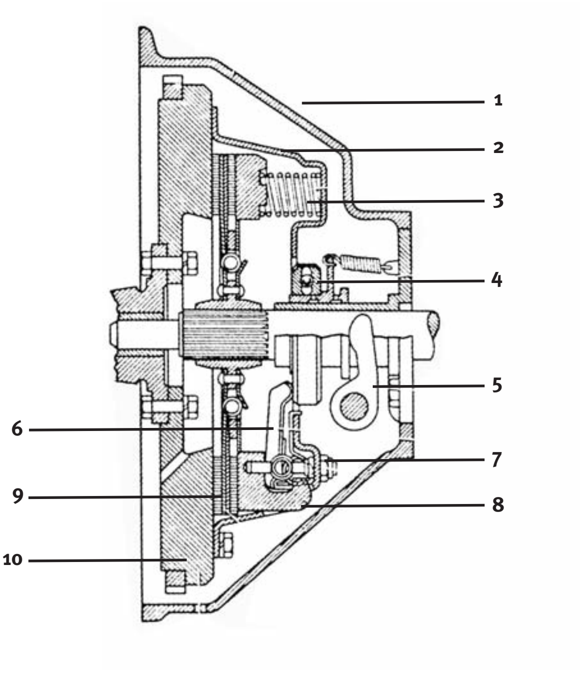

1 = Bell housing
2 = Clutch cover
3 = Clutch compression spring
4 = Release bearing (permanently lubricated)
5 = Clutch fork
6 = Release lever
7 = Adjusting nut
8 = Pressure plate
9 = Driven plate (spring-loaded and damped
10 = Flywheel

In Britain and the US, the Borg & Beck model with springs located under the clutch cover was the most popular ...

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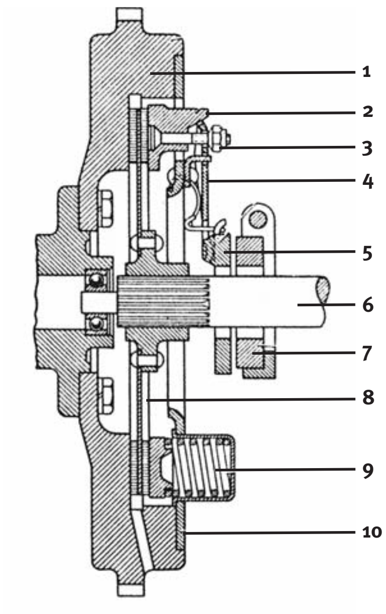


1 = Flywheel
2 = Pressure plate
3 = Adjusting nut
4 = Release lever
5 = Thrust plate
6 = Clutch shaft
7 = Release bearing with graphite ring
8 = Driven plate
9 = Clutch compression spring
10 = Clutch cover

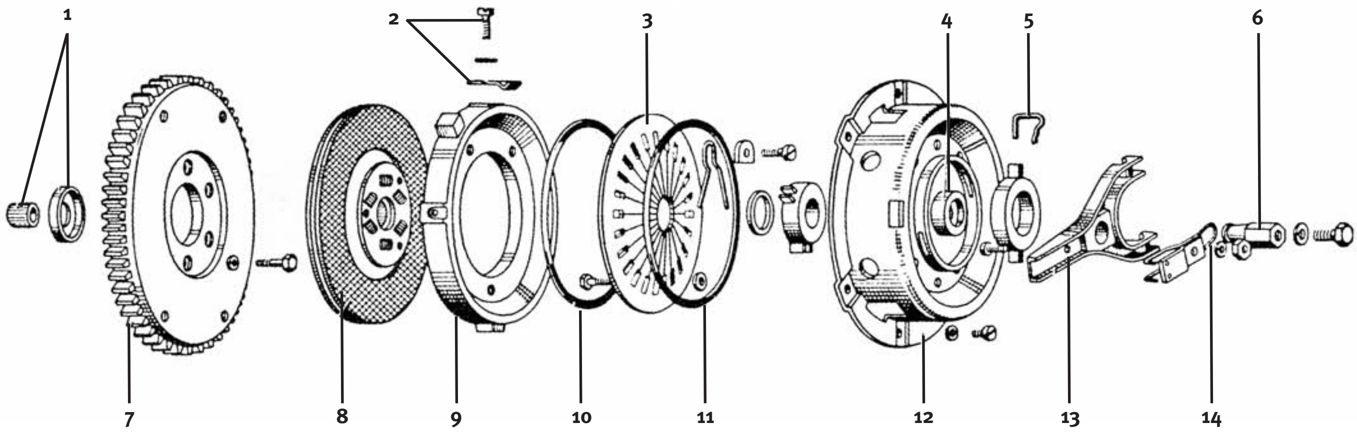
... while on the European continent the version with springs externally located above the clutch cover prevailed

As the diaphragm spring is rotationally symmetric and therefore speed-insensitive, ist hour of triumph occurred in the 1960s, when high-speed engines with overhead camshafts (Glas, BMW, Alfa Romeo) largely superseded the push-rod designs. By the end of the 1960s, nearly all manufacturers had shifted to diaphragm spring clutches. Here LuK played a pivotal role in making the diaphragm spring clutch ready for mass production. The replacement of the complete lever – coil spring system by a diaphragm spring that assumed both functions brought many advantages: Simple mechanical construction, constant clamp loads, less space required for relatively high clamp loads (very important with transversely installed engines) and speed-insensitivity. Thanks to these features, the diaphragm spring clutch is today nearly the only type used, it is also finding increasing applications in utility vehicles – long a domain of coil spring clutches.

Parallel to this development, the clutch plate was optimised. The continually changing speed and fluctuating torque of an internal combustion engine produce vibrations that are transmitted from the crankshaft, clutch and transmission shaft to the transmission. Noise and severe tooth profile wear are the result. Lower flywheel mass and light construction in modern vehicles amplify these effects, so clutch plates were provided with torsion dampers and spring-loaded facings.

While clutch operation for a long time required strong legs, as pedal loads had

19



1 = Spigot bearing
2 = Retaining spring with screw
3 = Diaphragm spring with pre-formed fingers
4 = Release ring
5 = Retaining spring
6 = Ball pin for clutch fork
7 = Flywheel
8 = Drive disc
9 = Pressure plate
10 = Inner fulcrum ring
11 = Outer fulcrum ring
12 = Clutch cover
13 = Release fork
14 = Return spring of release fork

With the multi-plate clutch developed by Chevrolet, also known as the Chevrolet or Inboard clutch, the coil springs were replaced by a diaphragm spring

to be transmitted via the linkage and shafts, comfort was improved in the 1930s with the use of control cables, and in the 1950s with the use of hydraulic actuation.

Easy operation was also promoted by various attempts to automate the clutch process: in 1918 Wolseley had the first idea of an electro-magnetic clutch. In the early 1930s the French firm Cotal built a pre-selector gear box with an electromagnetic clutch, which was used in luxury cars. Best known were the centrifugal clutch, which regulated its clamping load by the centrifugal force, and automatic clutches such as Saxomat (Fichtel & Sachs), LuKomat (LuK), Manumatik (Borg & Beck) and Ferlec (Ferodo).

None of these was able to prevail; the competition from manual and automatic transmissions with torque converters was too great.

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Conventional Clutch

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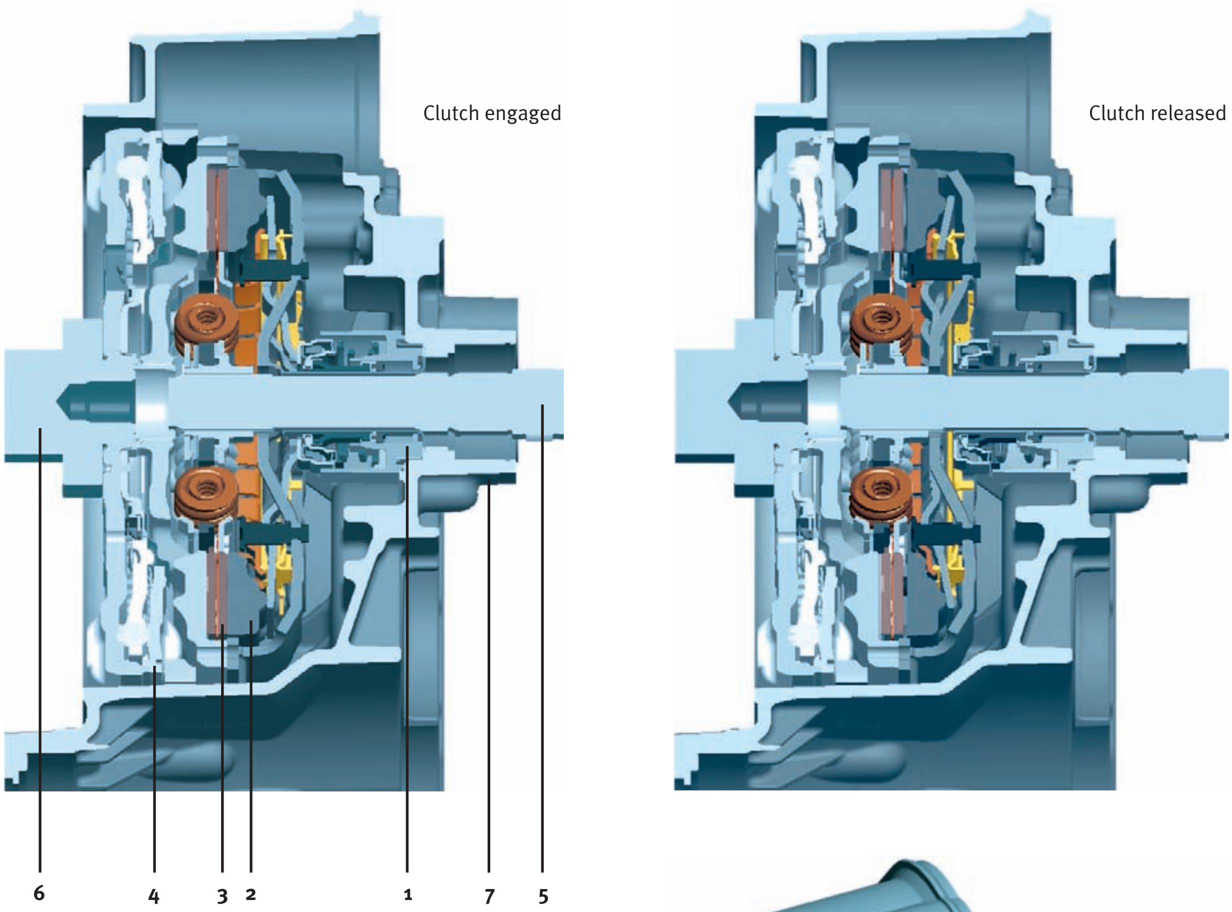

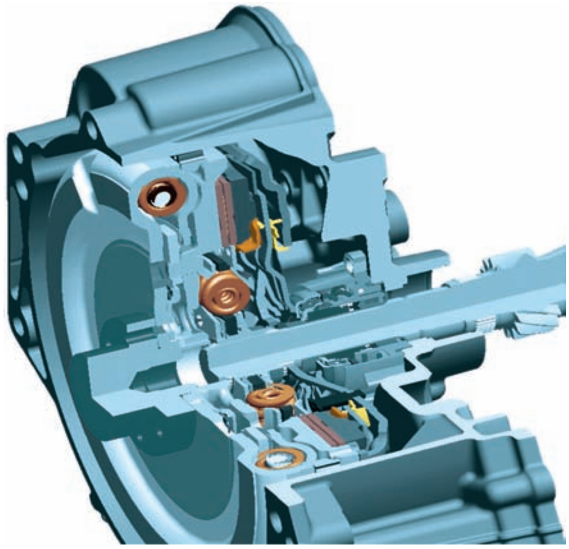

1 = Hydraulic CSC
2 = Self-adjusting clutch
3 = Damped clutch disc
4 = Dual mass flywheel
5 = Transmission input shaft
6 = Crankshaft stub
7 = Gearbox housing

Internal combustion engines only provide useful output over a certain speed range. To be able to use this range for various driving conditions, vehicles must have a gearbox. Today, the transmission is generally connected to the engine via a “single-disc dry clutch”. Twin-disc dry clutches are only found in special cases, such as sports cars and heavy-duty commercial vehicles. Unlike “dry” clutches (i.e. clutches operating in air as the medium), wet clutches operate immersed in oil or oil mist. They are typically used as multi-disc clutches in automatic transmissions, building machinery, special vehicles and in most motorcycles.

The LuK clutch course outlines the basics of state-of-the-art clutch technology and designs

- Diaphragm spring clutches, as displayed in figure 20, are also being increasingly used in commercial vehicles. They have the following advantages over previously used clutch designs:
  - less overall height
  - speed resistance
  - low release loads
  - increased service life

Figure 20 shows a typical clutch installation layout and highlights its basic use as a connection/separation element between engine and transmission.

Besides the main function of connecting or separating the crankshaft and the transmission input shaft, a modern clutch performs a number of other important tasks.

It must:

- enable gentle and jerk-free starting
- ensure fast gear changing of the transmission
- isolate the transmission against engine torsional vibration and thereby decrease noise and wear
- serve as an overload protection for the entire drive train (e.g. in case of shifting errors)
- be durable and easy to replace

The main components of a clutch unit are: the clutch cover assembly consisting of the clutch housing (also clutch cover), the clutch pressure plate as the clutch disc friction partner on the clutch side, the diaphragm spring which generates the clamp load, the tangential leaf spring – a spring-loaded connection between the cover and pressure plate to provide pressure plate lift, the support ring and the spacer for positioning and providing a mounting for the diaphragm spring; the clutch driven plate which consists of the hub, torsion damper with friction device and stop pin, the segment cushion springs and the friction material riveted to them; the flywheel with the pilot bearing (also clutch guide bearing); the release mechanism with guide sleeve, release bearing and release fork.

Operating principle of the clutch

The two diagrams on the left detail the operating principle of a single-disc dry clutch with diaphragm spring. With the clutch engaged (left), the drive from the crankshaft is transmitted via the flywheel to the clutch pressure plate. The positively engaged clutch driven plate transmits the drive via the hub assembly to the transmission input shaft. The diaphragm spring presses the axially variable pressure plate against the driven plate and flywheel. Thus the connection between engine and transmission is made.

Depressing the clutch pedal disconnects the drive between engine and transmission. By actuating the release mechanism (rod link, cables or hydraulic system) the release fork and the release bearing connected to it moves towards the clutch cover assembly and depresses the diaphragm spring fingers. The diaphragm tips act as a lever. As further pressure is applied, the force direction is reversed by the diaphragm spring mounting; the pressure plate is relieved, and with the aid of the leaf springs

The transmittable torque of a single-disc clutch is calculated as follows:

$$M_d = r_m \times n \times \mu \times F_a$$

The abbreviations meaning:
r_m = Mean friction radius
n = Number of clutch facings
μ = Clutch facing friction coefficiente
F_a = Clamp load
M_d = Transmittable torque

Example

Facing inner diameter d_i = 134 mm
Facing outer diameter d_a = 190 mm
Clamp load F = 3500 N

$$d_m = \frac{d_i + d_a}{2} = \frac{134\,\text{mm} + 190\,\text{mm}}{2}$$

$$= 162\,\text{mm mean friction diameter}$$

$$r_m = \frac{d_m}{2} = \frac{162\,\text{mm}}{2} = 81\,\text{mm}$$

$$= 81 \times 10^{-3}\,\text{m mean friction radius}$$

Friction coefficient μ =
0,27–0,32 (for organic facings)
0,36–0,40 (for inorganic facings)

$M_d = (81 \times 10^{-3}\,\text{m}) \times 2 \times 0{,}27 \times 3500\,\text{N}$
M_d = 153 Nm

The transmittable torque of a clutch must always be greater than the maximum engine torque.

moves away from the driven plate. The clutch disc is now able to rotate freely – engine and transmission are separated.

The cushion spring can be seen on the cut-away diagram (centre) of the disengaged clutch (simplified schematic). By steady pressure build-up it ensures a smooth and gentle clutch engagement.

Although not a requirement for the basic operational function of a clutch, the importance of the torsion damper cannot be understated. Through a combination of springs and friction elements specially matched to each engine application it can dampen out the rotational irregularities of the crankshaft, thus reducing noise levels, and avoiding premature wear on transmission assemblies.

The pilot bearing serves as a guide and support for the transmission input shaft. The guide sleeve guides the release bearing centrally to the clutch.

The purpose of the shaft seals at the engine and transmission is to keep the clutch housing free of oil. Even the slightest amount of grease or oil on the clutch facings can considerably decrease the friction coefficient.

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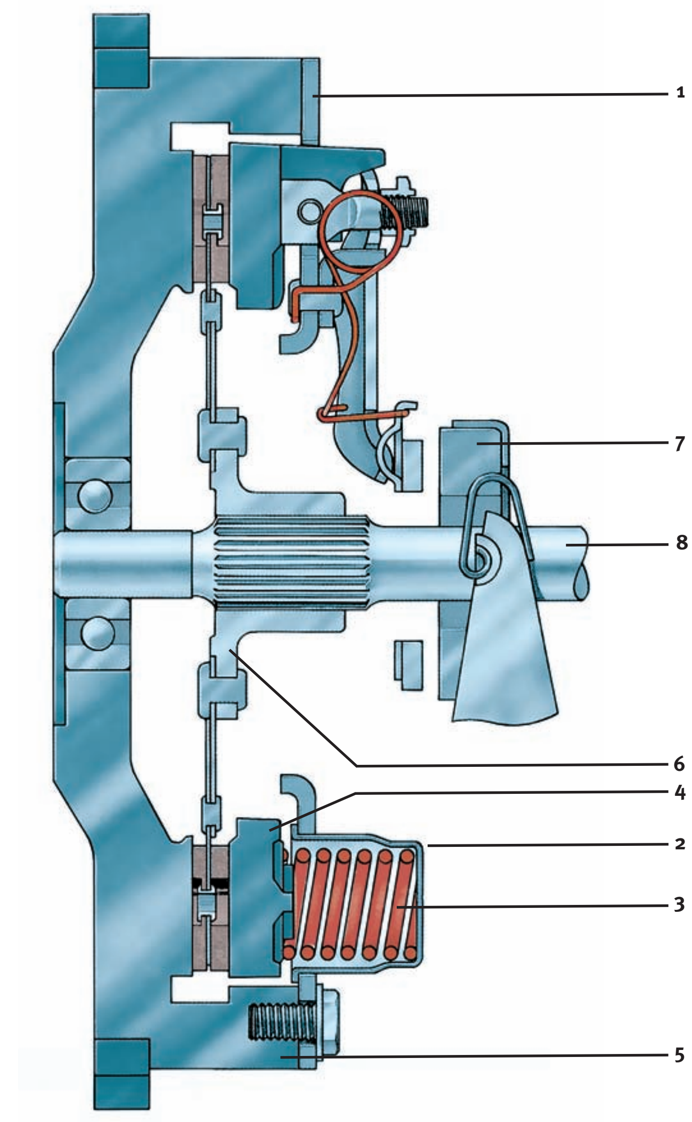


Coil spring clutch

Coil spring clutch

For the purpose of showing a complete picture, we have included the design of a coil spring clutch. The clutch housing (1) contains metal housings (2) for retaining the coil springs (3). These springs press the pressure plate (4) towards the flywheel (5) and thereby clamp the clutch plate (6). The torque can thus be transmitted via the flywheel (5), the clutch housing (1) and pressure plate (4) to the clutch plate (6), located on the transmission input shaft (8). Where as in the case of a diaphragm spring clutch the clamping element and lever form a single part, the coil spring clutch requires a release lever as well as clamp load springs. The pressure plate is moved through the entire lifting stroke against the increasing spring pressure. This is responsible for the comparatively higher actuating force in a coil spring clutch for the same clamp load. Further disadvantages are the relatively low speed-resistance as well as the greater space requirement for coil spring clutches.

The clutch disc

Function

The clutch disc is the friction partner of the flywheel and clutch pressure plate and as such transfers the engine torque to the transmission input shaft.

Friction linings are used to synchronize the engine and gearbox speeds and to transfer the engine torque. The materials used must not only fulfil high technical requirements, i. e. be hard-wearing and provide a constant friction coefficient and smooth torque build-up, but also comply with current environmental standards. LuK clutch linings are partly developed and produced by the company itself and conform to both the technical and legal requirements.

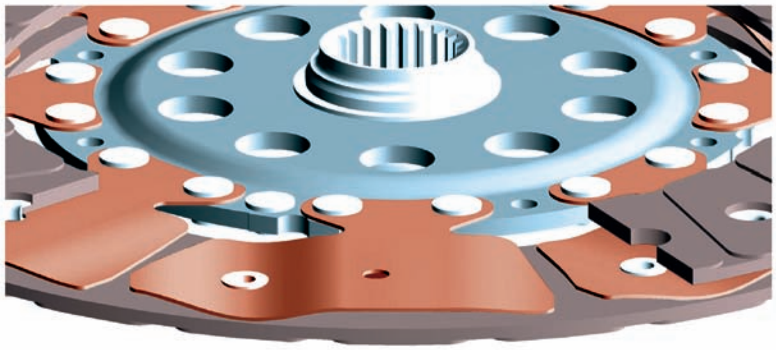
Every LuK clutch disc features a lining resilience system that is optimized to the particular application and enables smooth torque build-up when moving away and an ergonomically synchronised pedal force curve.

On some applications with higher system requirements a special version with multiple-wave double segments is used instead of the standard version with individual segments. An effective lining resilience system produces an even contact pattern, thereby reducing the running in and sagging under temperature and minimizing changes in the cushion deflection throughout the disc life.

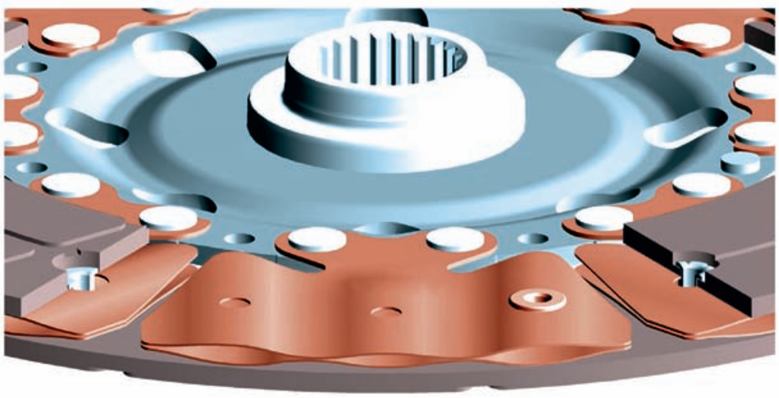
Clutch disc with torsion damper

Torsion dampers are used to reduce the rotational irregularities induced by internal combustion engines, which create resonance in the gearbox and lead to undesirable noise emissions. The torsion damped clutch disc is the ideal solution where use of a dual mass flywheel is impossible be-

22



Single segments



Double segments

In demanding operating environments multiple-corrugated double segments are used instead of conventional single segments

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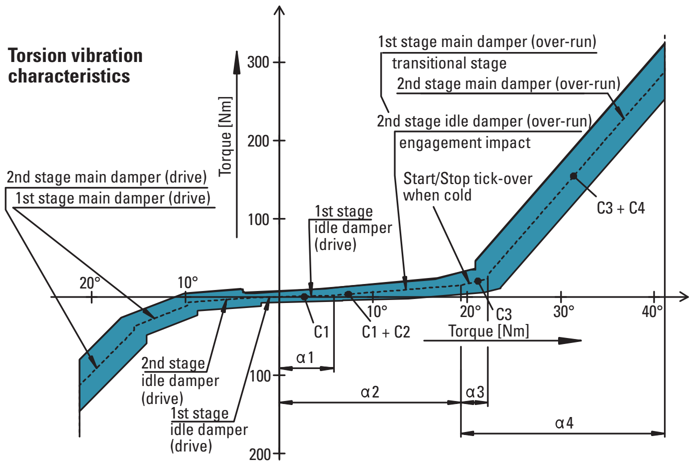
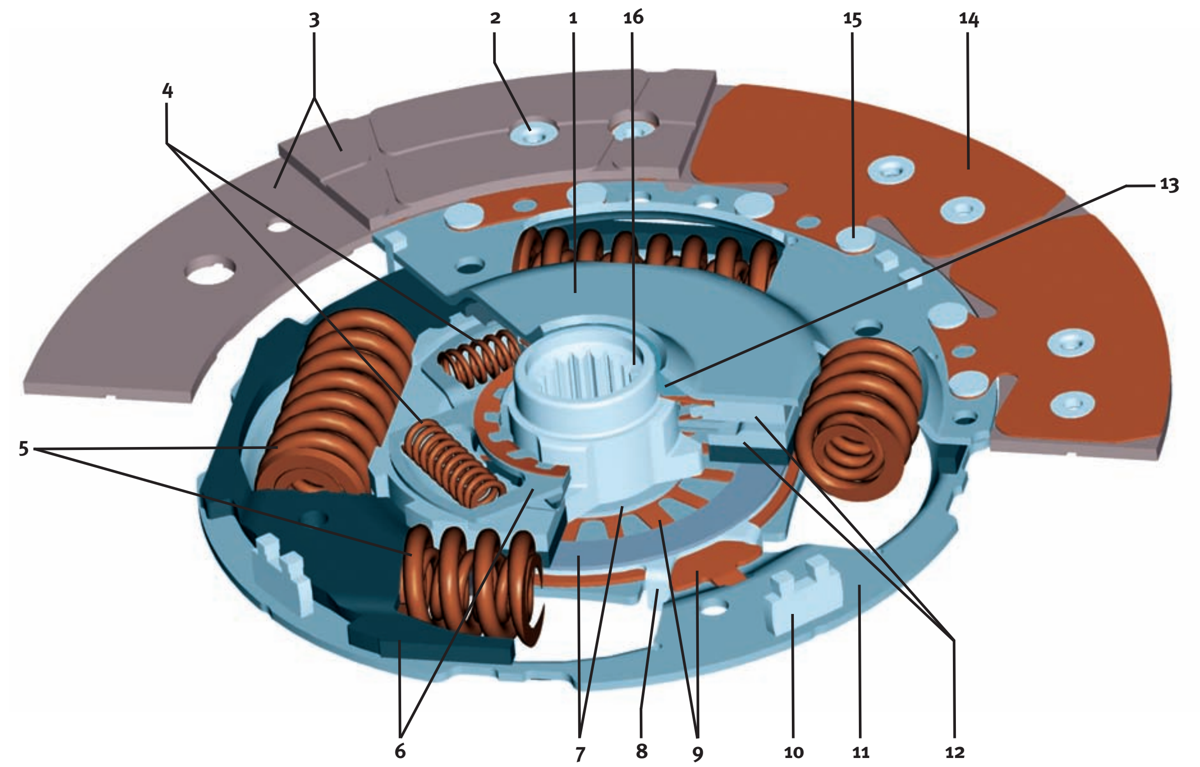


The torsion damper curve is geared to individual customer needs

24



1 = Drive disc
2 = Friction lining rivet
3 = Friction lining
4 = Pressure springs (idle/low-load damper)
5 = Pressure springs (load damper)
6 = Hub flanges
7 = Friction rings
8 = Support disc
9 = Diaphragm springs
10 = Spacer plate
11 = Mating disc
12 = Damper cage assemblies (idle/low-load damper)
13 = Centering taper
14 = Axial spring cushion segment
15 = Spring cushion rivet
16 = Hub

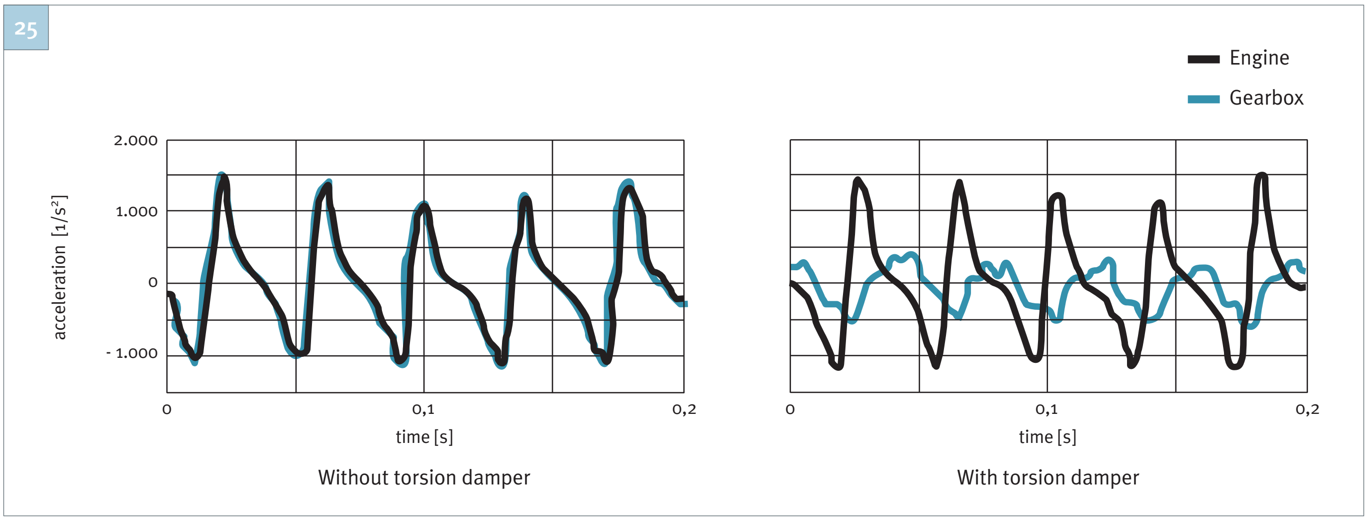
Modern vehicles use sophisticated spring/damper systems

cause of the costs involved or the lack of installation space.

In order to comply with today's requirements despite the demand for weight-optimized drive-train concepts, emission-optimized and more fuel-efficient engines, sophisticated spring/damper systems with integrated friction control elements are required. It is therefore necessary to define a separate torsion damper characteristic for each operating state and load level with specified spring stiffness and friction damping (hysteresis).

The torsion damper characteristic curve can be tailored to individual customer requirements, ranging from multi-stage designs with optimum adjustment of all characteristics and cost-effective solutions with pre-damping during engine idling to applications with a single-stage characteristic.

The cone centering device developed by LuK corrects potential axle displacement between engine and gearbox and guarantees precise functioning of the damper especially designed for idling load conditions (pre-damper). Pre-dampers achieve good vibration isolation efficiency also at low idling speeds and thus help to reduce fuel consumption and emissions.



Torsion damping effect (engine idling) – Cutting-edge measuring techniques are used to identify torsion damper characteristics

Adjustment of the characteristic curve

To define the requisite torsion damper characteristic curve LuK draws on the most advanced measurement and simulation technology available as well as its extensive engineering expertise.

The drive train of the vehicle to be optimized is equipped with sensors which collect and evaluate readings from various vibration conditions. Based on the results a simulation model is created and matched.

After theoretical parameter variations to determine the appropriate characteristic, constructive examination of the part's functionality and trials of prototypes on the vehicle, a torsion damper specifically tailored to the target requirements of the customer becomes available.

Clutch disc for the DMF

If a DMF is used to damp torsional vibration in the power train, the clutch disc has either a single-stage damper or, depending on the application, no torsion damper at all. In the latter case, either a rigid disc or a clutch disc with displacement correction is used.

Engine and gearbox tolerances, especially on transmission intake shafts without pilot bearing, result in a displacement between crankshaft and gearbox. In conjunction with rigid clutch discs, this displacement can cause idling noises and increased profile wearing in critical cases. One remedy to this problem is the displacement correction clutch disc, which enables radial displacement of the hub and thereby prevents potential radial forces in the idling and low load ranges. The efficiency of the pressure springs in the displacement correction clutch disc is limited to the low load range.

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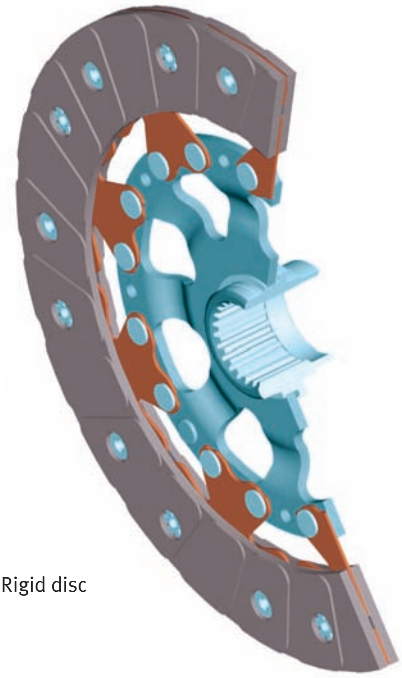
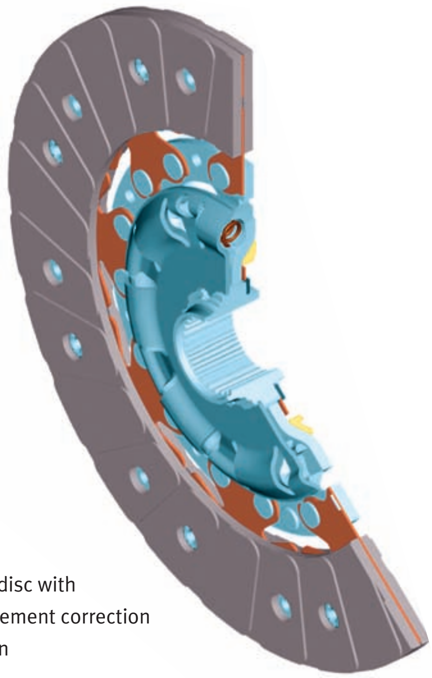
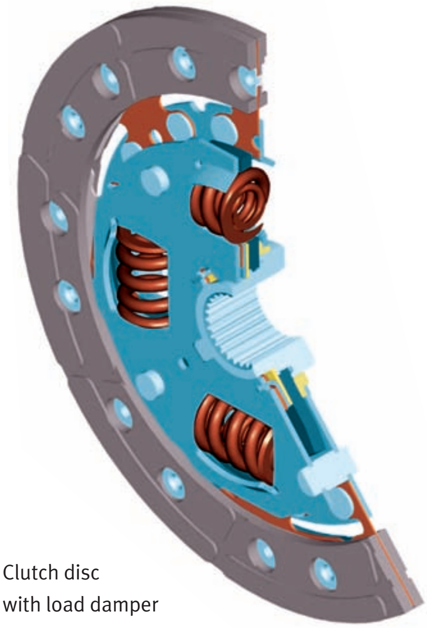


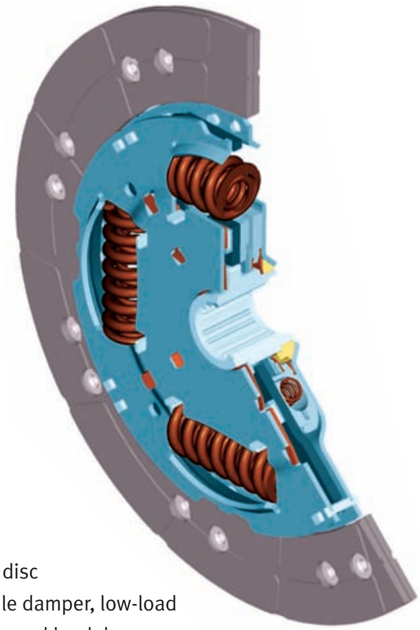
Rigid disc



Clutch disc with
displacement correction
function



Clutch disc
with load damper



Clutch disc
with idle damper, low-load
damper and load damper

Clutch disc designs

Clutch pressure plate

Tasks
The clutch pressure plate forms a frictional system together with the flywheel and the clutch disc. It is secured to the flywheel via the bolts of the clutch cover and transmits the engine torque via the clutch disc to the transmission input shaft. One of the most important components of modern vehicle clutches is the diaphragm spring (3). It has almost completely replaced the conventional coil springs in car clutches.

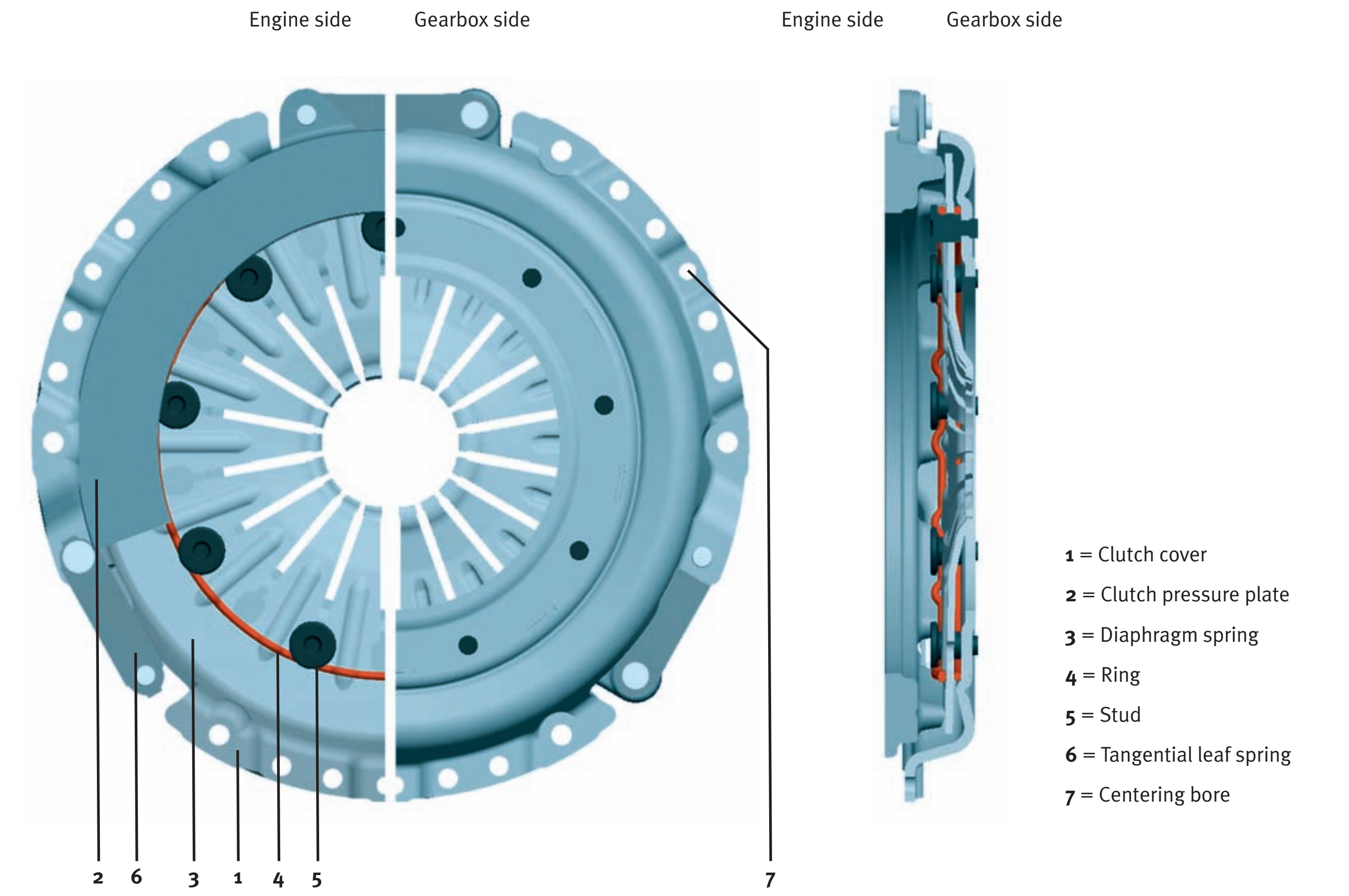
Other important components: The clutch cover (1) serves as a carrier for the diaphragm spring (3), which is supported against the cover by studs (5) and/or rings (4). The diaphragm spring (3) presses the clutch pressure plate (2) against the clutch lining. Tangential leaf springs (6) form an axially variable connection between the housing (1) and the pressure plate (2). Centering bores (7) allow for precise alignment of the clutch cover (1) on the flywheel.

Diaphragm spring
A core component of every designs detailed here is the diaphragm spring. It is flatter and lighter than coil springs. Especially important is the characteristic curve of the diaphragm spring, which differs substantially from the linear curve of a coil spring.

Precise modelling of the diaphragm spring's inner and outer diameters, its thickness, opening angle and material hardness allows a characteristic curve to be produced as represented by the continuous curve in the left-hand diagram in figure 27.

While the clamp load with a coil spring clutch decreases linearly with decreasing facing thickness due to wear, here it initially increases and then drops again. The clutch is designed to begin to slip before the wear limit of the facing is reached. The necessity of a clutch replacement is thus signalled in due time, so that further damage, e.g. by the scoring of the facing

27



Standard diaphragm spring clutch

rivets, is avoided. Moreover, because of the diaphragm spring characteristic curve the requisite pedal forces are less than with coil spring clutches.

Clutch characteristic curves and force diagrams

Figure 28 shows some examples of clutch characteristic curves and force diagrams. They do not directly refer to the designs pictured above them, but apply generally.

The load is specified along the y-axis on the left, while the y-axis on the right gives the lift of the pressure plate; the co-ordinate represents the release travel (and also the release bearing travel in the diagram on the left).

The continuous curve in the diagram on the left represents the development of the clamp load. With a newly installed clutch disc the position of maximum spring load of the diaphragm spring is exerted (operating point: new clutch). As the facing thickness begins to decrease the clamp load of the diaphragm spring first increases to the maximum load level, then gradually drops again to around the load level of the newly installed clutch when the facing is worn to the wear limit.

The clutch disc thickness decreases by about 1.5 to 2.0 mm during its service life. The clamp load is calculated in such a way that the clutch begins to slip shortly before the rivets of the clutch facing score on the pressure plate or the flywheel and would cause additional damage.

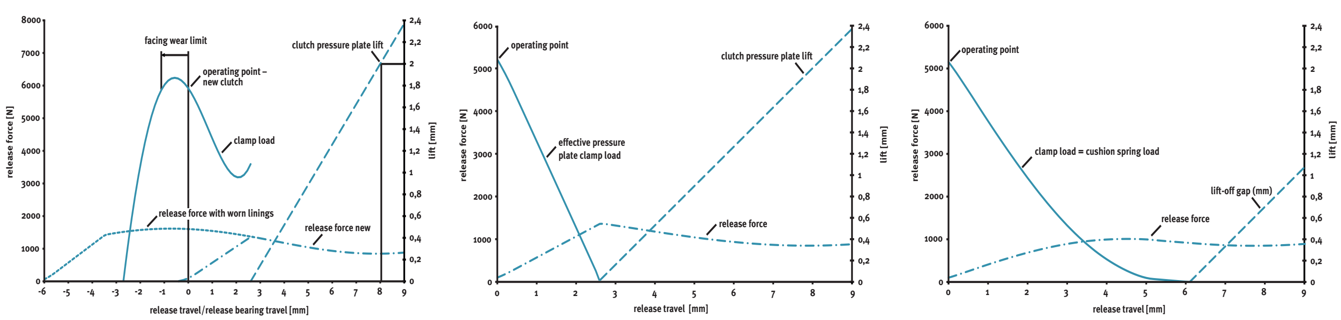
The dashed/dotted line shows the development of the release load, i.e. the load required to actuate the clutch when the clutch is new and – shown by the dotted line – that load after facing wear. The release load initially rises until the operating point is reached, and then slowly drops again. The curve for the release load with facing wear has been moved to the left to illustrate more clearly the ratio of clamp load to release load. The higher clamp load at the operating point with facing wear is reflected by a correspondingly higher release load.

The dashed line shows the development of the pressure plate lift above the release bearing travel. The diagram clearly shows the lever ratio in the clutch: 8mm release travel corresponds to 2mm lift, i.e. to a ratio of 4:1 (not considering the elasticities of the clutch). This also applies to the above mentioned ratio of clamp load to release load.

In the centre and right-hand diagrams, the measurements are contrasted for clutches with and without considering the cushion deflection of the clutch disc. The advantages of cushion springs are gentle clutch engagement and more favourable wear characteristics. Without a lining resilience system, the effective clamp load (solid line) falls linearly and relatively sharply during disengagement. Conversely, it increases just as steeply and suddenly during clutch engagement.

In the diagram on the right, however, we see that the available release travel along which the clamp load diminishes is about twice as large. On the other hand, as the clutch is engaged, the clamp load slowly increases along a curve, as the cushion springs must first be compressed. Thanks to the relatively gentle decline and/or climb of the clamp load curve (solid line);

28



Clutch characteristic curves (example)

Force diagram without cushion deflection (example)

Force diagram with cushion deflection (example) the pronounced release load peak is reduced. As long as the pressure plate (2) still makes contact with the clutch plate, the clamp load and cushion spring load correspond to one another.

Designs
Depending on the design and actuation system of a clutch, one distinguishes between:

- the pushed diaphragm spring clutch and
- the pulled diaphragm spring clutch

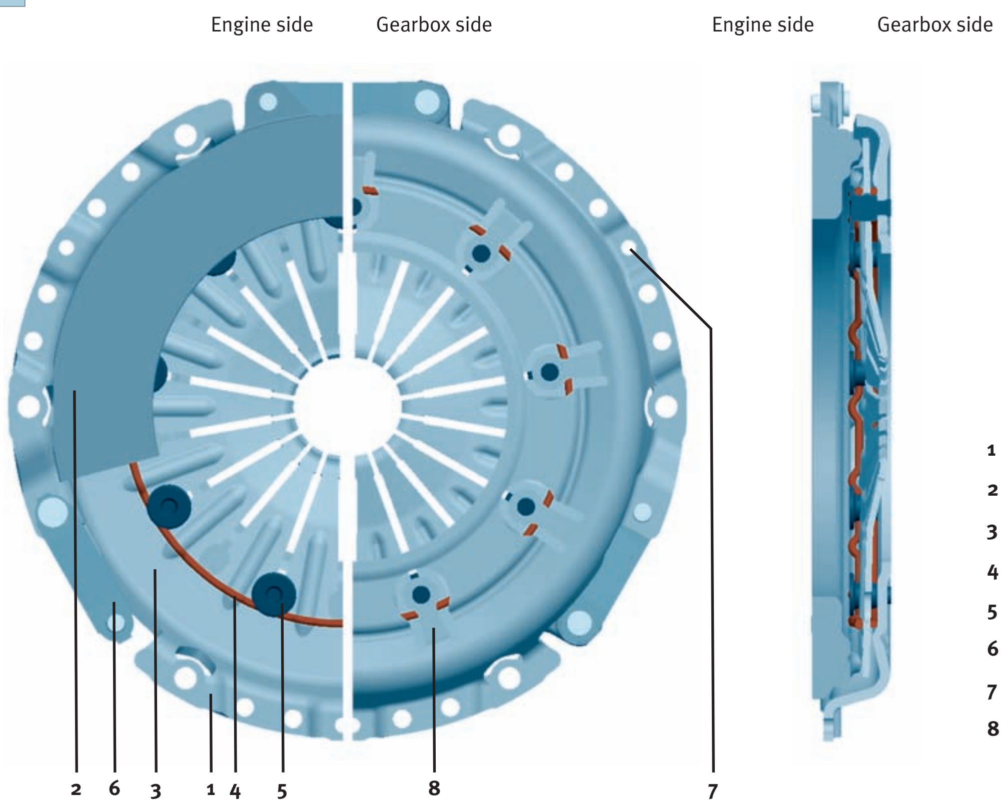
Standard diaphragm clutch
Figure 27 shows the standard version of a diaphragm spring clutch. The clutch cover encloses the diaphragm spring (3) and the pressure plate (2). The pressure plate (2) is linked to the clutch cover (1) via tangential leaf springs (6). They are riveted to the pressure plate (2) at three tabs. Tangential leaf springs (6) perform three basic functions:

- lifting the pressure plate during clutch disengagement
- transmitting the engine torque from the cover to the pressure plate
- centering the pressure plate

The diaphragm spring is clamped between the pressure plate (2) and the clutch cover (1) so as to produce the load required to clamp the clutch disc non-positively between the flywheel and pressure plate (1). In so doing, it is supported by a ring (4) or – optionally a rib – in the clutch cover (1). A ring (4) fixed with studs serves as mating bearing surface. The outer diameter of the diaphragm spring is seated on the pressure plate (2). If the clutch is actuated, the release bearing pushes onto the tips of the diaphragm spring fingers (3). The pressure plate (2) lifts and the clutch disc is disengaged.

Diaphragm spring clutch with keyhole tabs
Figure 29 shows a diaphragm spring clutch with keyhole tabs (8). This design is a further development on the standard diaphragm spring clutch shown in figure 27. The keyhole tabs are modelled in such a way that they pull the studs (5) outwards. As a result, no clearance occurs at the diaphragm spring (3) despite worn diaphragm spring mountings. The advantage of this design is uniform lift throughout the entire clutch life.

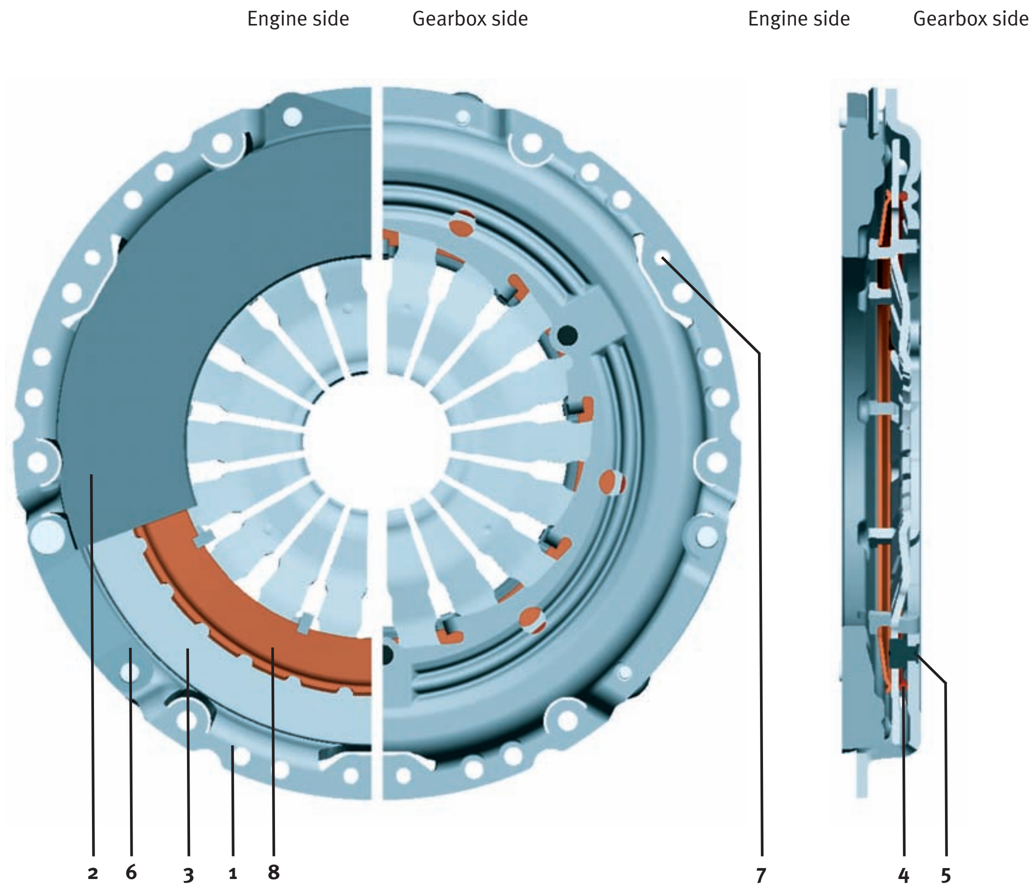
29



1 = Clutch cover
2 = Clutch pressure plate
3 = Diaphragm spring
4 = Ring
5 = Stud
6 = Tangential leaf spring
7 = Centering bore
8 = Keyhole tabs

Diaphragm spring clutch with keyhole tabs

30



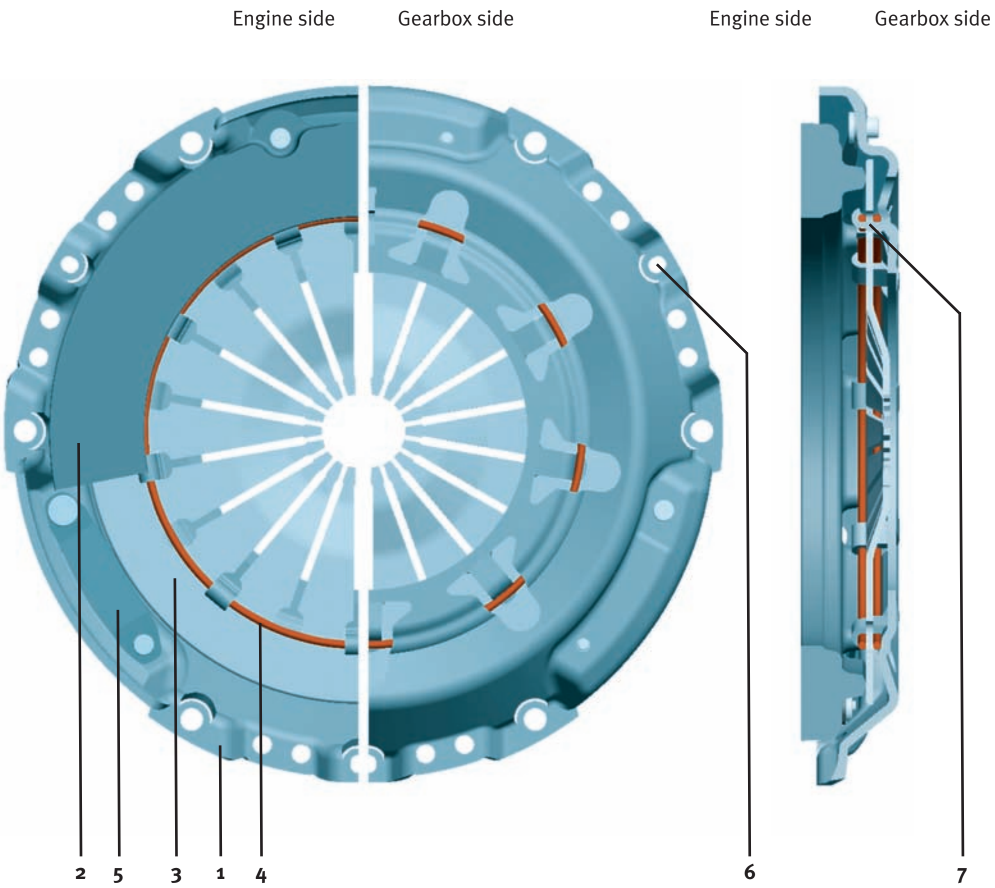
1 = Clutch cover
2 = Clutch pressure plate
3 = Diaphragm spring
4 = Ring
5 = Stud
6 = Tangential leaf spring
7 = Centering bore
8 = Support spring

Diaphragm spring clutch with support spring

Diaphragm spring clutch with support spring

The diaphragm spring clutch with support spring is a special version. The diaphragm spring is supported against the cover (1) by a ring (4), which can optionally be replaced by a rib in the clutch cover. The support spring (8) serves as mating bearing surface. This design allows for a play-free and lossless mounting of the diaphragm spring with automatic wear adjustment. Otherwise this type does not differ from those shown in figure 27 and 29.

31

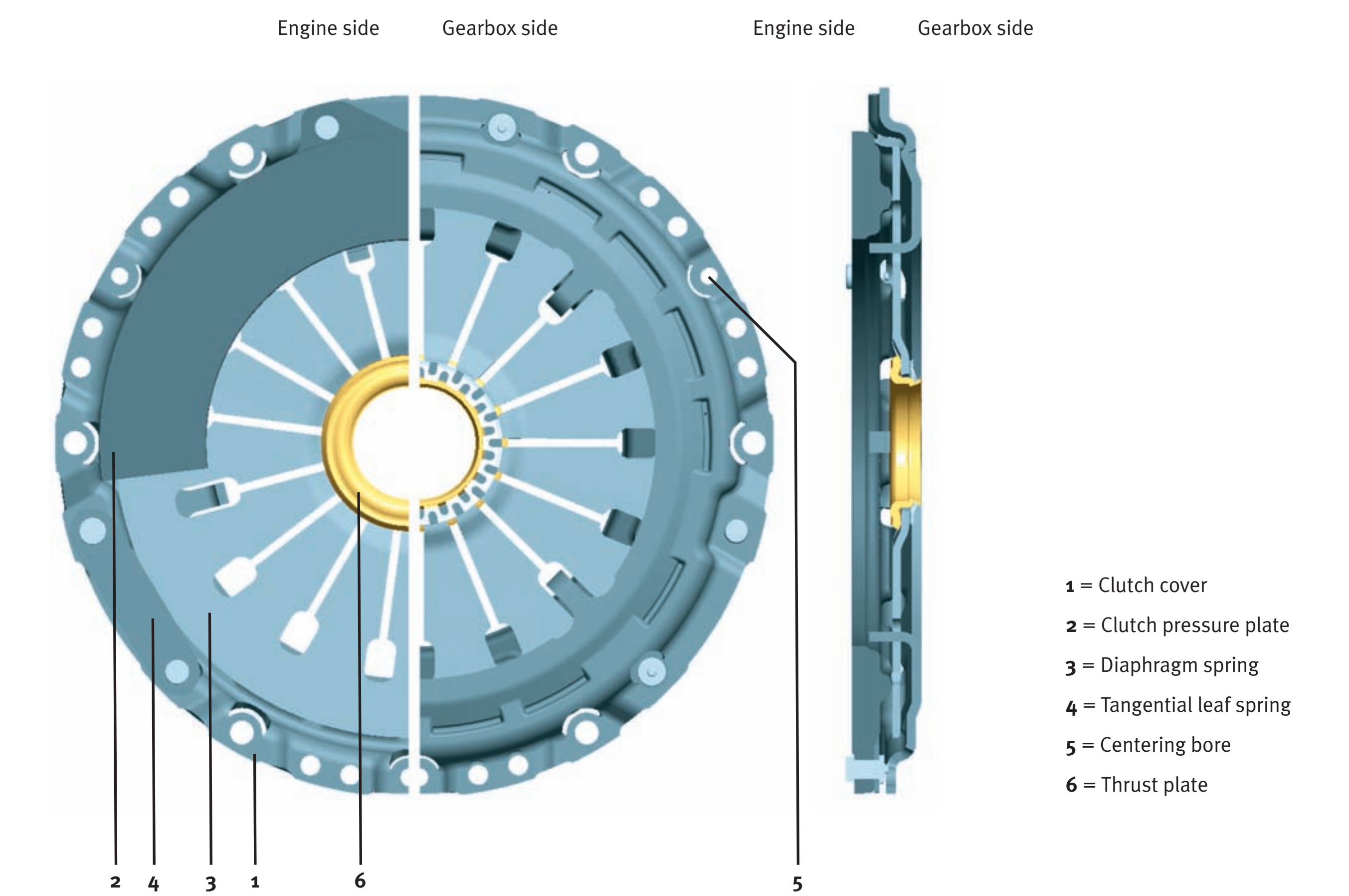


1 = Clutch cover
2 = Clutch pressure plate
3 = Diaphragm spring
4 = Ring
5 = Tangential leaf spring
6 = Centering bore
7 = Tabs

Bolt-free diaphragm spring clutch

Boltless diaphragm spring clutch
Another special type is the boltless diaphragm spring clutch shown in figure 31. Similar to the design with support spring, the diaphragm spring is supported against the cover (1) by a ring (4), which can be optionally replaced by a rib in the clutch cover. As with a pin clutch mechanism, a wire ring serves as mating bearing surface. As a special feature, however, the ring (4) is retained by tabs (7) formed from the clutch cover. By analogy with the keyhole design, here too, the tabs are pre-loaded, so as to provide automatic compensation for the wear occurring at the diaphragm spring mount and to prevent clearance of the diaphragm spring mount throughout the entire clutch service life.

32



Pulled diaphragm spring clutch

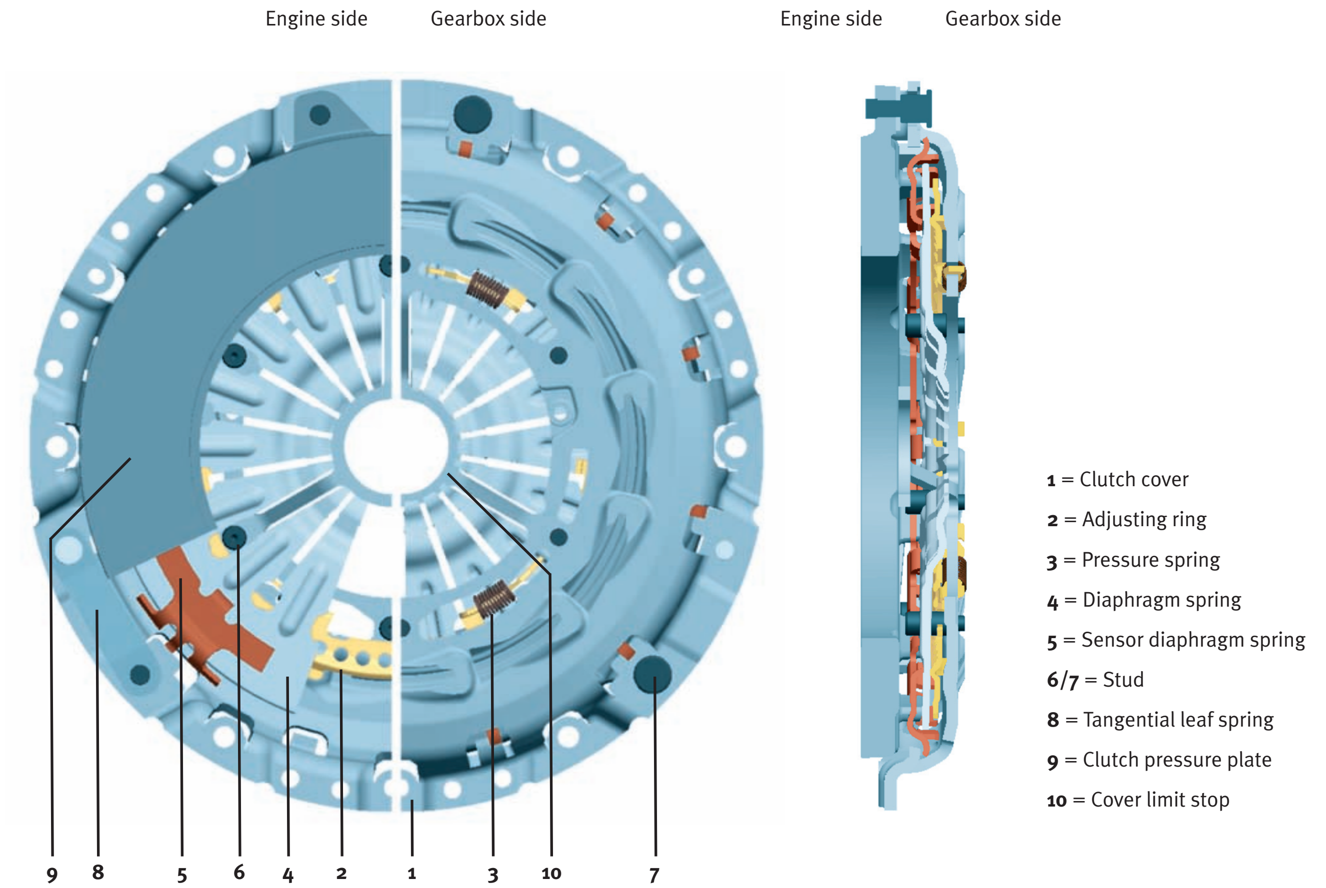
Pulled diaphragm spring clutch

Figure 32 shows the pulled diaphragm spring clutch. Contrary to the pushed type, this design is characterized by the reversed installation of the diaphragm spring. Consequently, clutch actuation is only possible by pushing.

This has the following effect on the clutch design: the outer edge of the diaphragm spring (3) is supported by the clutch cover (1) and the inner edge by the pressure plate.

The benefit of this design is that the leverage ratio yields lower release forces compared with a pushed diaphragm spring clutch while requiring the same clamp load. In addition, the pulled type is more efficient than the pushed type owing to the diaphragm spring being supported at the outer diameter of the clutch cover. One drawback of the pulled clutch is that the gearbox is much more difficult to install and the release bearing requires a much more complex design.

33



Self-adjusting diaphragm spring clutch (SAC)

Self-Adjusting Clutch SAC

In recent years self-adjusting clutches have become the standard equipment in applications with higher engine torque or with increased requirements for wear reserve

The essential advantages of the SAC over conventional designs are:

- Low release loads which remain constant throughout the entire service life
- Excellent driving comfort over the entire service life
- Increased reserve for wear and consequently extended service life thanks to automatic wear adjustment
- Release bearing over-travel is limited by the diaphragm spring end stop

This results in a number of secondary advantages:

- No further need for servo systems (on commercial vehicles)
- Simplified release system layout
- Shorter pedal travel
- Constant pedal forces across the entire engine model range
- New opportunities to reduce the clutch diameter (torque transfer)
- Shorter release bearing travel throug hout the bearing life

Operating principle of the self-adjusting clutch SAC:

Load sensor

On the clutch with wear adjustment the load sensor detects the increased release load due to wear and correctly compensates for the reduction in facing thickness. Figure 35 shows a function scheme of the SAC. Unlike the conventional clutch, the (main) diaphragm spring is supported by a sensor diaphragm spring instead of being riveted to the cover.

In contrast to the strongly regressive main diaphragm spring, the sensor diaphragm spring provides a sufficiently wide range of almost constant load.

The constant load range of the sensor diaphragm spring is designed to be slightly higher than the desired release load. As long as the release load is smaller than the load of the sensor spring when disengaging the clutch, the pivot point of the main diaphragm spring remains stationary. When facing wear increases the release load increases, the counterforce of the sensor spring is overcome and the pivot point moves towards the flywheel to a position where the release load again falls below the sensor load. When the sensor spring deflects, a gap develops between pivot point and cover, which can be compensated for by introducing for example a wedge-shaped component.

Design of a self-adjusting clutch with load sensor

The load sensor with the thickness adjustment wedge can be realized in a simple and elegant manner. In comparison to the conventional clutch, the only additional parts required by this design are a sensor diaphragm spring (red) and a ramp ring (yellow). The sensor diaphragm spring is suspended in the cover. Its inside fingers support the main diaphragm spring. Because of centrifugal forces, the wedges that provide the actual adjustment are positioned in circumferential direction. A steel adjusting ring with ramps moves on opposing ramps in the cover. The steel adjusting ring is preloaded in circumferential direction with pressure springs which force the ring to fill the gap between the diaphragm spring and the cover when the sensor spring deflects.

Figure 34 shows the release load curves for a conventional clutch with new and worn facings. In contrast, compare the significantly lower release load of the SAC, which has a characteristic curve that remains virtually unchanged over its service life.

An additional advantage is the higher reserve for wear, which no longer depends on the length of the diaphragm spring curve (as in conventional clutches), but rather on the ramp height, which can easily be increased to 3 mm for small and up to 10 mm for very large clutches. This represents a decisive step towards the development of clutches with high durability.

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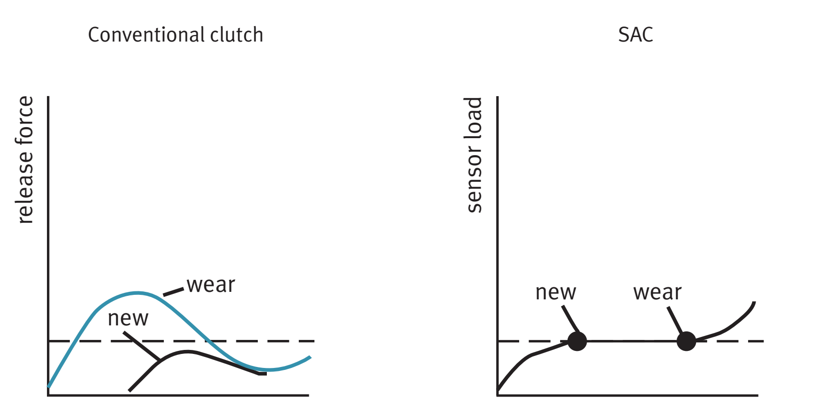
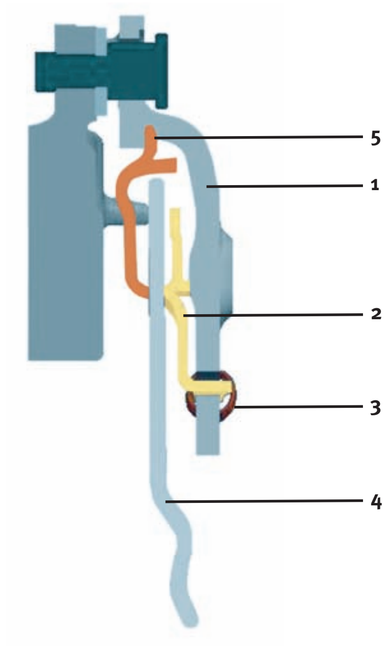


Conventional clutch vs. SAC

35



1 = Clutch cover
2 = Adjusting ring
3 = Pressure spring
4 = Diaphragm spring
5 = Sensor diaphragm spring

Operating principle of the self-adjusting clutch (SAC)

Multiple-disc self-adjusting clutch SAC

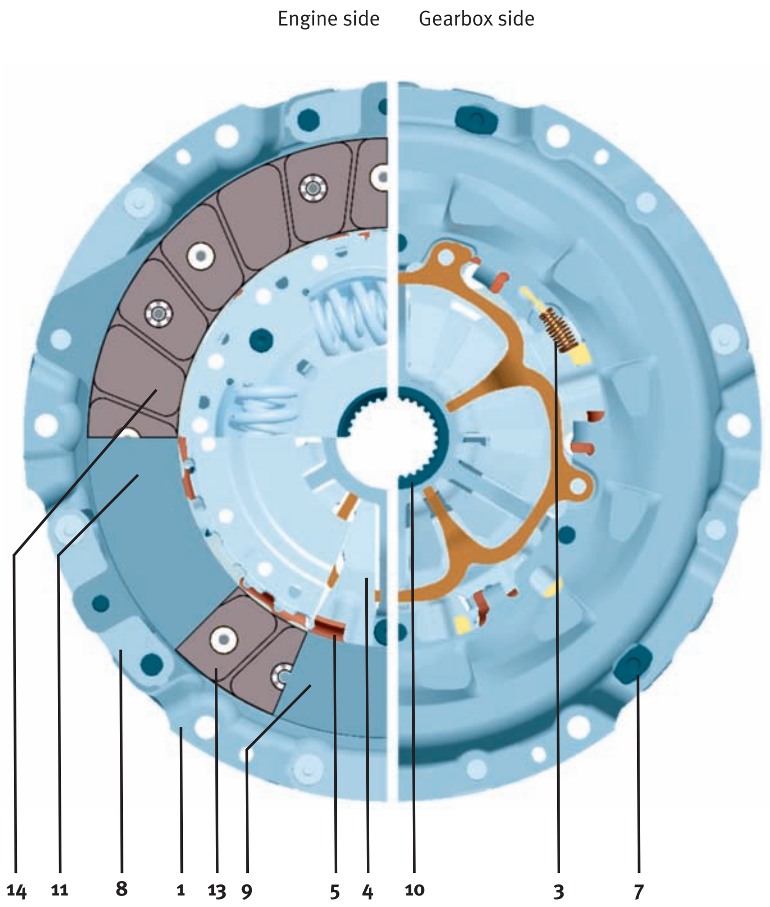
High-performance engines which generate engine torques above 500Nm require clutches which are able to transfer these torques. This involves an almost inevitable increase in pedal forces despite the use of a self-adjusting clutch. A variety of technological approaches kept the increase within reasonable limits (e.g. improved release systems), however calls for a clutch with reduced actuation forces grew louder.

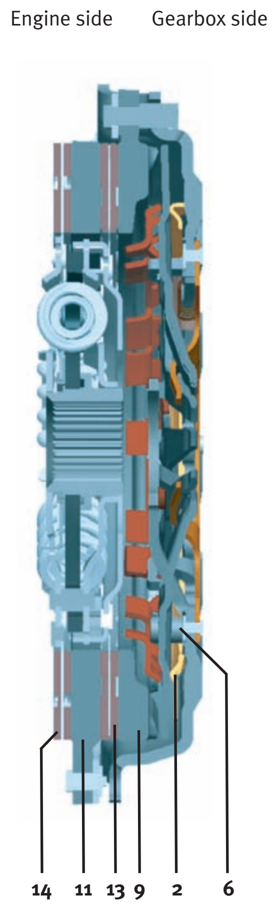
Design of a multiple-disc self-adjusting clutch SAC

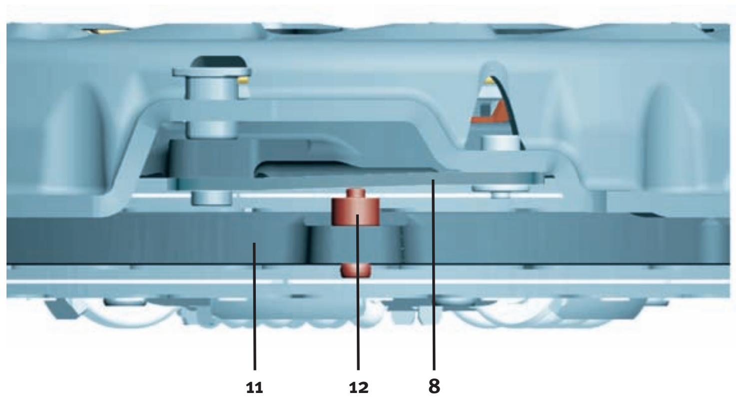
In contrast to the single-disc version, the multiple-disc SAC has an additional intermediate pressure plate and three more tangential leaf spring packages which ensure the lift of the intermediate pressure plate. To realise even wear of both clutch discs lift bolts are used to control the intermediate pressure plate. They make sure that the lift of the intermediate pressure plate is half as big as the lift of the pressure plate. A special version of the clutch disc can be modelled to suit vehicle applications which require a damped clutch disc to provide better insulation.

The benefit of the multiple-disc SAC is that it permits a reduction in release load for the same engine torque or, conversely, an increase in engine torque transfer at the identical release loads level. With engine concepts, where high engine torque is paralleled by high engine speeds, the multiple-disc SAC also offers the option of decreasing the facing outer diameter, which in turn improves the burst speed characteristic of the clutch discs. Furthermore, the downsizing of the clutch disc helps to stabilize or even slightly decrease the disc's mass moment of inertia compared to a single-disc system of corresponding clutch diameter.

36







1 = Clutch cover
2 = Adjusting ring
3 = Pressure spring
4 = Diaphragm spring
5 = Sensor diaphragm spring
6/7 = Stud
8 = Tangential leaf spring
9 = Clutch pressure plate
10 = Cover limit stop
11 = Intermediate pressure plate
12 = Lift bolt
13 = Clutch disc 1
14 = Clutch disc 2

Multiple-disc SAC

The DMF

The increase in noise sources owing to inadequate natural damping is a feature of modern automotive construction. The causes lie in the reduced weight of the vehicles and wind-tunnel optimised bodies, whose low wind noise now makes other noise sources perceptible. In addition, lean concepts, extremely low-revving engines, 5 or 6 speed transmissions and high-velocity oils are a further contributory factor. The periodic combustion processes of IC piston engines induce torsional vibration in the power train, manifesting themselves as gear rattle and body boom and interfering with the comfort requirements of the driver.

The significance of medium-sized cars and compact vehicles with transversally installed engine is growing; the demands for more fuel-efficient and low-emission engines are also on the increase. However this leads to greater rotational irregularities, especially on DI diesel engines. In order to provide these vehicles with the same driving comfort as top-of-the-range models, LuK has developed the dual mass flywheel (DMF).

The DMF neutralises engine vibration already at engine idling; i.e. no more gearbox rattle, nor any unpleasant body boom in a particular engine speed range. With the dual mass flywheel LuK provides vehicle manufacturers with a highly efficient system for damping torsional vibration in the power train.

Design

The division of the conventional flywheel into two discs results in a primary flywheel mass with starter ring gear on the engine side, and a secondary flywheel mass with vent slots for heat transfer, which increases the mass moment of inertia on the transmission side.

The two decoupled masses are connected via a spring/damper system and supported by a deep groove ball bearing or plain bearing so they can rotate against each other. Two moulded sheet metal parts laser-welded to the outer edge form the ring-shaped grease cavity, in which the arc springs with spring guides are located. Sealing is provided by the diaphragm.

Designed as a diaphragm spring, the wings of the flange engage with the arc-shaped pressure springs. The flange can be made of rigid sheet metal, include a damper or be designed as a slip clutch in order to act as peak torque limiter.

An additional friction device, bearing floated on the hub, is carried by one of the retaining plates and is able to generate the required friction at large torsional angles.

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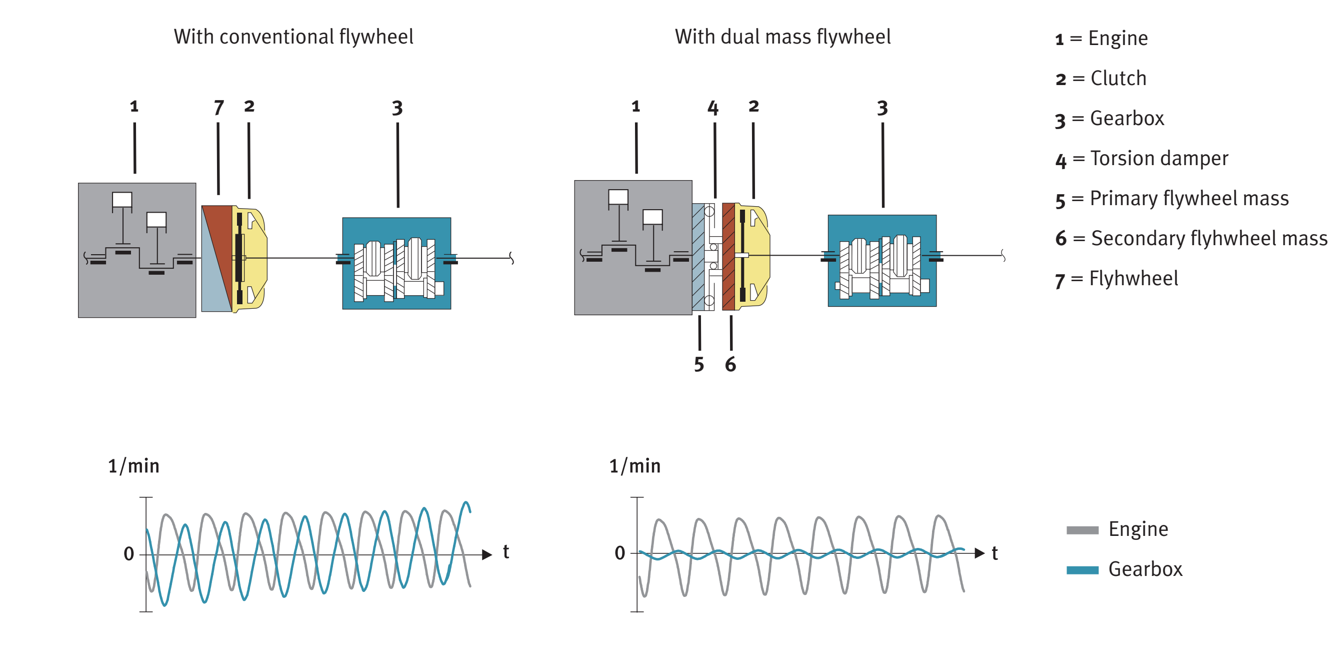
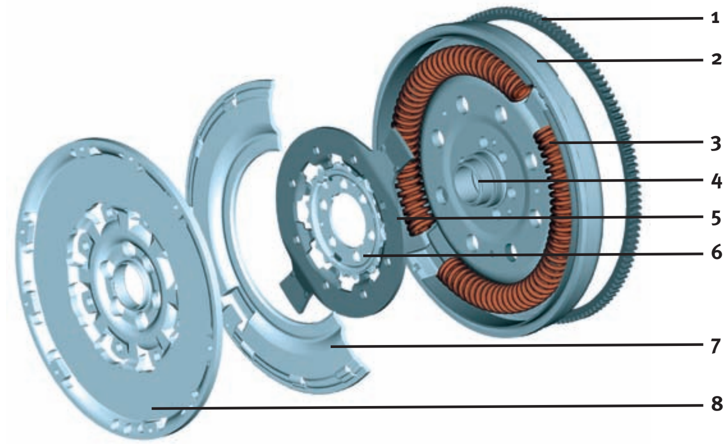


Transmission of torsional vibration

38



1 = Starter ring gear
2 = Primary flywheel mass
3 = Arc spring
4 = Plain bearing
5 = Flange
6 = Floating friction device
7 = Primary cover (cross-sectional view)
8 = Secondary flywheel mass

Standard DMF design

As the DMF has an integral spring/damper system, a rigid clutch disc without torsion damper is frequently used. Usually a diaphragm spring clutch, positioned via centering pins, is used as a clutch pressure plate.

Function

Physical study of drive trains has revealed that the resonance speed range can be shifted by changing the allocation of the mass moments of inertia. As the transmission mass moment of inertia increases, the resonance speed, which generates loud noise, drops below the idle speed and thus falls outside the engines rev range.

Using the dual-mass flywheel (DMF), LuK was able to develop a high-volume product that embodies this principle and thereby keeps resonance amplitude extremely low.

Unlike in conventional layouts, the mass moment of inertia in a DMF is decreased in front of the torsion damper and increased behind it. The engine’s moment of inertia is now assigned to the primary mass of the DMF, while that of the transmission is assigned to the secondary mass including

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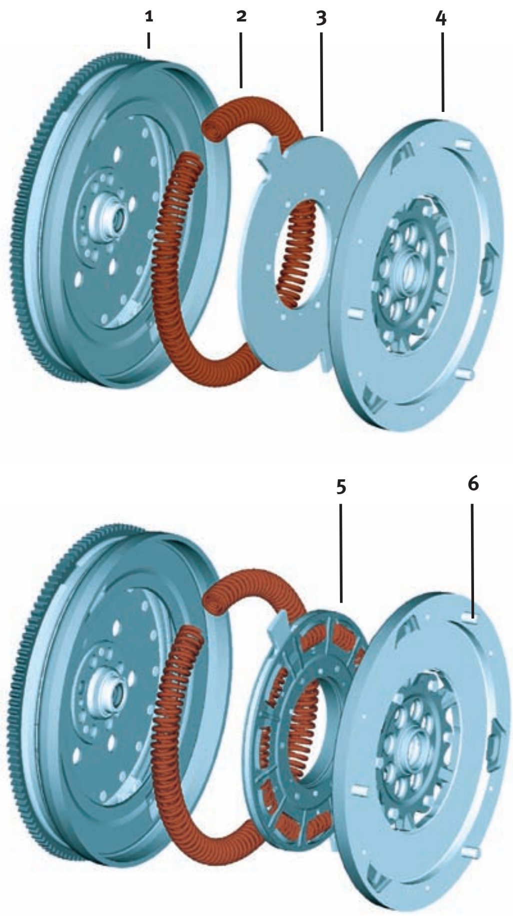

1 = Primary flywheel mass
2 = Arc springs
3 = Flange
4 = Secondary flywheel mass
5 = Flange with inner damper
6 = Centering pin

DMF with different flange designs

the clutch disc and the clutch pressure plate. In this way, the resonance speed is shifted from approx. 1,300 rpm to about 300 rpm and can no longer interfere with driving comfort as the engine is not operated in this speed range.

Until now, i.e. with a conventional flywheel and torsion-damped clutch disc, the torsional vibrations in the idling range were transferred practically unfiltered to the gearbox, causing the gearwheel tooth edges to knock together (gearbox rattle). In contrast, the complex spring/damper design of the DMF filters out torsional vibration caused by the engine. This prevents gearbox components from knocking – rattling does not occur and the driver's demands for high comfort are fully met.

Arc spring

The spring damping system must fulfil two contradictory requirements.

1. Under normal operating conditions, the cyclic irregularities of the engine produce only small working angles in the damper. In this operating range, low spring rates combined with low damping capacity are sufficient to attain optimum vibration damping.

2. Typical load changes (e g. full acceleration), cause load change vibration, which lead to considerable noise. This can only be offset by a torsion damper, which has an extremely low spring rate and, at the same time, high damping capacity.

The arc spring integrated in the DMF resolves this contradiction: at large operating angles it provides high damping capacity at very low spring rates. Yet at normal operating angles it provides low damping capacity at low spring rates, thus providing perfect isolation against vibration.

Aside: the compact DMF

The compact DMF is the ideal solution for the limited installation space in FWD vehicles. The particularly space-saving version of the clutch assembly incorporates the dual mass flywheel, clutch pressure plate and clutch disc.

The module is supplied pre-assembled ready to be mounted on the crankshaft.

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Compact DMF: Pre-assembled unit comprising DMF, clutch disc and clutch pressure plate

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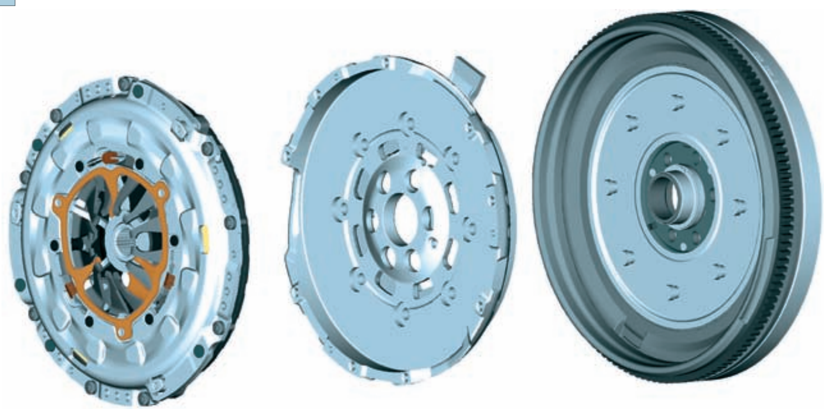

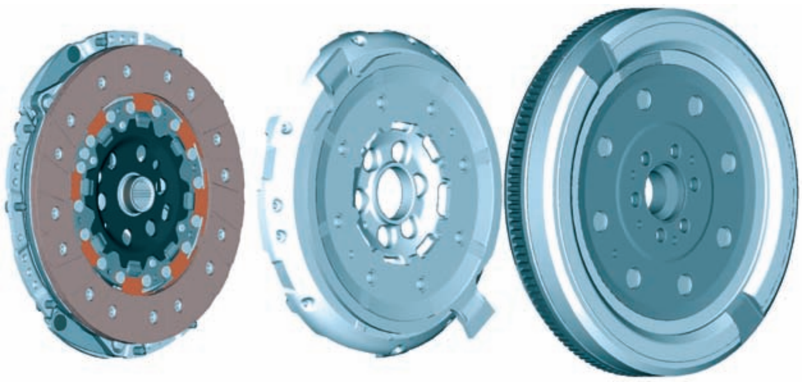

Compact DMF

The crankshaft bolts can be tightened through openings in the diaphragm spring of the clutch pressure plate and the clutch disc. This also simplifies processes for the customer: Instead of matching three individual components and two sets of bolts, there is now only one full-service package.

The benefits of the LuK dual mass flywheel at a glance:

- Excellent driving comfort
- Absorbs vibration
- Neutralizes noise
- Fuel reduction thanks to low engine speeds
- Increased shifting comfort
- Reduced wear of the synchronizing components
- Overload protection of the power train

Environmental benefits:

- Owing to the outstanding noise behaviour in the low-rev range there are fewer shifting actions, the average engine operation speed decreases
- Overall efficiency is improved, leading to lower fuel consumption and pollutant emissions

Hydraulic clutch release systems

Function

On vehicles with foot actuated single disc clutch, a mechanism is required to transfer power from the pedal to the clutch. Developing such a mechanism has brought fourth a number of different solutions. Originally, a cable was used to transmit the pedal forces to a lever mechanism in the clutch bell; the clutch then was actuated by the lever and a release bearing. Today, these systems have only little share of the market, as it is more and more difficult to install the clutch cable in a straight line between the pedal and the lever due to less and less space available in the engine compartment. Installing a cable in narrow radii is impossible as friction and wear rise to impermissible levels and driving comfort and clutch actuation smoothness are impaired. Therefore, modern foot actuated clutch systems use hydraulic clutch actuation mechanisms.

There are two basic systems: In semi-hydraulic systems the clutch cable is replaced by a hydraulic system consisting of a master cylinder at the pedal, a hydraulic line and a slave cylinder on the gearbox outside. On release systems with concentric slave cylinder (CSC) the lever in the transmission housing and the conventional release bearing are replaced by a hydraulic cylinder with integrated bearing which is located inside the clutch housing and positioned centrally on the clutch disc hub and clutch diaphragm spring (figure 42 Clutch release system with CSC). Fully hydraulic release systems are more economic for vehicle manufacturers as they are easier to install owing to the reduced number of individual parts, and the installation of the hydraulic line offers a lot of flexibility in designing engine compartment layouts.

Design and function of the system components

Master cylinder

The master cylinder (figure 43) consists of a housing, a piston with piston rod and a pair of seals (primary and secondary seal). It has a hydraulic port, normally a quick connector, for the pressure line through which it is connected to the slave cylinder. On some models the screw connector commonly used on brake systems can still be found. The master cylinder is connected to the hydraulic fluid circuit. In many cases, it is connect-

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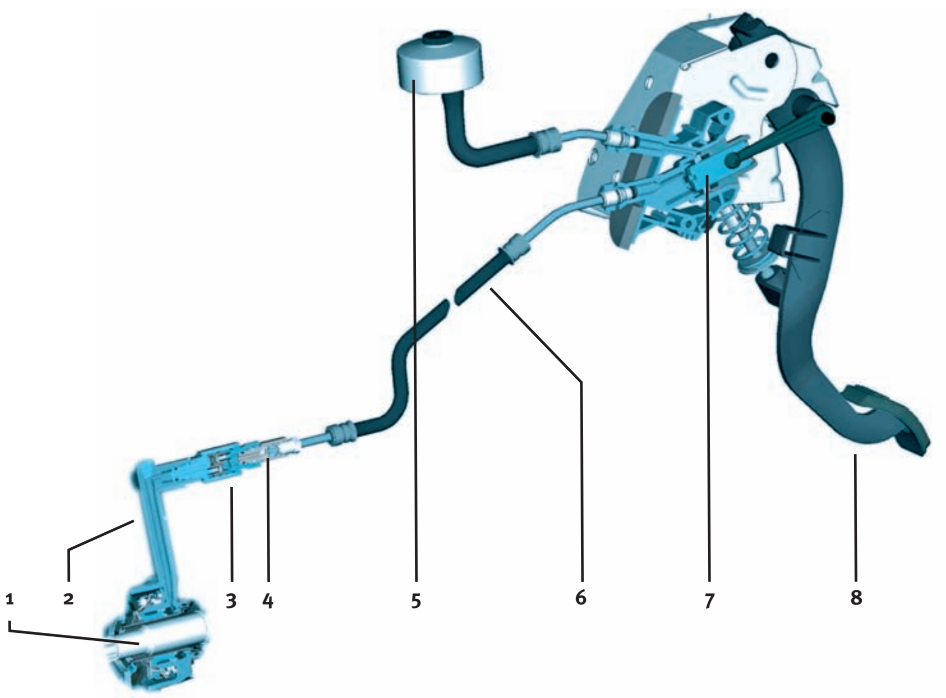
1 = Transmission input shaft
2 = Concentric slave cylinder CSC
3 = Anti-vibration unit
4 = Peak torque limiter
5 = Reservoir with hydraulic fluid
6 = Hose
7 = Master cylinder
8 = Pedal



Clutch release system with CSC

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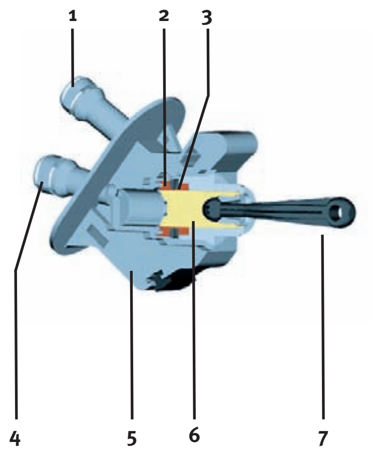


1 = Connection to the reservoir

2 = Primary seal

3 = Secondary seal

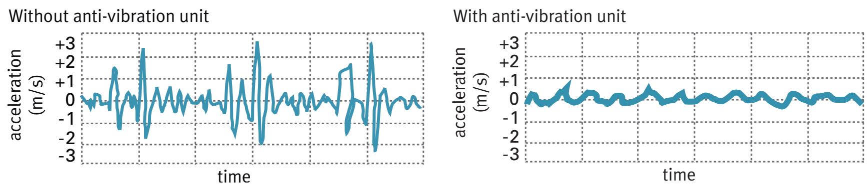
4 = Connection to the pressure line

5 = Housing

6 = Piston

7 = Piston rod

Master cylinder



Vibration at the clutch pedal

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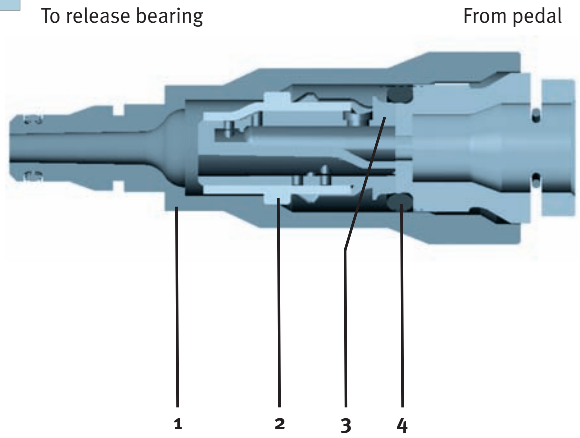

1 = Housing

2 = Hose element

3 = Hose support

4 = Swell seal ring

Anti-vibration unit with tube valve

ed to the brake fluid reservoir via a hose; but there are also solutions providing a separate reservoir for the clutch cylinder. The primary seal separates the reservoir from the hydraulic chamber, thus generating the pressure required to actuate the clutch. The secondary seal seals up the low-pressure chamber of the reservoir against its environment. With the clutch pedal released, a spring at the pedal or in the master cylinder pushes the piston back entirely. In this position, the connection between the reservoir and the pressure chamber is open which allows the air trapped in the system to leak. It also facilitates filling the system of a new part.

Hose

Analogous to the brake line, the hydraulic pressure line consists of a flexible hose and rigid tubing. The hose is required to compensate for the movement between the chassis and the power train. When installing the hose, it is important to make sure there is no direct contact with other components in the engine compartment. Furthermore, it must be ensured that the tubing is not damaged, kinked or corroded. There is a growing trend for fluid-connectors made from rubber. Here, it is mandated that they are not installed adjacently to hot components (such as turbo charger or exhaust manifold).

Vibration damper

The internal combustion process of the engine causes vibration which is passed from the clutch components via the release bearing to the clutch pedal. This is noticeable to the driver in the form of a tingling in the foot and noise. In order to prevent this, filter elements can be integrated in the tubes or hoses. This could be either membrane dampers or anti-vibration units (figure 44) with two non-return valves acting in opposite direction.

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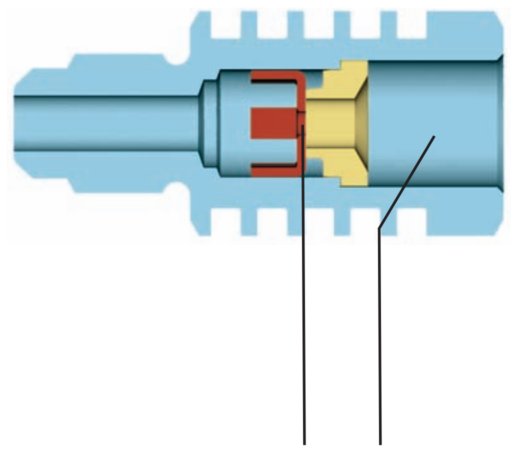

Peak torque limiter. Movable orifices inside the hydraulic line control the flow volume when the clutch is engaged at high speeds

Peak torque limiter

Peak torque limiters (figure 45) are movable orifices within the hydraulic line which reduce the volume flow during clutch engagement. They protect the drive train from overload caused by a sudden clutch engagement, e.g. if the driver's foot slips off the clutch pedal. During maintenance, peak torque limiters must never be removed from the hydraulic system, as this can damage the transmission, the drive shafts or the dual mass flywheel.

Slave cylinder

In semi-hydraulic systems the slave cylinder is typically located at the outside of the transmission housing or serves as actuation device for the clutch lever. In this case, the slave cylinder comprises the housing, the piston with sealing, a precharge spring and a bleed screw. The precharge spring applies constant preload pressure on the release bearing which therefore rotates even if the release system is free of load. In this manner, no undesirable noise occurs. The bleed screw facilitates the flushing of the system during maintenance. In systems with CSC (figure 47), the release bearing is directly connected to the piston and tensioned against the diaphragm spring tips of the clutch by the integrated precharge spring. The release movement of the clutch is initiated by hydraulic pressure: when engaging the clutch, the diaphragm spring pushes the central piston into its original position and the fluid flows back into the master cylinder. Designed with a large travel, the slave cylinder is able to adjust to tolerances occurring during installation or as a result of clutch wear and tear.

Concentric slave cylinder

In designs with CSC (Fig. 48) the release bearing is directly connected to the piston. The release movement of the clutch is initiated by hydraulic pressure: when engaging the clutch the diaphragm spring pushes the central piston towards its original position and allows the fluid to flow back into the master cylinder. Designed with a large travel, the slave cylinder is able to compensate for tolerances occurring during installation or as a result of clutch wear and tear.

Sensors

Master and slave cylinders are increasingly equipped with sensors which measure actuation travel and transmit the reading to the engine and transmission control unit. Whether or not a system is sensor-equipped can be seen from a little case attached to either the master or slave cylinder which houses plugs and cable connectors. Sensors are precisely tuned to the master or slave cylinder to which they are attached and must therefore never be removed from and attached to another cylinder. Failure of one component mandates the replacement of the entire cylinder/sensor unit.

Hydraulic fluid

If not otherwise indicated by the vehicle manufacturer, hydraulic systems use brake fluid. When leaving production, the system is pre-filled with fluid. When in operation, water accumulates in the brake fluid and causes the boiling point to decrease. In the worst case when ambient temperatures are high, this can lead to vapour bubbles in the slave cylinder, which in turn can cause clutch decoupling problems. In order to prevent this from happening, it is recommended to change the brake fluid two to three times a year. It is mandatory to observe the manufacturers recommendations to make sure the correct fluid is used. Failure to do so can damage the seals or cause noise emissions at the master cylinder.

Maintaining a hydraulic release system normally requires no more than replacing

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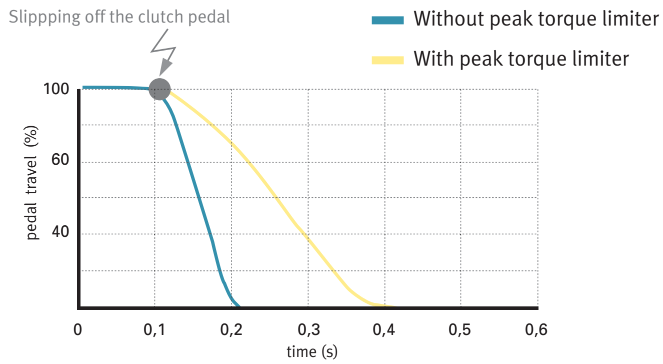

Peak torque limiters are used to control flow volume

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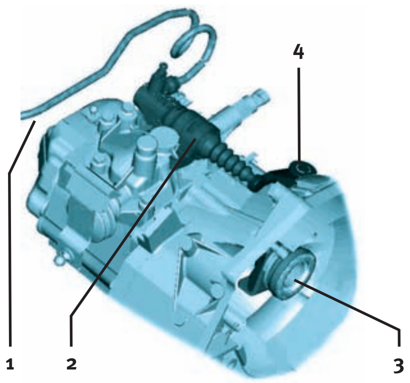

1 = Clutch pressure line
2 = Slave cylinder
3 = Release bearing
4 = Release fork

Slave cylinder attached to the gearbox exterior

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1 = Pre-load spring
2 = Piston
3 = Seal
4 = Housing
5 = Dust bellows
6 = Release bearing

In designs with CSC the release bearing is directly connected to the piston

the brake fluid at regular intervals. Professional workshops use a special filling device for a quick and clean procedure. If no special tool is available, the re-filling of the system with fluid can be performed by repeatedly depressing and releasing the pedal while opening and closing the bleed screw simultaneously. To make sure the system is flushed entirely and to avoid air bubbles trapped in the system, here too, the manufactures specifications should be observed.

Cleanliness when working on the hydraulic system is a crucial factor. Even the smallest contamination can cause leakages and malfunctions. Systems designed for brake fluid must be protected from mineral oil ingress. Re-lubrication of the cylinders or connectors is forbidden for the same reason. Even the smallest amount of mineral oil can destroy the seals. For clutch systems using the same reservoir as the brake system, there is the risk of carrying over contaminations from the brake system.

Important

When replacing the clutch, performing a visual inspection of the CSC is recommended. Replace the CSC, if there are any signs indicating leakage, excessive thermal load, stiff operation of the bearing or hydraulic system, or advanced wear of the bearing ring at thediaphragm spring.

Benefits of hydraulic release systems:

- Flexible installation of the hydraulic line
- Good actuation comfort through less friction
- Vibration and noise optimised
- Ease of installation and maintenance
- Integrated wear adjustment

Automated Shift Gearbox (ASG)

Use

The automated shift gearbox is a further development on the commonly used manual gearbox. While providing the same driving comfort as a manual transmission, the ASG changes gears automatically and even reduces fuel consumption provided that the system components are precisely tuned to each other.

These qualities make the ASG a good solution for small and medium-size models, in particular because it is less costly than the fully automatic gearbox. The automated shift gearbox is also used on commercial vehicles, sometimes even as standard equipment in heavy-duty applications.

Description

Handling

Like automatic transmissions, the ASG has a selector lever with the positions ‘neutral’, ‘reverse’, ‘automatic’ and ‘manual’ The connection between the selector lever and the transmission is made electronically and not mechanically.

As the automated shift gearbox is based on manual transmission, no ‘park’ position is required. As with the manual gearbox the selected gear remains engaged and the clutch is closed automatically when the ignition is switched off.

Like the automatic transmission, to start the engine

- the selector lever must be in ‘neutral’
- the gearbox must show 'neutral'
- and the brake pedal must be depressed.

Technology

Two electric motors mounted to the transmission perform the tasks of coupling and switching. A transmission control unit is responsible for coordination and signal processing.

Most signals are read and emitted by the CAN bus. The CAN bus is connected to the engine control unit, the ABS or ESP control units and the cockpit control unit, which also displays information about the currently selected gear and gearbox status.

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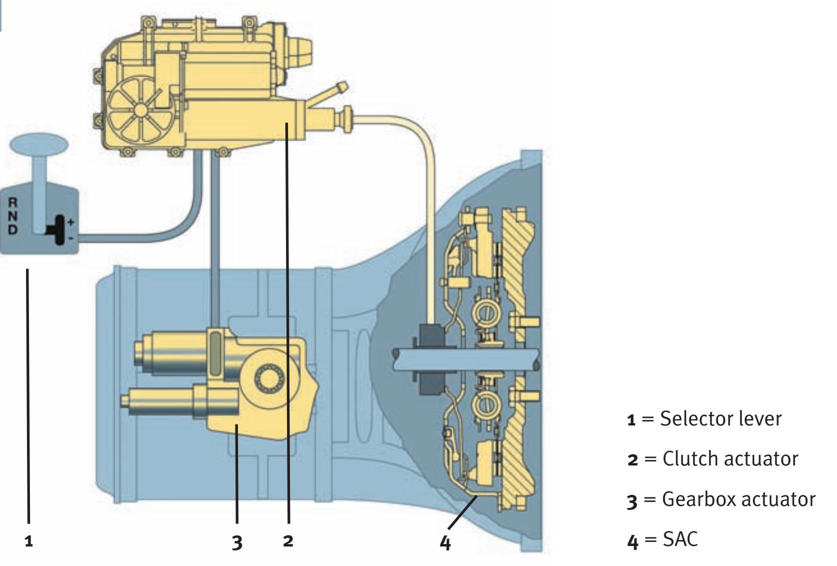

ASG function diagram

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ASG selector lever
Source: Opel/Vauxhall

In the LuK ASG a common housing accommodates the control unit, the electric motor and the components for mechanical clutch actuation.

When replacing the control unit, the software corresponding to the vehicle model must be installed and an initial start-up procedure performed. The remaining control units are also synchronized with the transmission control unit, which is why it doesn’t make sense to retrofit a manual transmission with an automated shift gearbox either from a technical or from an economical point of view.

To make the electric motors as small, light and responsive as possible, the clutch actuation force must be reduced to a minimum. This is achieved by using a self-adjusting clutch (SAC).

To change gears, an assembly group of two electric motors is used which replaces the manual gearshift dome. The first electric motor selects the gutter, which corresponds to the transverse motion of the hand during manual gearshift. The other, bigger, electric motor engages the gears.

Functions

Creep function

Releasing the brake makes the clutch engage slightly. On a level surface the car then starts to roll without having to push the accelerator. To protect the clutch, the applied torque is limited and will be reduced when the clutch temperature rises.

Determining the clutch touch point

Due to thermal fluctuations and other external factors the point at which the clutch begins to transfer engine torque to the wheels varies. This point is called touch point. The automated shift gearbox adjusts the touch point whenever the brake pedal is depressed and the engine runs for a prolonged period of time, for example at traffic lights.

The system closes the clutch until the soft engagement of the clutch pressure plate and the driven plate causes a reaction of the engine. The clutch then immediately disengages again.

This process is normally imperceptible to the driver and requires a stable idling behaviour of the engine. When replacing the clutch or control unit it is also essential to perform a successful commissioning procedure using the appropriate test device to ensure correct functioning. The correct touch point adjustment ensures that the clutch engages smoothly without extended slipping periods.

Clutch protection

The automated shift gearbox detects if the clutch is running hot, for example due to repeated hill starts. In order to slow down the heating-up of the clutch, the system deactivates the creep function gradually and closes the clutch faster when starting. This avoids driving with a slipping clutch over long periods of time.

ABS brake support

If the ABS control unit signals a braking action with ABS support, the automated shift gearbox can disengage the clutch. Decoupling the engine improves ABS control efficiency.

Safety monitoring

In the ISM (Intelligent Safety Monitoring System) the main processor is monitored by a second processor. It checks the functions of the memory and the program sequences, and whether the LuK ASG control unit responds accurately to the current driving situation.

In case of a malfunction, the control unit will react in two possible ways:

- Switching off the actuators output stage; this means no more movement is possible
- Control unit reset; this means after several seconds the control unit goes back to normal operation.

Both measures prevent unpredictable vehicle reactions caused by a defect of the control unit.

Advantages

- Good efficiency and low fuel consumption thanks to ideal shift points
- Automated or manual gear shift
- Easy manoeuvering without risk of stalling
- Small and light-weight components
- High driving comfort
- Cost-effective

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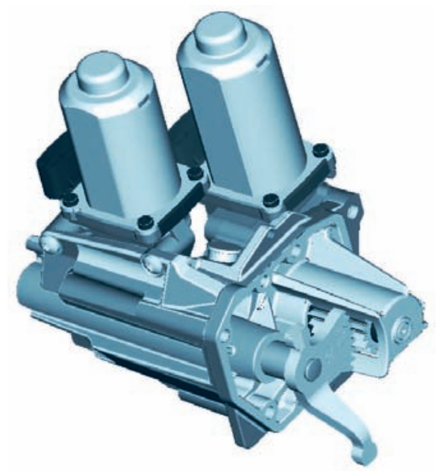

Gearbox actuator

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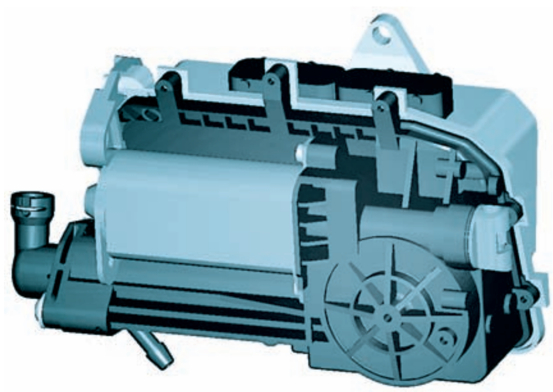

Clutch actuator

The double-clutch gearbox

Parallel shift gearbox PSG – the two-in-one transmission

In times of increasing fuel costs automobile manufacturers and suppliers face the challenge of developing innovative drive concepts to produce engines with lower fuel consumption and reduced emissions. This applies in particular to automatic transmissions, which currently fail to achieve a synthesis between driver comfort and fuel efficiency.

By combining the advantages of automatic and manual transmissions LuK has developed a smooth-running automatic gearbox which offers the efficiency and responsiveness of a manual gearbox.

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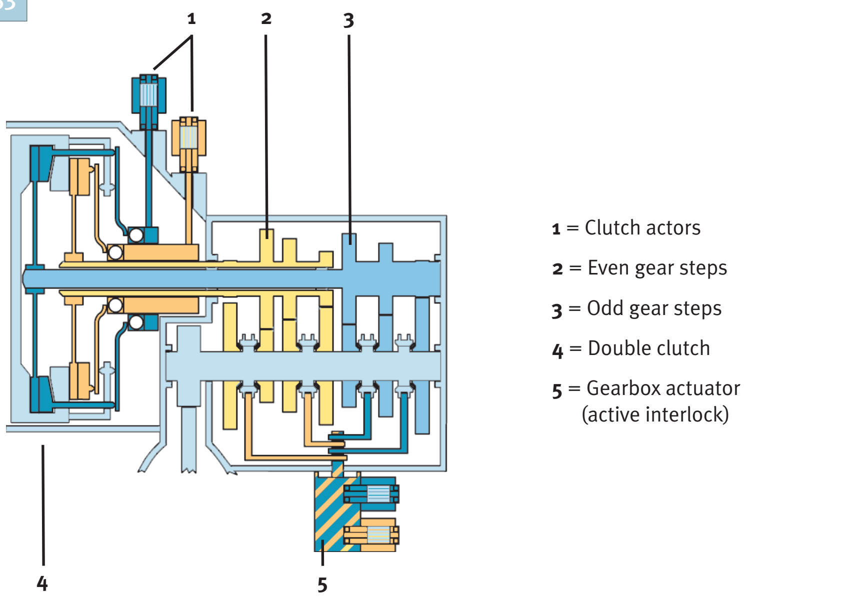

Engine torque is continuously passed from one partial gear to the other

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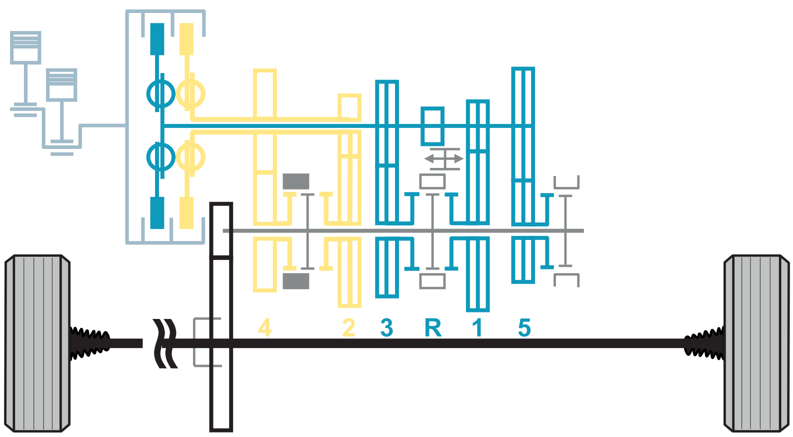

Parallel shift gearbox

Basic principle of the parallel shift gearbox

The idea underlying the parallel shift gearbox is simple: In order to achieve high efficiency, the gearbox functions as if manually operated. However, in order to provide the comfort of an automatic transmission, traction must not be interrupted during gear shift. The parallel shift gearbox is therefore divided into two partial gears: one for the even, the other for the odd gear steps. Each of the partial gears is assigned a separate clutch. An intelligent gearbox control unit controls, opens and closes both clutches by actuating the engagement system.

During operation one partial gear is positively engaged at all times, while the inactive partial gear already preselects the next gear step according to the driving situation. This allows for an uninterrupted change of gears. As both clutches then open and close simultaneously, engine torque is continuously passed from one partial gear to the other. Thus traction is not interrupted during gear shift. Gear shift and gear selection are controlled automatically without the driver hardly even noticing.

The problem facing engineers is now how to design two parallel transmission trains to require as little installation space as possible.

Double clutch design

The key component of the transmission is the double clutch. Its task is to transfer engine torque to both partial gears. The partial gears are arranged in line and each of the clutch discs drives one of the nes-ted transmission input shafts. Owing to its overall weight the double clutch, unlike manual gearbox clutches, is not

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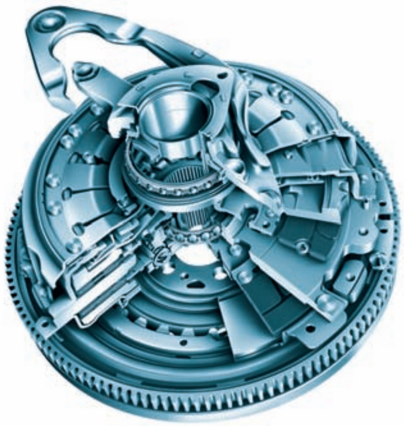

Double clutch

mounted on the crankshaft together with the DMF, but is supported by the second transmission input shaft.

Engine torque is transferred to the central plate of the double clutch via the DMF's internal toothing. With one of the clutches engaged torque is transferred from the central plate to the respective transmission input shaft via the clutch disc.

As both clutches are actuated from the same side, the force generated by clutch 1

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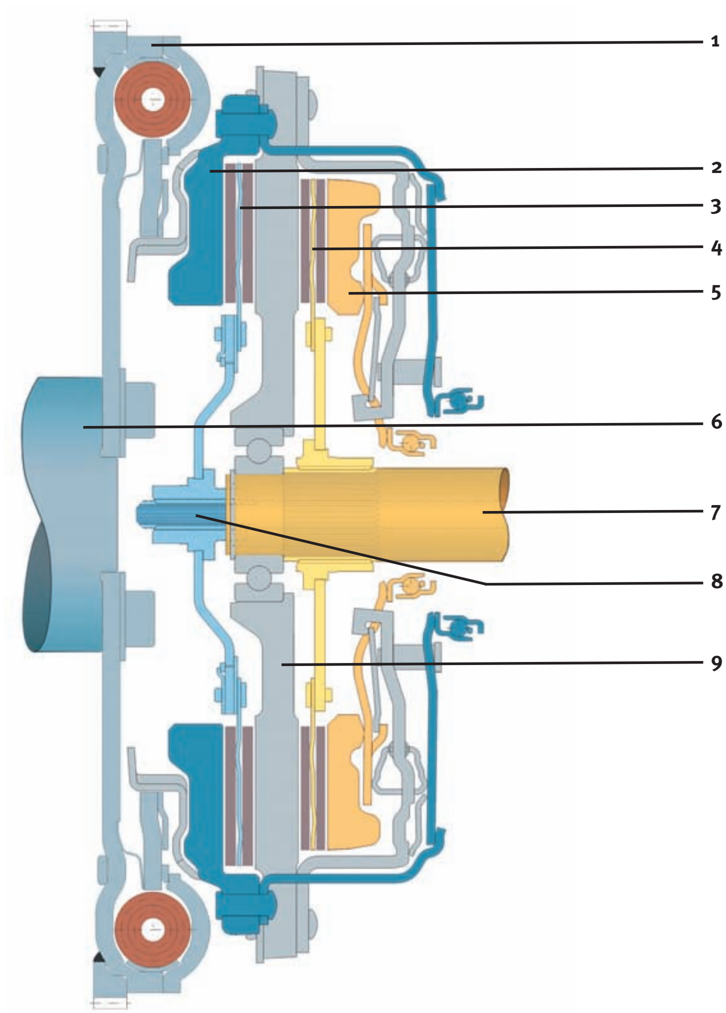

1 = Dual mass flywheel
2 = Pressure plate C1
3 = Driven plate C1
4 = Driven plate C2
5 = Pressure plate C2
6 = Crankshaft
7 = Transmission input shaft (hollow shaft) C2
8 = Transmission input shaft (solid shaft) C1
9 = Drive disc

Double clutch design

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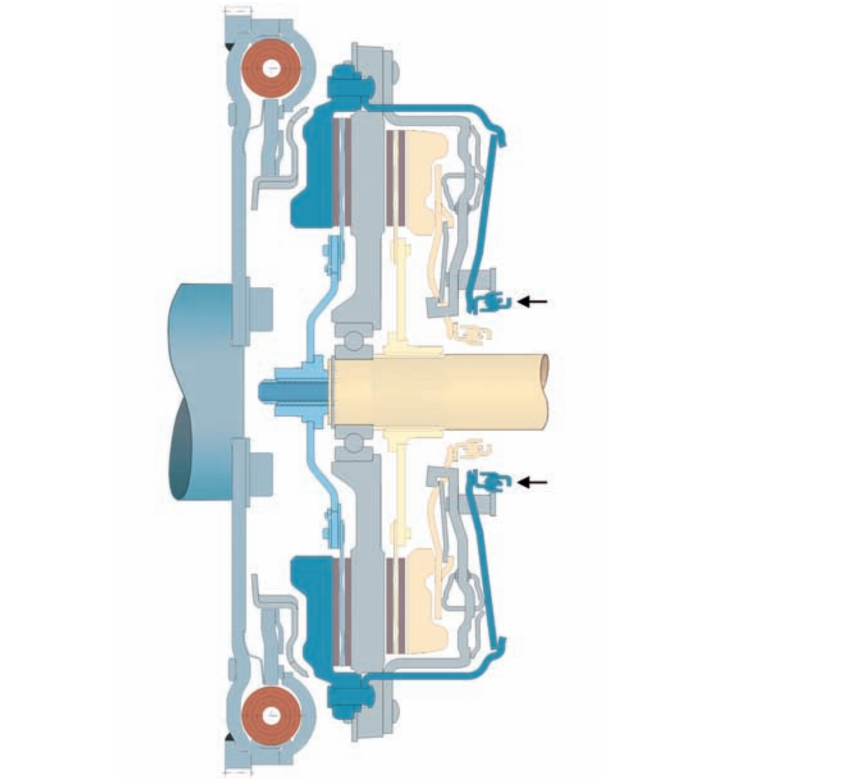

Clutch 1 closed

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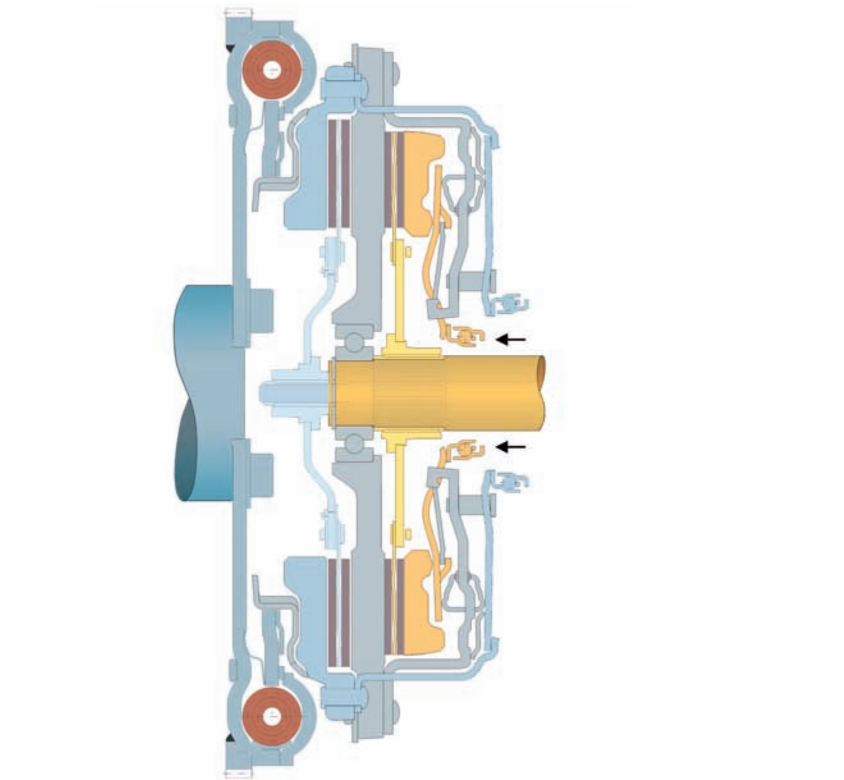

Clutch 2 closed

is reversed, so as to turn the pressing motion of the engagement system into a pulling motion at the clutch pressure plate. This clamps the clutch disc between the central plate and the pressure plate and allows torque to be transferred.

Force reversal is not required for clutch 2. The lever spring is supported against the clutch cover and presses the clutch pressure plate towards the central plate.

The double clutch, like the LuK SAC, is designed with a self-adjusting mechanism which compensates for clutch disc wear throughout its service life. Firstly, this guarantees for evenly low actuation forces over the entire clutch life and, secondly, it ensures a constant clutch characteristic which benefits the controllability of the system.

Parallel shift gearbox – advantages and innovation

Without a doubt, the double clutch is the one most important and significant innovations. Contrary to conventional double-clutch gearbox concepts, LuK uses a specially developed dry double clutch. Since the clutch linings are not immersed in an oil bath, this design is distinguished by its very good efficiency. In conjunction with the high spread of gear ratios provided by a 7-gear transmission, LuK engineers have managed to reduce both fuel consumption and CO_2 emissions considerably – to such a degree even, that some applications with parallel shift gearbox consume less fuel than models with manual gearshift.

Operation and comfort of the new gearbox generation is similar to automatic transmissions. The clutch design enables the vehicle to roll off in the first or reverse gears, even if the gas pedal is not depressed. This facilitates manoeuvering or driving in stop-and-go traffic considerably. The start-off characteristic, however, bears greater resemblance to a manual transmission. As the clutch and gearshift operations are controlled automatically, the clutch pedal has disappeared and the gearshift lever been replaced by a selector lever. Of course, the driver has the option to change gears manually using the Tiptronic mode.

At the beginning of 2007 volume production of the first dry double-clutch gearbox was launched. The new gearbox/clutch concept provides an exceptional combination of comfort and responsiveness on the one hand with fuel efficiency on the other.

CVT Functions and components

Continuously Variable Transmission CVT
Stepped automatic transmissions and manual gearboxes have fixed gear steps which do not allow the engine to always operate in the ideal range. This is only possible if the transmission can vary continuously from maximum (start) to minimum ratio. Eliminating fixed gear steps thus leads to a significant improvement in driving comfort and performance while optimising fuel efficiency.

Since 1993 LuK has been developing components for continuously variable transmissions using the belt/chain principle. The aim was to develop a technology which was able to safely transfer engine torques of up to 300 Nm while improving both engine performance and fuel efficiency. In so doing, LuK has managed to set itself apart from its competitors.

In this design the LuK chain runs between tow sets of tapered pulleys, each consisting of a fixed and a moveable pulley. The axially moveable pulley is supported by the shaft; its axial movement is hydraulically controlled.

The axial displacement of the moveable pulley causes a change in the running radius of the chain and in turn a change in the transmission ratio.

Analogous to a conventional clutch, torque is transferred by means of friction. Therefore, it is essential to make sure that the clamping load applied on the tapered pulleys is sufficient to safely transfer not only normal torque levels, but also torque peaks occurring on the wheelside without slip of the linking element. The clamping load and ratio control of the pulley sets are regulated hydraulically.

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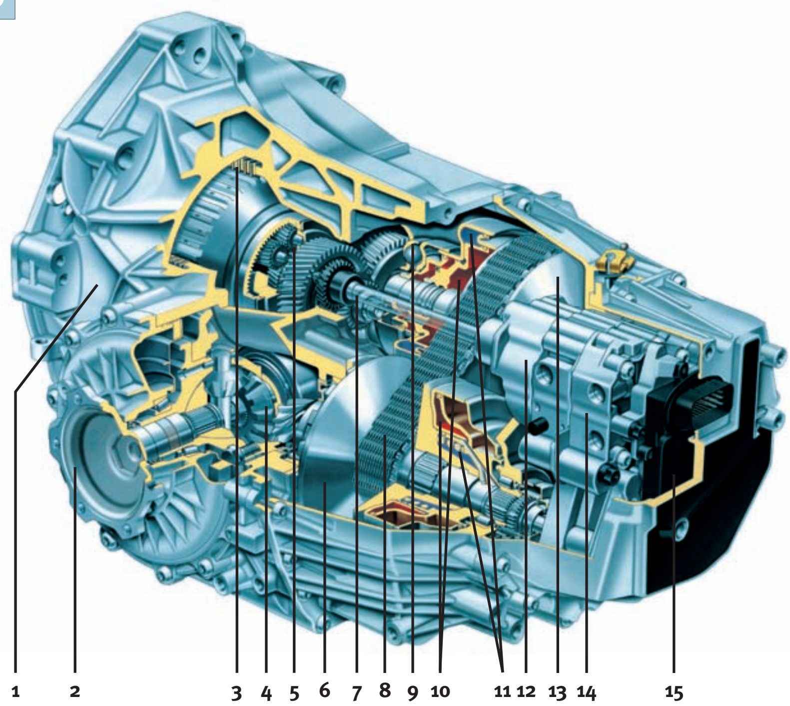

1 = Light-alloy housing
2 = Output
3 = Multiple-disc clutch
4 = Differential
5 = Planetary reversing gear set
6 = Set of secondary pulleys
7 = Pump drive
8 = Link-plate chain
9 = Torque sensor
10 = Clamp load chambers
11 = Ratio control chambers
12 = Pump
13 = Set of primary pulleys
14 = Hydraulic control
15 = Control electronics

Audi multitronic with LuK CVT components

Design of a CVT
Besides adjusting the transmission ratio, the gearbox fulfils other important functions such as start-up and reverse function. The figure above shows the design of a CVT gearbox, here the Audi multitronic® used in large-volume production since 1999 on a variety of models.

Figure 59 shows a planetary reversing transmission with forward and reverse clutch. The double planetary gear set used here allows for an identical transmission ratio during both forward and backward motion.

Besides selecting the correct clamping loadand ratio control, these functions are also ensured by hydraulically actuating the relevant clutches. The hydraulic system, in turn, is regulated by the control electronics.

The multitronic® uses a wet multi-disc

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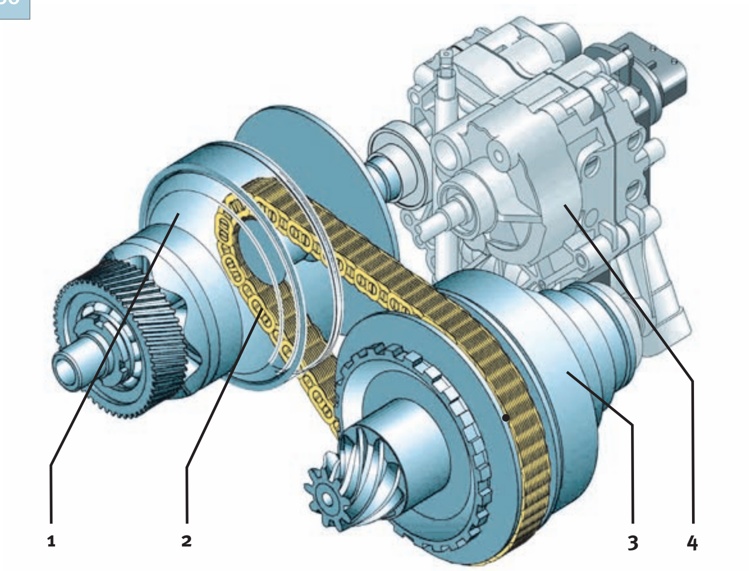

LuK CVT components

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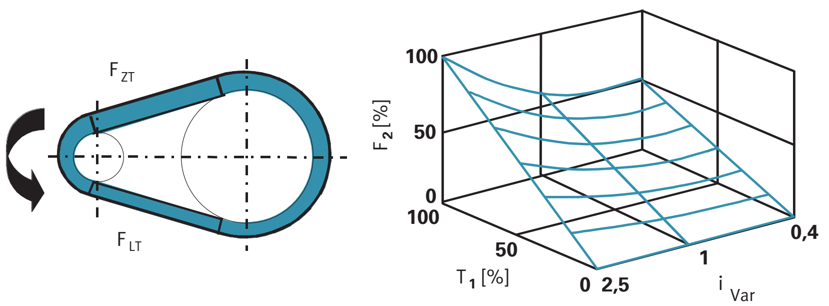

Continuously variable power transmission by means of frictional engagement

clutch as start-up device. Alternatively, a hydrodynamic torque converter or a hydraulic clutch can be used on the CVT. The torque is transferred to the primary set of pulleys via a gear stage which allows the overall transmission ratio to be adjusted to different engine types. The dual-stage torque sensor, whose function will be described in detail later, is visibly positioned on the primary set of pulleys. The sets of pulleys are designed according to the dual piston principle, i.e. separate cylinders are used for clamping load adjustment and ratio control. The secondary set of pulleys is positioned directly on the pinion shaft, which in turn drives the ring-gear. Torque is transferred via the differential and the flanges to the vehicle's drive shafts. Figure 60 shows the hydraulic system (including the pump) with the control electronics attached to it. The graphic also shows the pump drive, an either internal gear or vane cell pump, both developed by LuK.

Continuously variable power transmission by means of frictional engagement

The continuously variable power transmission by means of frictional engagement can only be assured, if the system is able to generate the required clamping load under all operating conditions. The ideal clamping load must be high enough to prevent variator slip while avoiding overload and consequently poor efficiency rates. Figure 61 highlights the relationship between the input torque and the required clamping load at the secondary set of pulleys as a function of the transmission ratio.

Taking into consideration the above said, engine torque fluctuation as well as sudden torque input from the wheel going along with very high engine speeds and torque gradients is of particular importance, e.g. when actuating the ABS brake system during the transition from an icy patch to asphalt or when the car skids off the sidewalk with the wheels spinning. LuK was able to resolve this problem by developing the hydro-mechanical torque sensor.

The dual-stage torque sensor

The principle behind the function of the dual-stage torque sensor is described in the following paragraph. Both, the single and the dual-stage torque sensors are highlighted.

Torque is induced using a ramp plate, through which the power flows over balls to an axially moveable sensor piston supported by oil pressure.

Oil coming from the pump flows out through a discharge bore whose flow resistance changes with the movement of the sensor piston until equilibrium is established between the axial force of the ball ramp and the compression force. In this manner, the torque sensor adjusts the pressure routed directly into the clamping cylinder in exact proportion to the adjacent torque.

The moveable sensor plate closes the discharge bore if there is a sudden change in torque. If torque continues to rise, the sensor plate then actively forces the oil out of the torque sensor chamber

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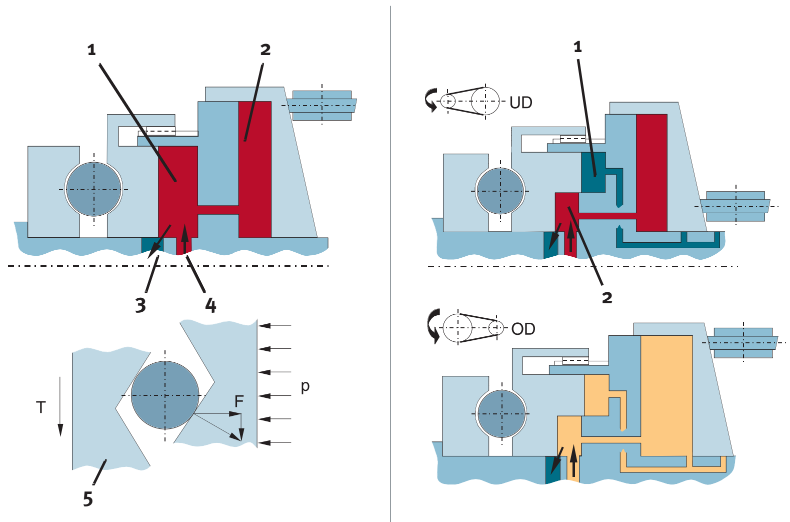

1 = Sensor chamber
2 = Clamping cylinder
3 = Discharge bore
4 = Inlet bore
5 = Ramp plate

1 = Sensor chamber 2
2 = Sensor chamber 1

Single and dual-stage torque sensors

into the pulleys to increase the clamping force. In other words, the torque sensor acts like a pump for a short time. This “back-up pump”, which works only when needed and does not require any drive power, can provide a short-term flow of more than 30 l/min in the event of a sudden change in torque.

To produce a two-stage characteristic curve, the pressure area of the sensor piston is divided into two parts. In underdrive, where clamping force must be higher to transmit the torque due to the small effective radius of the chain, pressure acts on only one part of the surface. To equalize the ramp force, supplied by torque, the pressure in the torque sensor must be high and consequently also in the clamping cylinder. In overdrive, be-yond the switch point, pressure acts on both parts of the surface. This is why the clamping force is lower at a given torque. Changing the gear ratio creates axial displacement of the moveable pulley flange of the primary pulley. This then switches the characteristic curve directly by enabling or disabling the second partial surface.

In underdrive, the second partial surface is ventilated by the right switch bore at atmospheric pressure as shown in the figure. However, in overdrive, this bore is closed by the moveable pulley flange, and the left switch bore provides a connection to the hydraulic fluid.

This system is currently undergoing further development. Alternatively, the said dual-stage torque sensor is also available as continuously variable system by adjusting

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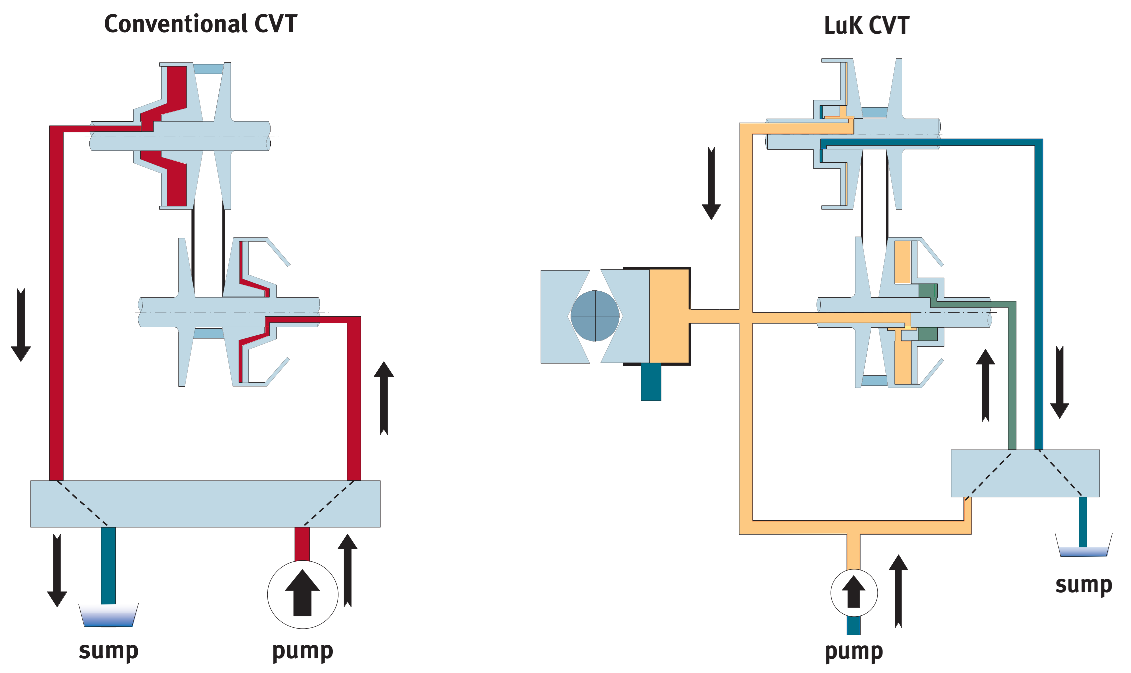

Comparison between conventional vs. LuK hydraulics

the design of the ramps accordingly. Optionally, an electronic clamping load control in combination with a slip-controlled clamping system can be used.

LuK dual-piston system with torque sensor

As shown in figure 63, conventional systems have one pressure cylinder on the drive-side pulley and one on the output-side pulley – also in a nested tandem arrangement. The oil flows from the pump to a control unit that directs the pressure to be induced in the cylinders. These cylinders combine clamping and ration controlfunctions into one component.

The primary cylinder surface is often designed to be significantly larger than the secondary surface. The main reason for this is the inability of many CVT hydraulic systems to set primary cylinder pressure higher than secondary cylinder pressure.

For a rapid adjustment into underdrive mode, the pump must satisfy the high flow requirements of the entire secondary cylinder surface. At the same time, hydraulic fluid is released from the primary pulley into the oil sump, which results in a loss of energy. This occurs similarly for overdrive adjustments. Therefore, a pump with a large volumetric flow capacity is neces-sary to fulfil the system's dynamic require-ments, with the corresponding negative effect on the pump's energy requirements.

The LuK double piston model divides the cylinder areas into partial surfaces (red) that ensure clamping, and smaller, separate partial surfaces (blue or green) that are responsible for the ratio control. As we have already discussed, the dual-stage torque sensor ensures clamping. Adjusting the pulleys requires only a small volume of oil to service the comparatively small surfaces of the adjustment cylinders. When the variable speed mechanism is adjusted, the clamping oil, which is under high pressure, is transported directly from one pul-ley to the other without requiring any additional expenditure of energy. This means the pump for the LuK double piston principle is significantly smaller than pumps for conventional CVT systems, which improves overall transmission efficiency and subsequently fuel consumption.

Primary set of pulleys – design

Figure 64 shows a sample design of the primary set of pulleys with the LuK dual piston and the dual-stage torque sensor which has been described in detail earlier in this chapter. Marked in red is the oil supply of the clamping cylinder, green is the supply of the ration control cylinder. The blue section is chamber 2 of the torque sensor and corresponding feed.Torque is transferred between the shaft and the moveable pulley by means of toothing.

The sets of pulleys can be produced inexpensively from sheet metal moulded parts, for the production of which LuK can rely on its vast expertise in clutch engineering. The parts' geometries have been optimised consistently thanks to FE calculations which allow LuK also to make best possible use of the maximum-spread of gear ratios.

Jacket seals are used for dynamic sealing, for static sealing, O-rings are used.

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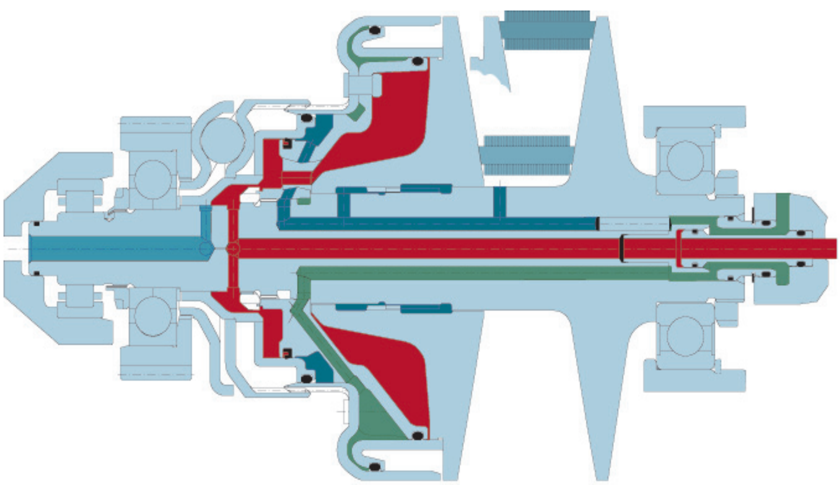

Primary set of pulleys with integrated torque sensor

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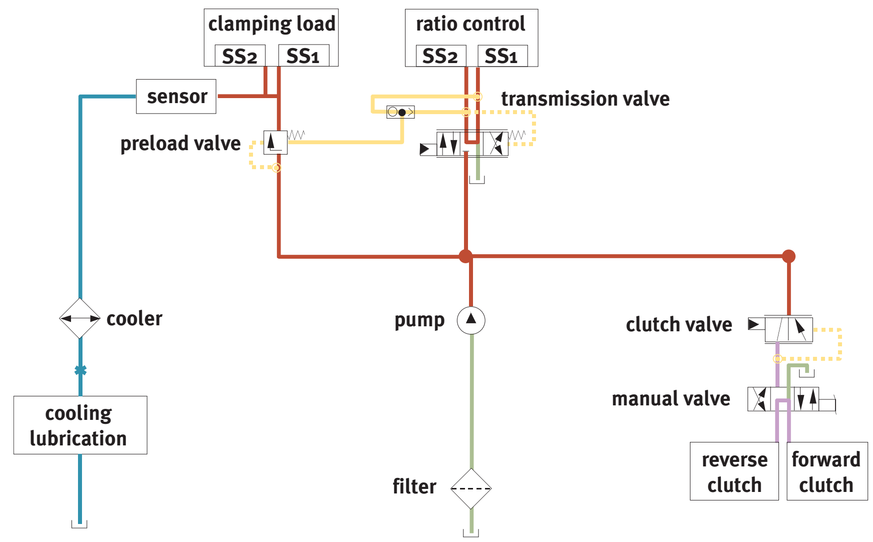

Simplified hydraulics schematic

Simplified hydraulics schematic –CVT with lock-up clutch

Figure 65 shows the simplified hydraulics schematic for a CVT with lock-up clutch. A pump with a primary suction filter feeds the system. In this manner, the preload valve, the transmission valve and the clutch valve are fed, too. A manual valve makes sure that pressure is applied on the forward and reverse clutch.

The preload valve is a pressure limiting valve. When sensor pressure is low while high transmission pressure is required, it is the function of the pre-load valve to deliver a pressure differential. Depending on the operating condition, either the pressure of the torque sensor or the adjusting pressure at one of the pulley sets is the determining factor. Via an offset it is ensured that the required pre-control pressure is always applied. The present schematic does not show the pre-control lines.

The oil flowing through the outlet bore of the torque sensor is passed through the

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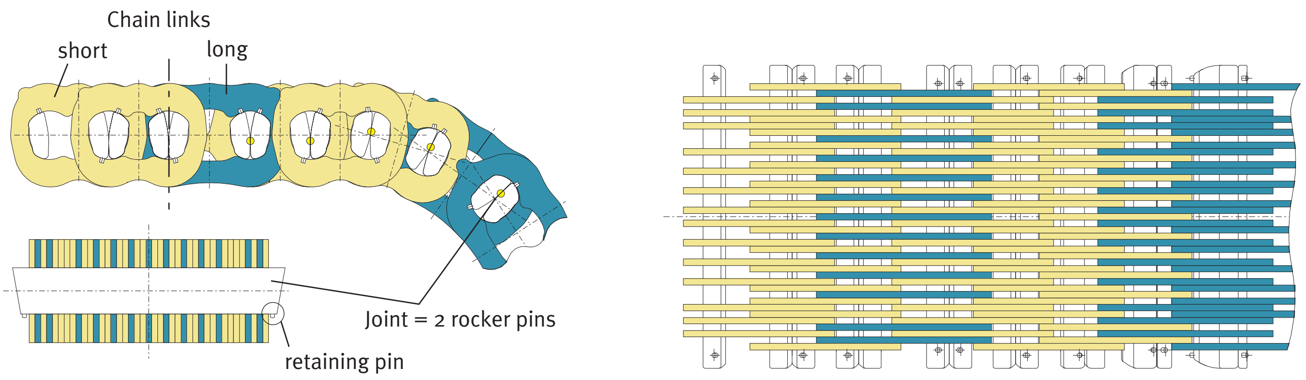

Design and components of the LuK chain

radiator where it is used for cooling and lubricating the system.

The control system requiring no more than 9 manual valves and 3 proportional valves is compact and light weight. At full load, pressure increases up to 60 bar with peak pressures of 100 bar. Owing to high-precision manufacturing technique, valve play is minimised and leakages further reduced.

The LuK chain

Using a rocker pin chain made by P.I.V. Antrieb Werner Reimers as a starting point, LuK made further improvements to the CVT chain for automotive applications. The development process focused on improving its strength to achieve the high power density required and on improving its acoustic properties.

Figure 66 illustrates the CVT chain for applications producing torque up to 300 Nm. It is constructed of various links, which form the strands, the rocker pins, and retaining elements.

The CVT chain has the following characteristics:

- It has low fuel consumption and excellent power transmission performance. This is made possible by the rocker pin design of the CVT chain, which allows short rotations around the pulley flanges and a high spread of gear ratios.
- The CVT chain allows for the transmission of high torque levels. Because the rocker pins are able to “seesaw”, the chain is able to equalise load distribution.
- The chain excels in low internal friction losses owing to the meshing of the rocker pins,thus ensuring high transmission efficiency
- The CVT chain is resistant to axle offset due to the rocker pins’ crowned faces and its linked plate structure. In combination with cambered pulley flanges, these elements reduce additional axle offset created whenever ratio is changed. Furthermore, the CVT chain is resistant to pulley deformation under load, angular errors and relative rotations between the fixed and moveable pulley flanges.

Torque converter

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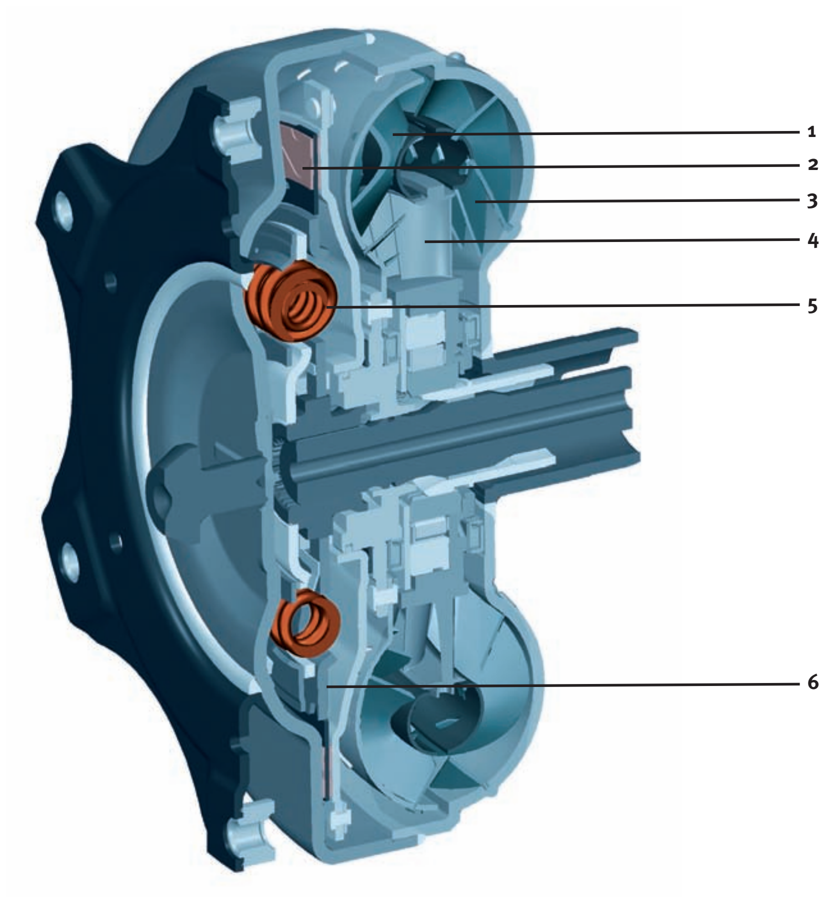

1 = Turbine

2 = Friction lining

3 = Pump

4 = Stator

5 = Torsion damper

6 = Lock-up clutch with torsion damper

Cross-sectional view of a hydraulic torque converter with lock-up clutch

LuK torque converter – competitive edge through optimization

For decades the torque converter has been used as starting and transfer element in automatic transmissions. Forecasts show that alternative concepts will not entirely edge out torque converters if the existing system of converter plus transmission keeps being improved.

Torsion damper

As a torque converter can in principle only transfer torque with slip, its operation always involves power loss. An ideal solution would be to use the benefits of the torque converter for driving off, and then immediately close the lock-up clutch for normal driving. To absorb noise and vibration LuK offers specially synchronized high-performance torsion dampers, produced by the millions, which are able to minimize vibration through targeted damping and adjustment of the spring rates. In contrast to conventional dampers, turbine torsion dampers, dual torsion dampers and mass dampers can be used to avoid any slip at all – depending on the design concept.

Controlled operation at low continuous slip

Another possibility of avoiding vibration is operating the vehicle at low continuous slip. In combination with this approach for vibration minimization LuK offers an innovative system for cooling friction linings, distinguished by its long durability and outstanding cooling performance. The pioneering system allows for a simplified design and the use of cost-effective standard friction linings. A beneficial side effect is that it also extends the service life of the gearbox oil which is all to the good of state-of-the-art transmissions which are subject to particularly high loads. For its vibration optimization concepts LuK can draw on its long-existing experience in the power train and on simulation technologies, which also help to reduce project realization times.

Flow circuit optimization

For flow circuit optimization LuK engineers rely on 3D flow simulation software, which enable them to make precise parameter forecasts at an early stage. This also makes it possible to reduce the overall size as well as the mass moment of inertia for the same flow performance or, conversely, leave the overall size unchanged in order to make the torque converter more fuel-efficient than competitors' products.

Thus LuK offers an innovative and flexibly adjustable range of torque converter applications for automatic transmissions which proves that sustained further improvement can yield a leap in performance on systems claimed to have reached its limits.

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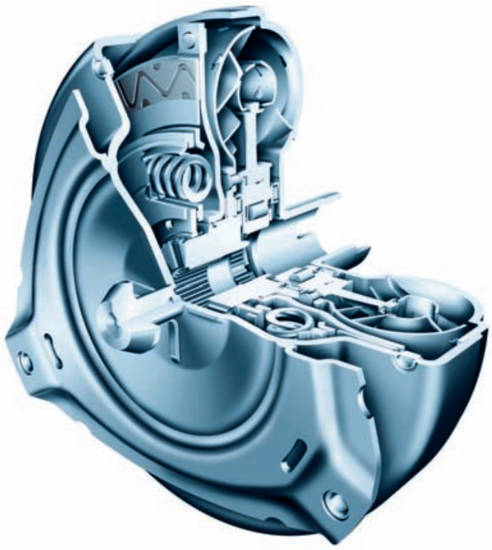

Hydraulic torque converter with lock-up clutch

LuK torque converters – the benefits:

- High hydrodynamic efficiency
- Earlier lock-up through innovative dampers
- New higher-performance systems for friction cooling

Closing remarks

Ever growing demands in comfort, reliability, efficiency and environmental friendliness are the driving forces behind pioneering solutions. Vehicle engineers strive to successfully implement new materials and alternative energy sources in state-of-the-art motor vehicles. LuK has always been excited by new challenges and will continue to develop trend-setting standards in automotive technology also in the future.

Join us on our mission.

Notes

LuK	✓	✓	✓	✓	✓
INA	✓	✓			
INA	✓	✓	✓	✓	
INA	✓	✓	✓	✓	
INA	✓	✓			
LuK	✓	✓	✓	✓	
LuK	✓	✓			
FAG	✓	✓	✓	✓	

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