LuK Clutch Course

An introduction to clutch technology for passenger cars

passenger cars

Introduction

Knowledge is power …

… as the saying goes, and these words acquire new meaning in an age of changing technologies and increasing standards of convenience and engineering. It is no longer a matter of gaining mastery over others, but of mastering technologies and the problems they pose.

This assumes appropriate training and a constant flow of information about changes and about experience gathered. As a maker of clutches for almost every automotive manufacturer in the world, LuK knows that its products can only display their qualities to the full when they are properly installed and professionally maintained.

This brochure is intended to give all readers an overview of the basic principles and designs of modern clutch technology. An important aim is to make clear that clutches have become precision parts which need careful handling and must be installed and removed exactly as per the assembly instructions.

Supplementary training documents, such as the NOK brochure are likewise available for purchase. AS/LuK has also developed an interactive teaching programme.

The European Training Concept

This deals with the basic principles of clutch technology, correct diagnosis of damage and essential service tips. This training programme, which works with the aid of video, can be purchased as a self-tutoring unit (for private study) or as a trainer guide (for trainers).

AS can provide you with more information if you need it – please call +49 6103 753251.

LuK hopes that you enjoy working with this booklet and wishes you every success both learning and subsequently working "on the spot".

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Contents

A history of clutch technology

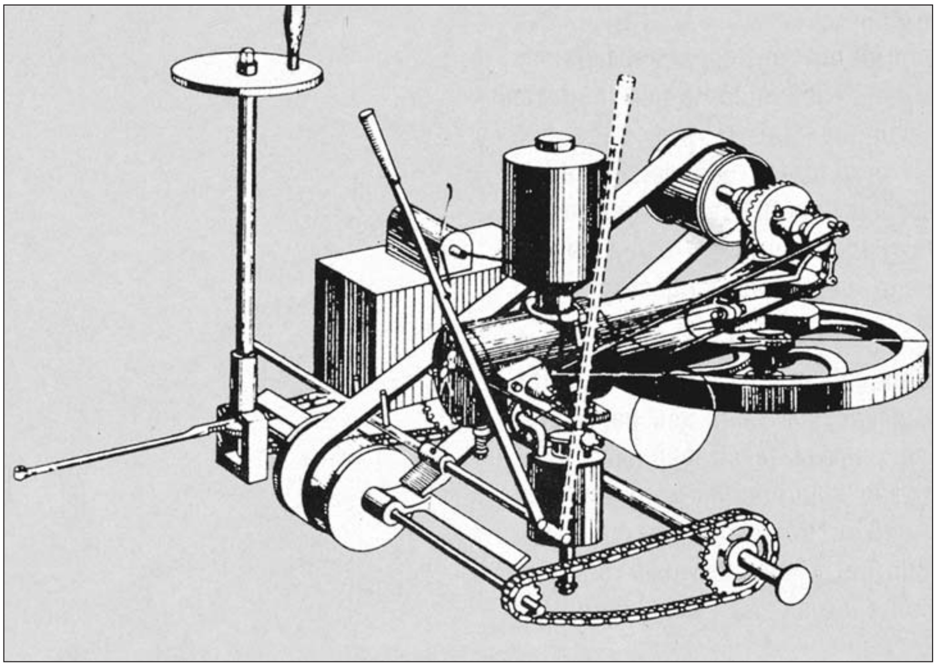
In the course of over 100 years of automotive history, nearly all components have undergone enormous technological developments. Reliability, production costs and service-friendliness as well as, more recently, environmental safety, have been and continue to be the criteria demanding new and better solutions 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.

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.

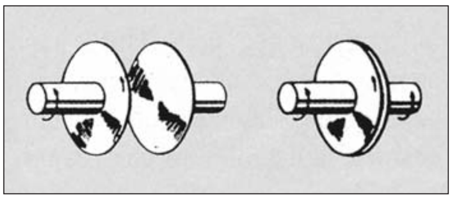


Transmission belt clutch from the Benz patented motor car of 1886.

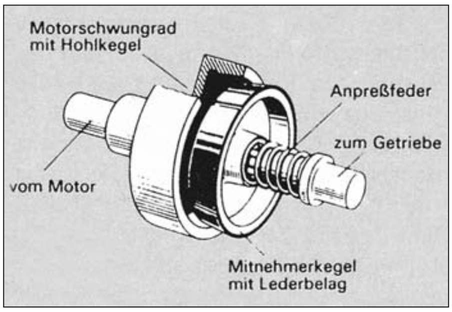
The results were a wide variety of clutch types – including the forerunners of our present-day clutches – all based on the principle of the friction clutch. Here the disc is located on the end of the crankshaft and is joined by a second, stationary disc. When the two make contact, friction is produced 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.

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.



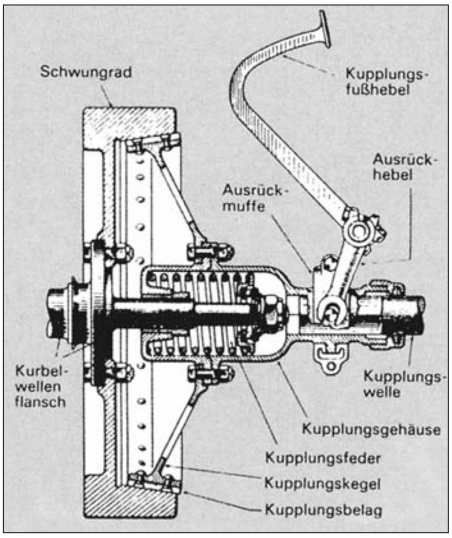
The basic principle of the friction clutch:
The driven disc is pressed onto the driving disc until the frictional connection is made.



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

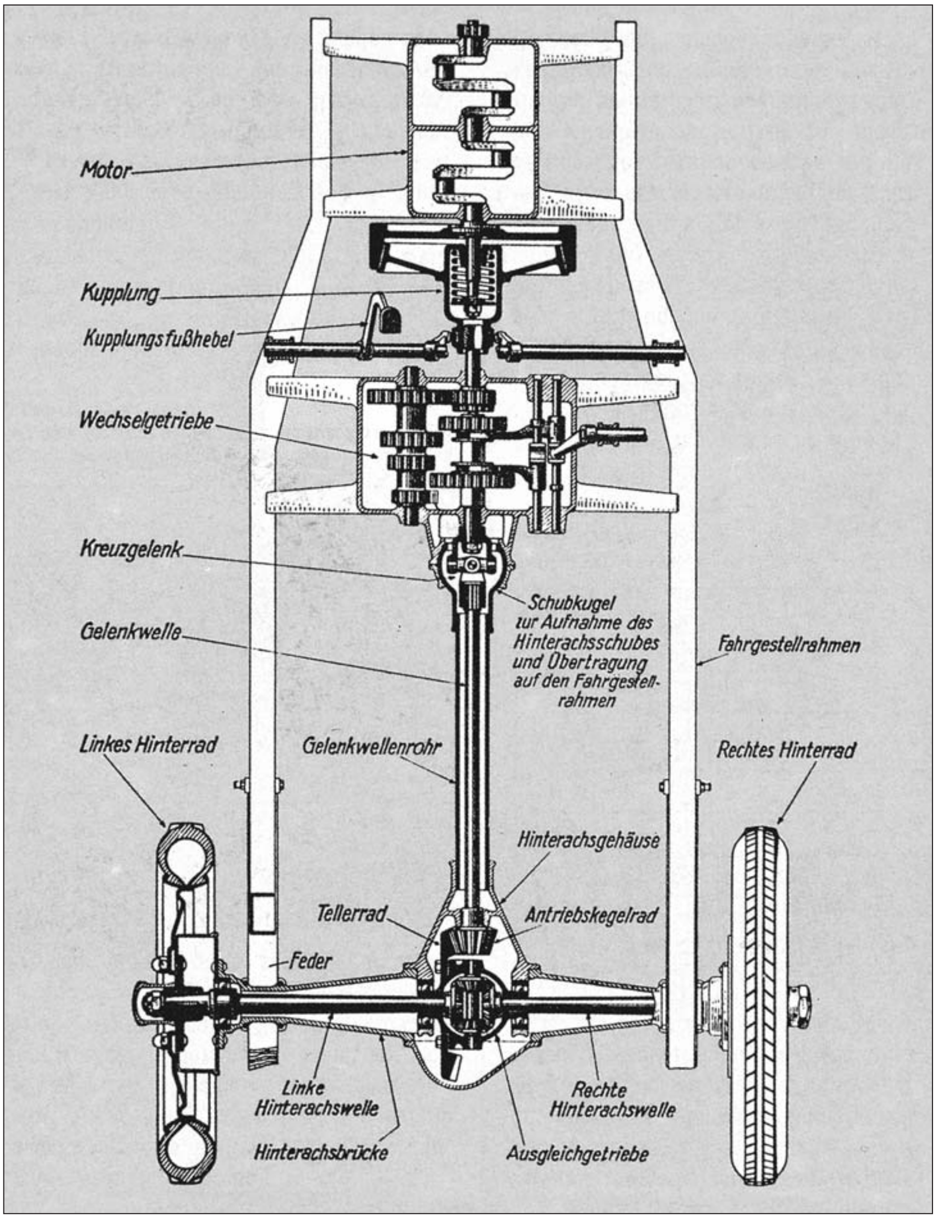
Motorschwungrad mit Hohlkegel	*engine flywheel with hollow cone*
vom Motor	*from engine*
Anpreßfeder	*coil spring*
zum Getriebe	*to transmission*
Mitnehmerkegel mit Lederbelag	*driver cone with leather lining*

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



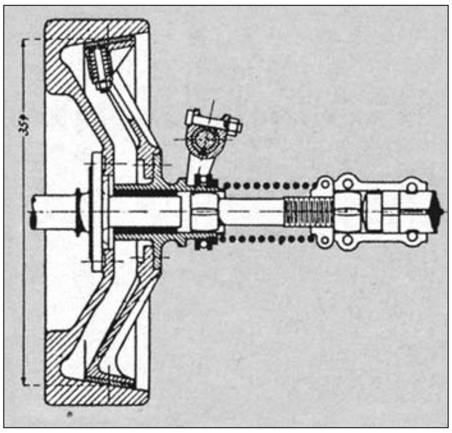
Cross section of a cone clutch showing the typical components: clutch cone and correspondingly turned-out flywheel.

Schwungrad	*crankshaft flange*
Kurbelwellenflansch	*flywheel*
Kupplungsfußhebel	*release sleeve*
Ausrückmuffe	*clutch pedal*
Ausrückhebel	*release lever*
Kupplungswelle	*clutch shaft*
Kupplungsgehäuse	*clutch housing*
Kupplungsfeder	*clutch spring*
Kupplungskegel	*clutch cone*
Kupplungsbelag	*clutch lining*



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.

Motor	*engine*
Kupplung	*clutch*
Kupplungsfußhebel	*clutch pedal*
Wechselgetriebe	*variable-speed transmission*
Kreuzgelenk	*universal joint*
Gelenkwelle	*propshaft*
Linkes Hinterrad	*rear left wheel*
Gelenkwellenrohr	*propshaft tube*
Tellerrad	*differential ring gear*
Feder	*spring*
Linke Hinterachswelle	*left rear axle shaft*
Hinterachsbrücke	*rear-axle housing*
Schubkugel zur Aufnahme des Hinterachsschubes und Übertragung auf den Fahrgestellrahmen	*torque ball for absorbing rear-axle thrust and transferring it to chassis frame*
Hinterachsgehäuse	*differential housing*
Antriebskegelrad	*differential bevel pinion*
Rechte Hinterachswelle	*right rear axle shaft*
Ausgleichgetriebe	*differential*
Fahrgestellrahmen	*chassis frame*
Rechtes Hinterrad	*rear right wheel*



Cone clutch with spring-compressed leather lining.

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.

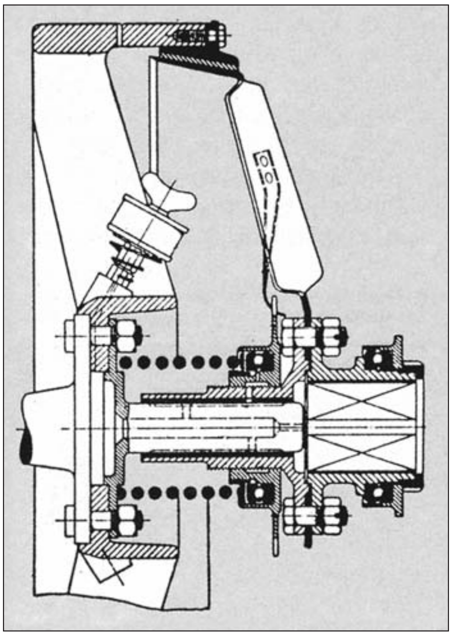
By the end of the First World War, metallic friction linings were becoming increasingly popular. Previously, one had experimented with other solutions: For example, the "Neue Automobil-Gesellschaft (NAG)" constructed a clutch containing a camel-hair lined cone, stamped from sheet metal and equipped with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

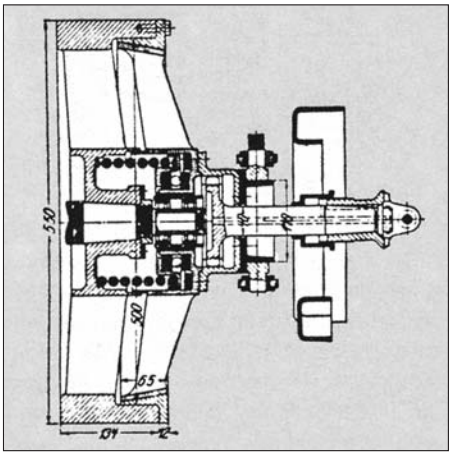
Cone clutches continued to dominate throughout the 1920s thanks to their simplicity. Metallic clutches with cylindrical friction surfaces did not win acceptance because of their poor operational characteristics. Only the spring band clutch, a derivative of the cylindrical clutch that had been installed 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. 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

A history of clutch technology

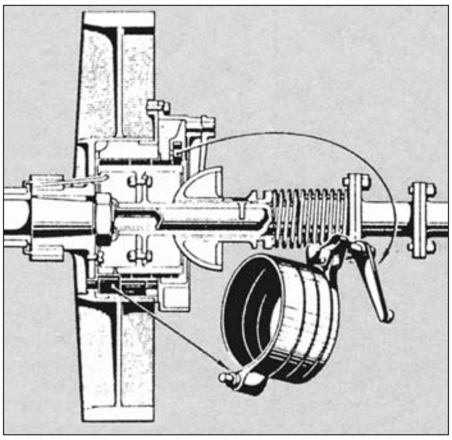


NAG clutch with two-part hollow cone ring, which greatly simplified maintenance.



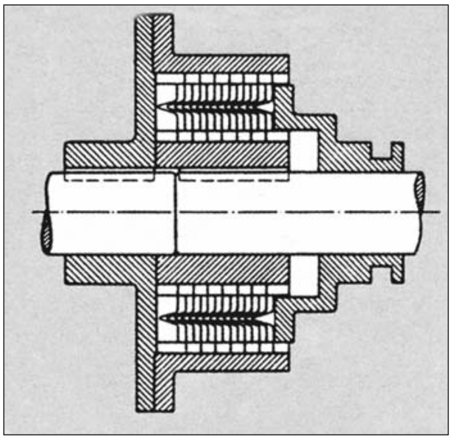
Cone clutch of the Daimler Engine Corporation, with aluminium cone.

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.



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

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



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

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.

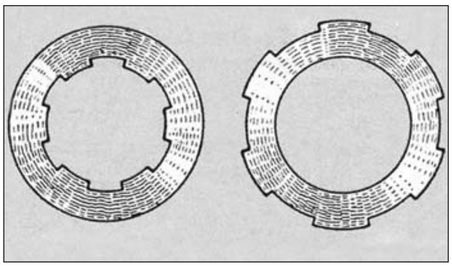


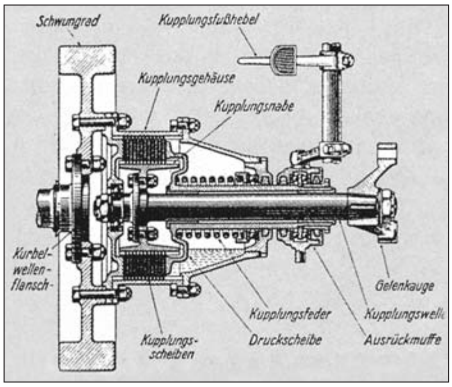
Plate pair from a multi-plate clutch: left, the inner clutch plate, right the outer plate.

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.

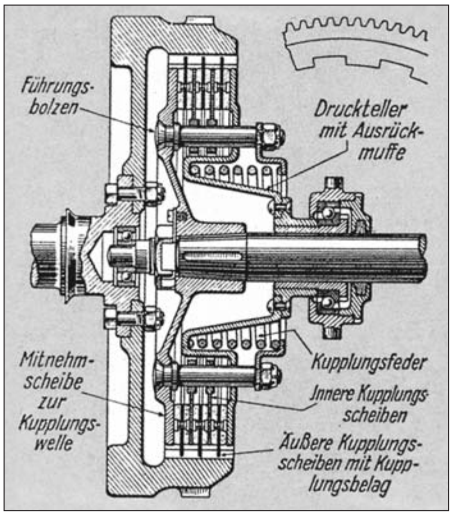
As the spring pressure eased off, the plates disengaged again, in part supported by the spring-loaded strips bent out from the plane of the plate. By varying the number of plate pairs, a basic clutch type could be adjusted to each engine output.

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



Oil-immersed multi-plate clutch.

Schwungrad	*flywheel*
Kurbelwellenflansch	*crankshaft flange*
Kupplungsfußhebel	*clutch housing*
Kupplungsgehäuse	*clutch hub*
Kupplungsnabe	*clutch pedal*
Kupplungsscheiben	*clutch plates*
Kupplungsfeder	*clutch spring*
Druckscheibe	*pressure plate*
Gelenkauge	*propshaft flange*
Kupplungswelle	*clutch shaft*
Ausrückmuffe	*release sleeve*

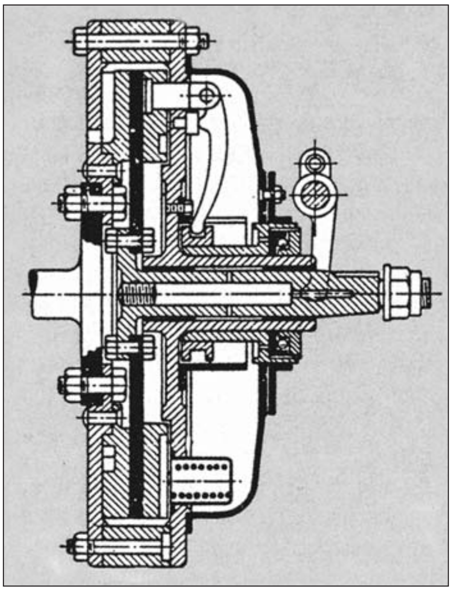


Multi-plate dry clutch with riveted lining.

Führungsbolzen	*guide pin*
Mitnehmscheibe zur Kupplungswelle	*drive disc for clutch shaft*
Druckteller mit Ausrückmuffe	*thrust plate with release sleeve*
Kupplungsfeder	*clutch spring*
Innere Kupplungsscheiben	*inner clutch plates*
Äußere Kupplungsscheiben mit Kupplungsbelag	*outer clutch plates with clutch lining*
Kupplungswelle	*clutch shaft*
Ausrückmuffe	*release sleeve*

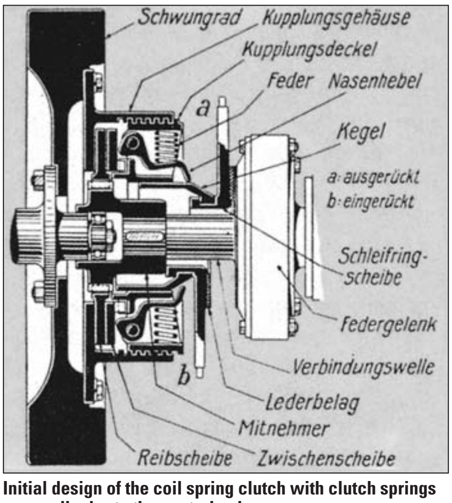
The greatest drawback of the multi-plate clutch was certainly the drag effect, especially in the oil bath, causing only partial disengagement, and thus making gear changing difficult.

By 1904, De Dion & Bouton had introduced the single-plate clutch principle, which because of the initially inadequate materials only came into widespread use in the US during the 1920s – largely on demand from the supply industry,



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

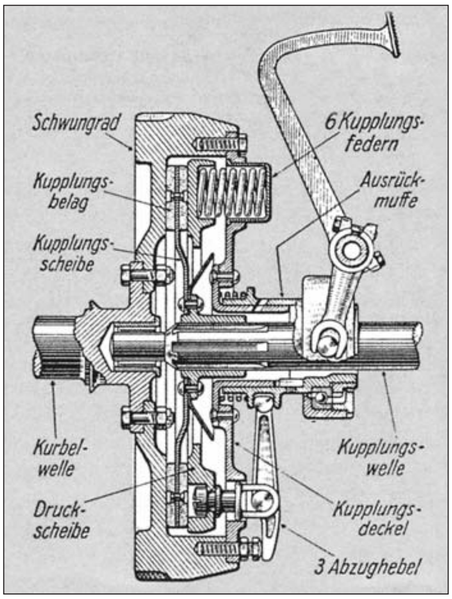
who towards the end of that decade granted licences to European manufacturers. Within a few years, the single-plate had superseded cone and multi-plate clutches.
While De Dion & Bouton still lubricated the friction surfaces of their multi-plate clutches with graphite, clutch technology greatly advanced 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



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

Schwungrad	*flywheel*
Kupplungsgehäuse	*clutch housing*
Kupplungsdeckel	*clutch cover*
Feder	*spring*
Nasenhebel	*lug lever*
Kegel	*cone*
a: ausgerückt / b: eingerückt	*a: disengaged / b: engaged*
Schleifringscheibe	*slip ring disc*
Federgelenk	*spring joint*
Verbindungswelle	*connection shaft*
Lederbelag	*leather lining*
Mitnehmer	*drive disc*
Reibscheibe	*friction disc*
Zwischenscheibe	*intermediate disc*

mass of the clutch plate allowed it to come to rest more quickly when released, making shifting much easier – farewell to transmission brakes.
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.



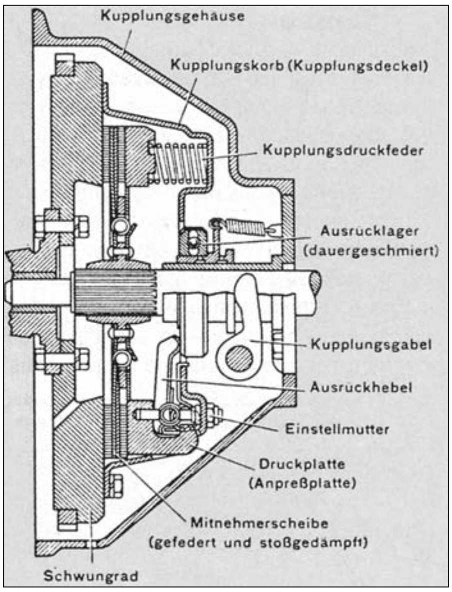
This form of coil spring clutch, with the clutch springs arranged parallel to the central axis, predominated through the 1960s.

Schwungrad	*flywheel*
Kupplungsbelag	*clutch lining*
Kupplungsscheibe	*clutch plate*
Kurbelwelle	*crankshaft*
Druckscheibe	*pressure plate*
6 Kupplungsfedern	*6 clutch springs*
Ausrückmuffe	*release sleeve*
Kupplungswelle	*clutch shaft*
Kupplungsdeckel	*clutch cover*
3 Abzughebel	*operating levers x3*

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 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.
As the diaphragm spring is rotationally symmetric and therefore speed-insensitive, its

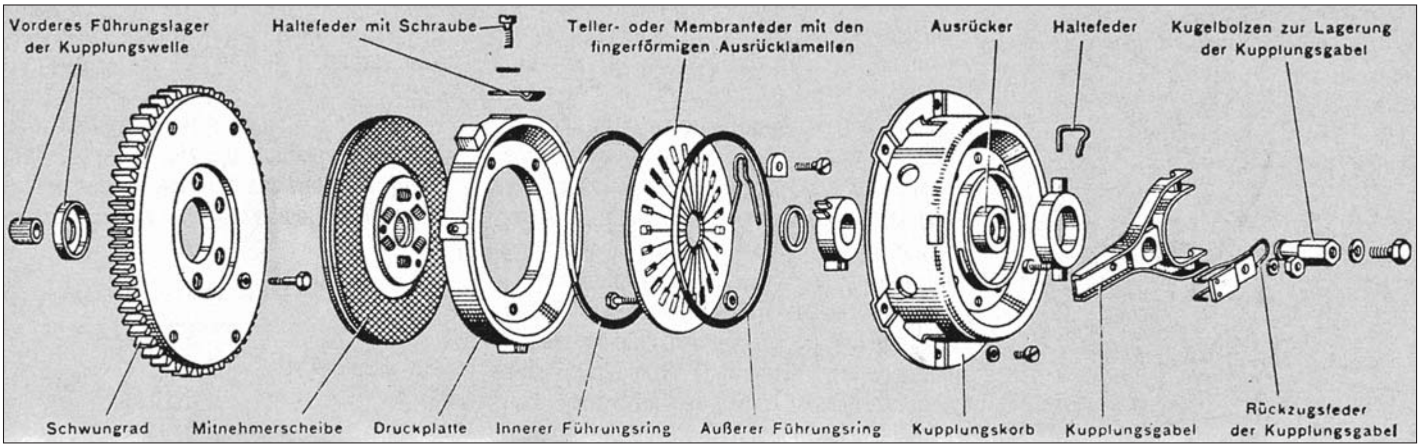


In Britain and the US, the Borg & Beck model with springs located under the clutch cover was the most popular …

Kupplungsgehäuse	*Bellhousing*
Kupplungskorb (Kupplungsdeckel)	*clutch cover*
Kupplungsdruckfeder	*clutch compression spring*
Ausrücklager (dauergeschmiert)	*release bearing (permanently lubricated)*
Kupplungsgabel	*release lever*
Ausrückhebel	*clutch fork*
Einstellmutter	*adjusting nut*
Druckplatte (Anpreßplatte)	*pressure plate*
Mitnehmerscheibe (gefedert und stoßgedämpft)	*driven plate (spring-loaded and damped)*
Schwungrad	*flywheel*

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

A history of clutch technology



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.

Vorderes Führungslager der Kupplungswelle	*spigot bearing*
Haltefeder mit Schraube	*retaining spring with screw*
Teller- oder Membranfeder mit den fingerförmigen Ausrücklamellen	*diaphragm spring with pre-formed fingers*
Ausrücker	*release ring*
Haltefeder	*retaining spring*
Kugelbolzen zur Lagerung der Kupplungsgebel	*ball pin for clutch fork*
Schwungrad	*flywheel*
Mitnehmerscheibe	*drive disc*
Druckplatte	*pressure plate*
Innerer Führungsring	*inner fulcrum ring*
Äußerer Führungsring	*outer fulcrum ring*
Kupplungskorb	*clutch cover*
Kupplungsgabel	*release fork*
Rückzugsfeder der Kupplungsgabel	*return spring of release fork*

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 that 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 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 electromagnetic clutch. In the early 1930s the French firm Cotal built a pre-selector gearbox 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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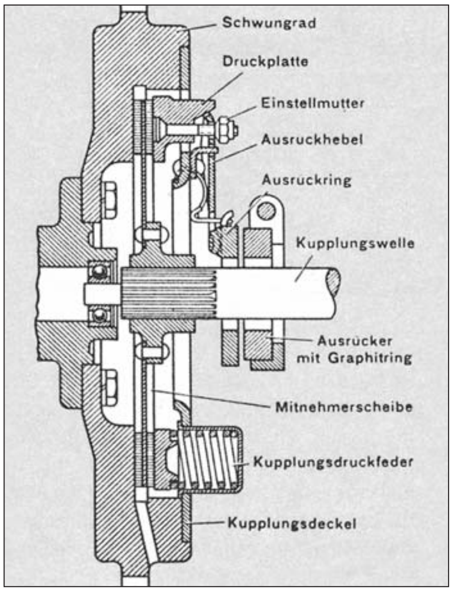
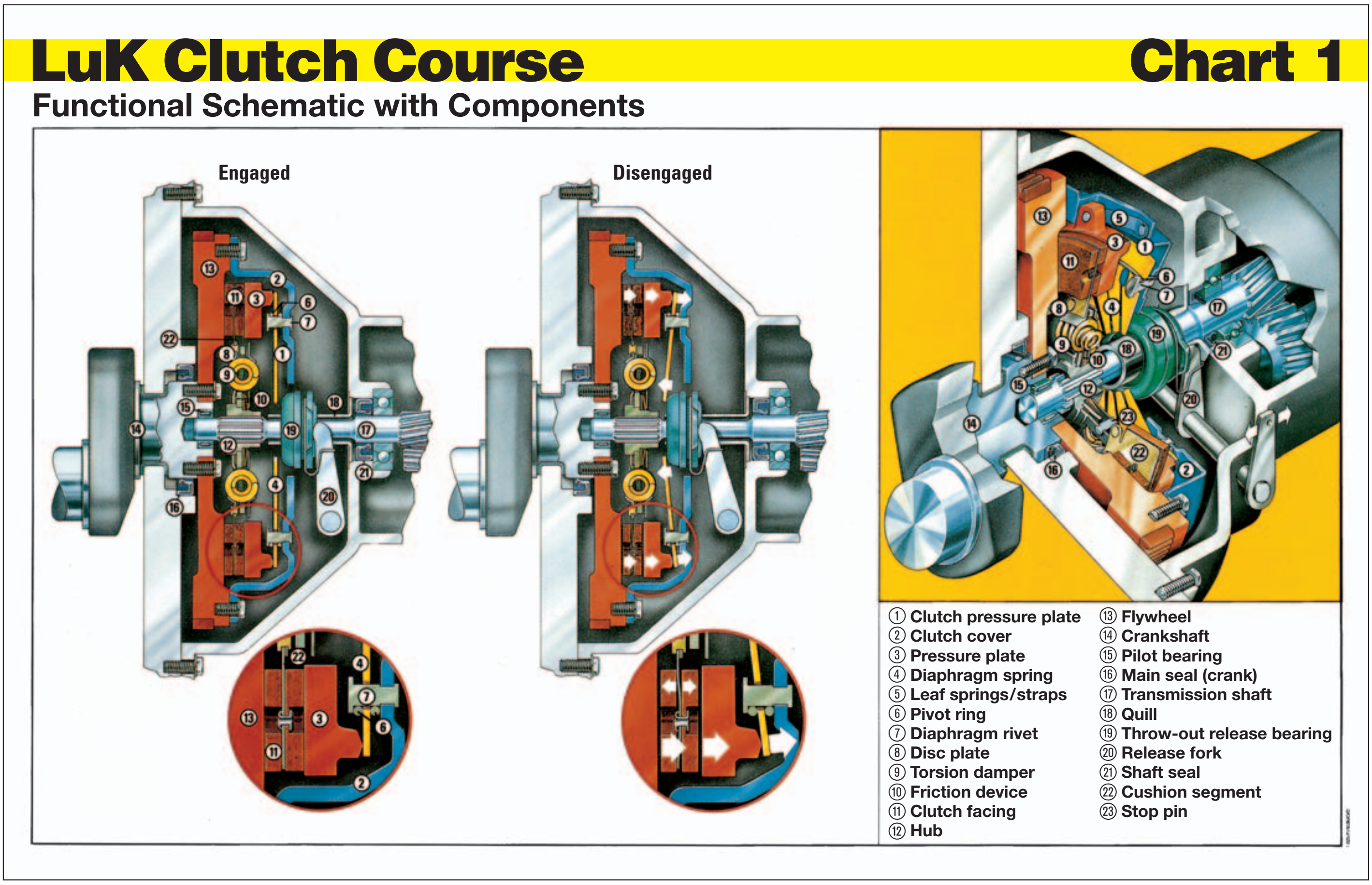


… while on the European continent the version with springs externally located above the clutch cover prevailed.

Schwungrad	*flywheel*
Druckplatte	*pressure plate*
Einstellmutter	*adjusting nut*
Ausrückhebel	*release lever*
Ausrückring	*thrust plate*
Kupplungswelle	*clutch shaft*
Ausrücker mit Graphitring	*release bearing with graphite ring*
Mitnehmerscheibe	*driven plate*
Kupplungsdruckfeder	*clutch compression spring*
Kupplungsdeckel	*clutch cover*

Functional Schematic with components — Chart 1



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

Diaphragm spring clutches, as displayed in chart 1, are also being increasingly used in utility vehicles. They have the following advantages over the coil spring clutches previously used:

- less space
- insensitivity to engine speeds
- lower release loads
- longer life

The diagram on the right, shows a typical clutch installation and highlights its basic use as a connection/separation element between the engine and transmission.

Besides the main function of connecting or separating the crankshaft (14) and the transmission input shaft (17), a modern clutch has several further tasks.

It must:

- enable gentle and jerk-free starting
- ensure fast gear changing of the transmission
- keep engine torsional vibrations as far away from the transmission and thereby decreasing noise and wear
- serve as an overload protection for the entire drive train (e. g. in case of faulty gearchanging)
- be durable and easily replaceable

Functional Schematic with components

The main components of a complete clutch unit are:

The clutch cover assembly (1) with individual parts consisting of the clutch cover pressing (2) clutch pressure plate (3) as the frictional counterpart on the clutch side for the clutch plate, the diaphragm spring (4) for generating the clamp load, the tangential leaf spring (5) as a spring-loaded connection between the cover pressing and pressure plate for providing pressure plate lift, the fulcrum ring (6) and the diaphragm rivet (7) for positioning and providing a mounting for the diaphragm spring.

The clutch driven plate (8) with individual parts consisting of the hub (12), torsion damper (9) with friction device (10) and stop pin (23), the segment cushion spring (22) and the facings riveted to them (11). The flywheel (13) with the spigot (pilot) bearing (15).

The release mechanism with release bearing guide tube (gearbox quill) (18), release bearing (19) and release fork (20).

How the clutch works.

The two diagrams on the left show how a single-plate dry clutch with diaphragm spring operates.

With the clutch engaged (left), the drive from the crankshaft (14) is transmitted via the flywheel (13) to the clutch cover assembly. The clutch driven plate, (8) which is positively engaged to the flywheel and clutch pressure plate through the action of the diaphragm spring (4) transmits the drive via the hub assembly (12) to the transmission input shaft.(17).

Thus the engine – transmission connection is made.

To disconnect the drive between engine and transmission requires the clutch pedal to be depressed, and via the release mechanism (cable or hydraulic) the release fork (20) and the release bearing (19) connected to it moves towards the clutch cover assembly (2), the release bearing acts against the fingers of the diaphragm spring and depresses the diaphragm spring fingers. As further pressure is applied, diaphragm spring load is relieved, and with the aid of the leaf springs (5) the clutch pressure plate moves away from the driven plate. the clutch driven plate is now able to rotate freely – Engine and transmission are now disconnected.
The facing segment cushion spring (22) can be seen on the cutaway diagram. By applying the load uniformly across the facing 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 (9) cannot be understated. Through a combination of coil springs and friction washers tailored to suit specific vehicle applications it can dampen out uneven crankshaft rotation, reducing noise levels, and avoiding premature wear on transmission assemblies. (illustrations 2 and 3 deal with torsion dampers in detail).

The spigot (pilot) bearing (15) serves as a guide and bearing for the transmission input shaft (17).

The guide sleeve (gearbox quill) (18) guides the release bearing (19) centrally to the clutch.

The shaft seals at the engine (16) and transmission (21) are designed to keep the clutch bell-housing free of oil. Even the slightest amount of grease or oil on the clutch facings can considerably impair the friction coefficient.

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

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

where:
rm = mean radius of facings
n = number of facings
m = coefficient of friction of facings
Fa = clamp load
Md = transmittable torque

An example:

Inner diameter of facing:	134 mm
Outside diameter of facing:	190 mm
Clamp load F:	3500 N

$$d_m = \frac{d_i + d_a}{2} = \frac{134\,mm + 190\,mm}{2}$$
$$= 162\,mm \text{ mean facing diameter}$$

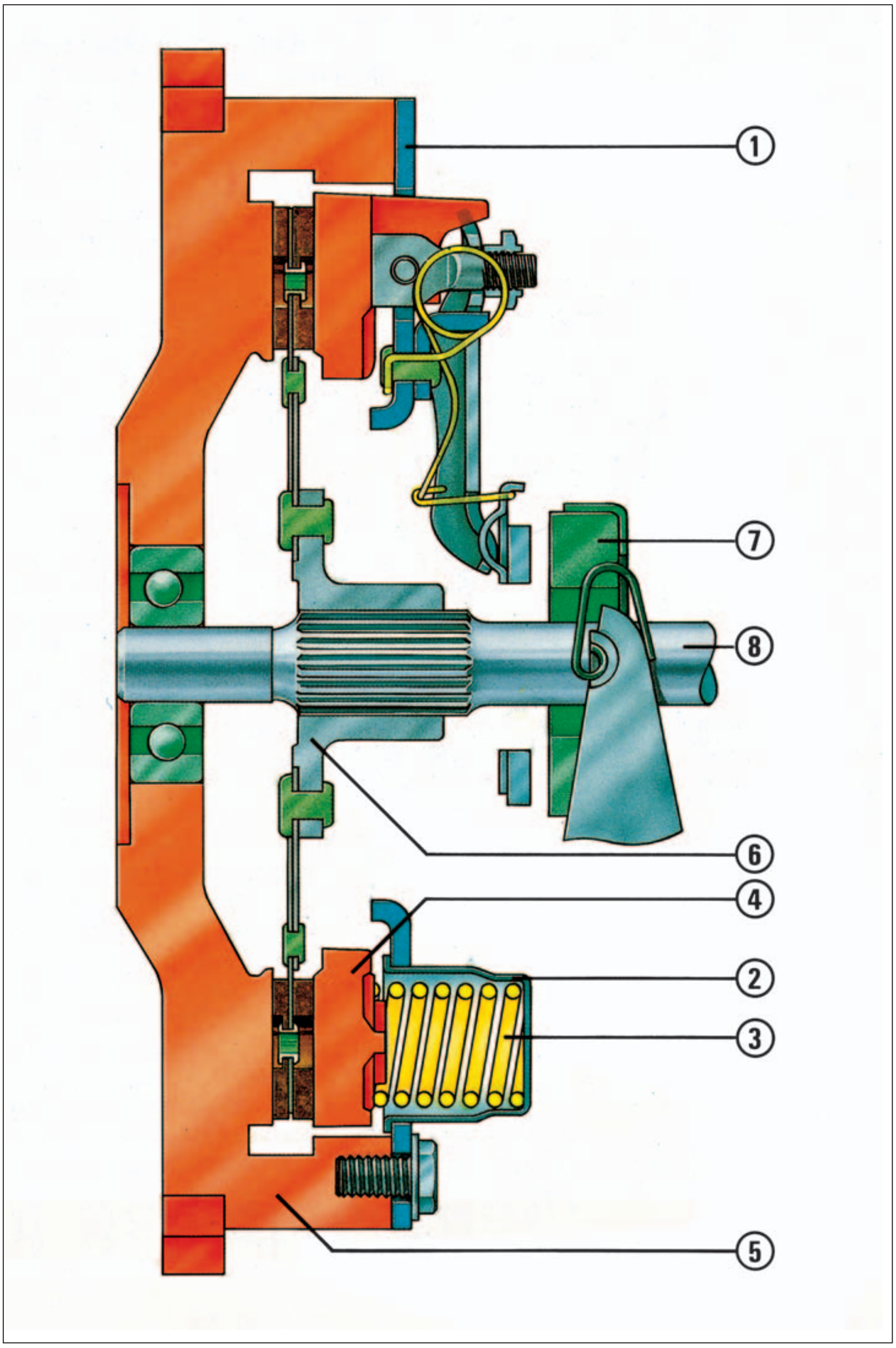
$$r_m = \frac{d_m}{2} = \frac{162\,mm}{2} = 81\,mm$$
$$= 81 \times 10^{-3}{}_m \text{ mean facing radius}$$

Coefficient of friction μ

0.27 – 0.32 (for organic facings)
0.36 – 0.40 (for inorganic facings)

M_d (81 x 10^{-3} m) x 2 x 0,27 x 3.500 N.
M_d: = 153 Nm

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

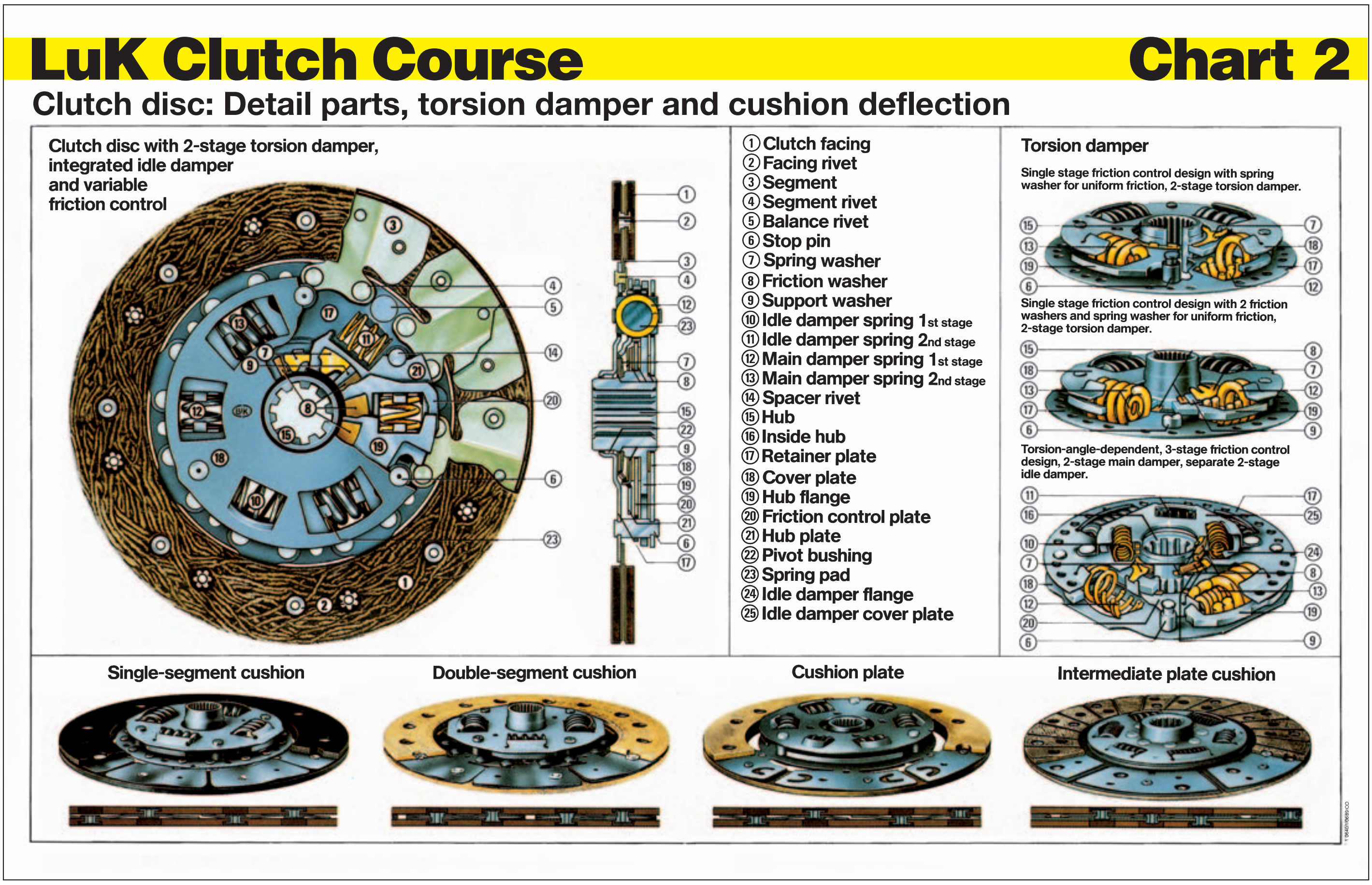


Coil spring clutch

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).

Whereas 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.

Clutch plate – Designs, torsion damper and



The clutch plate is the central connecting element of the clutch. It forms a friction system between the engine flywheel and the clutch pressure plate. With the clutch engaged, it is pressed between the flywheel and clutch pressure plate and actuated by friction. It transmits the drive torque to the transmission input shaft through the hub splines.

In modern vehicles, only clutch plates with torsion dampers and spring-loaded facings are used. Organic friction facings are used almost without exception in car construction. Metal-ceramic sintered facings are used primarily for special vehicles and tractors.

The diagram on the left shows a typical car clutch plate with a two-stage torsion damper, integrated idle damper and a variable friction control assembly.

Its components are: The clutch facings (1), which are riveted to the spring segments (3) with the facing rivets (2). These spring segments are fastened to the retainer plate (17) with rivets. The retainer plate (17) is aligned to rotate on the hub by means of the pivot bush (22).

The torsion damper is composed of the idle damper (with springs 10 and 11), the main damper (with springs 12 and 13) and the friction control assembly (friction washer 8, and friction control plate 20, spring washer 7, and support washer 9).

Torsion damper

Torsion dampers have the task of damping vibrations between the engine and transmission.

Unlike electrical motors and turbines, internal combustion engines do not deliver a constant torque. The constantly changing angular speeds of the crankshaft produce vibrations that are transmitted via the clutch and transmission input shaft to the transmission, where unpleasant rattling noises are the result. Torsion dampers are designed to minimise these vibrations between engine and transmission.

The constant reduction in flywheel mass and the lighter construction of modern vehicles increase these undesirable effects. Accordingly, every vehicle today must be subjected to special tuning, which has led to a wide variety of dampers and designs. Illustration 2 therefore shows only a few typical designs.
On the right of the illustration, three types of torsion damper are displayed.

They operate according to the following basic principle:
The hub (15), supported on bushes between the drive disc (17) and the retainer plate (18), is spring-loaded via the hub flange (19) and the damper springs (10-13) against the drive disc and the retainer plate, so that under load large or small angular motion is achieved. The spring compression is dampened by a friction assembly (7, 8, 9, 20). The transfer torque of the damper must always be greater than the engine torque, to prevent the hub flange (19) from striking against the stop pin (6).

In modern vehicle construction, two- and multiple-stage characteristic curves are often required. The stages are produced by springs with various spring rates and variously sized windows. The friction assemblies also differ largely due to different friction and spring washers. The characteristic curves are usually not symmetrical, but display in the direction of drive a steeper line with a higher stop torque than in the 'coast'or 'overrun' direction (for further details, see "Designs and Torsion Damping Diagrams" in the commentary to chart 3).

The upper torsion damper has a simple friction device with a friction washer producing constant friction and a two-stage characteristic curve. The hub flange (19) runs between the drive disc (17) and retainer plate (18), and is supported by the main damper springs of the 1st stage (12) and 2nd stage (13). The hub flange (19) can be turned up to 16 degrees against the drive disc (17) and retainer plate (18) before striking the stop pin (6). In this way the coil springs lying in the windows of the clutch and retainer plates, which have different spring rates, are tensioned. Vibration is converted to friction through the spring washer (7).

The middle torsion damper is designed similarly to the upper one, but is additionally provided with two friction washers (8). They are made of either organic material or plastic. Organic friction washers offer higher frictional coefficients, while plastic friction washers provide less friction, but excellent wear resistance.

The lower torsion damper has a 3-stage friction assembly dependent on the torsion angle, a two-stage main damper and a separate two-stage idle damper. The separate idle damper, consisting of an idle damper flange (24) and idle damper retainer plate (25) with idle damping springs of the 1st stage (10) and 2nd stage (11), is primarily used in cars with diesel engines. It acts at lower engine torque's and dampens during idling. The three friction washers (8) of the 3-stage friction assembly begin to act at different torsion angles. The 2-stage main damper (12) and (13) operates similarly to the above-described systems.

The facing cushion springs (segments)

The lower part of the illustration shows the four most common types of facing springs. The facing springs lie between the clutch facings. They ensure gentle clutch engagement and hence smooth take up of drive. The pressure plate of the clutch must at first press the clutch plate against the flywheel, acting against the spring load of the facing springs. As this load builds up slowly and draws out the engagement process, the transmission speed can be adjusted to the engine speed with a delay by the flexing of the plates. Besides smooth take off, good wear resistance, a better wear pattern and, a more uniform heat distribution are further advantages of the facing springs. We basically distinguish between four types of facing cushion spring systems (from left to right):

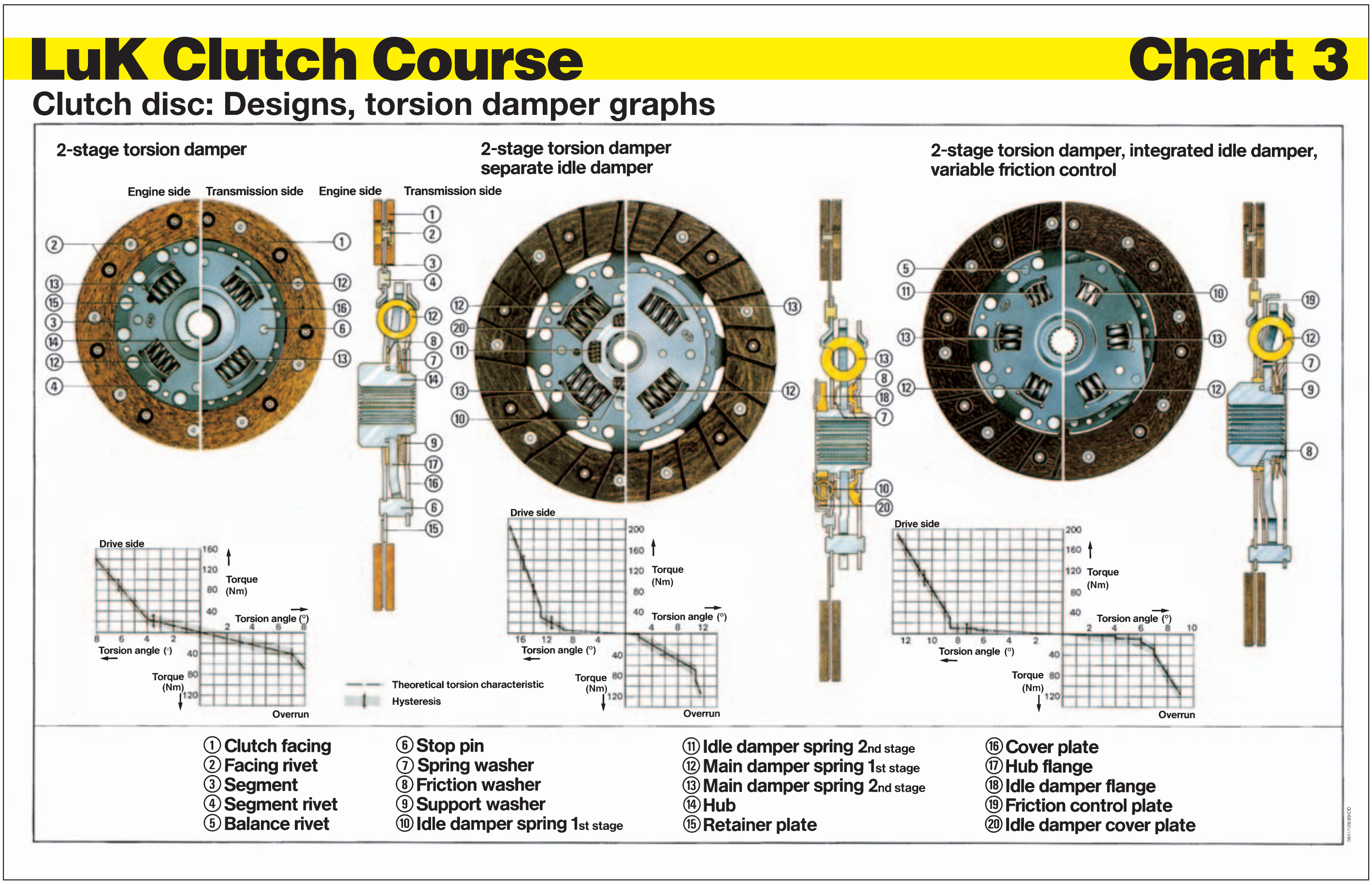
Single segment cushion spring, in which the facings on both sides are riveted to thin, bowed segments that in turn are riveted to the drive disc. The advantages are the small flywheel effect of the disc and the easier operation of the cushioning

With a **dual segment cushion spring,** the facings are riveted to two segments lying on top of one another and acting in opposite directions. As with single segment cushion spring, the segments are riveted at the drive disc. The advantage of better cushioning and better utilisation of the available cushion space is offset by the disadvantages of a greater flywheel effect and a higher price.

Multi-plate cushion spring is the most common version. The carrier plate of the facings is slotted and corrugated at the outer edge, where the facing lies. It functions in essentially the same way as the single segment cushion spring and is primarily used where there is no room for riveting the single segments to the carrier plate. With more heavily stressed clutch plates, the comparatively thin carrier plate in the area of the torsion damper must be reinforced by a second counterplate.

Intermediate cushion plate spring is generally used for heavy commercial vehicles. The segment-like, corrugated panels are riveted on one side of the carrier plate that is extended to the outer edge. They therefore act only along one direction. A disadvantage is the large flywheel effect of the disc.

Clutch plate – Designs, torsion damper graphs



Tasks:

As the friction 'mate' between the flywheel and the pressure plate, the clutch plate has the primary task of transmitting the engine torque to the transmission input shaft.

The main components are:

- drive disc (15)
- clutch facings riveted in pairs (1)
- splined hub

As already shown in chart 2, it also has several other tasks, which we briefly mention here again:

It must allow smooth take up and fast gear-changing, keep engine vibrations away from the transmission and thus prevent transmission noise generated by gear rattle.

The fulfilment of these tasks – an absolute "must" for the modern vehicles of today – requires a few additional components:

- segments (3)
- torsion damper (7–13)

Designs:

The designs are selected according to vehicle type and requirements. The mode of operation can be displayed by "torsion damping graphs", as shown in chart 3, below the clutch discs. The torsion angle of the torsion damper is plotted as a function of the given torque. The dashed/dotted line represents the theoretical torsional characteristic curve, while the shaded band shows the torsional characteristic curve when friction is taken into account (hysteresis).

Two-stage torsion damper

The left hand illustration shows a two-stage torsion damper. The four tangential windows contain coil springs (12, 13) with two different spring rates for the two stages. The springs lying opposite each other are identical.

The hub flange (17) lying between the drive disc (15) and retainer plate (16) can be turned against spring pressure. The drive disc (15) and retainer plate (16) are firmly connected to the stop pin (6).

Torque introduced via the drive disc (15) and retainer plate (16) is transmitted via the torsion springs to the hub flange (17) and thus to the input shaft.

As the springs alone cannot absorb vibrations, an additional friction assembly is required for damping. It is composed of the friction washers (8) lying to the right and left of the hub flange, the support washer (9) and the spring washer (7). The spring washer presses via a support washer (9) on to the right hand friction washer and also, via the fixed connection between the cover plate (16) and the retainer plate (15), on to the left hand friction washer lying between the retainer plate (15) and hub flange (17).

When the engine provides the torque, the two springs (12) are first compressed at the lower spring rate, i.e. damping stage 1 up to a torsion angle of 4 degrees. In this position, in the example shown, they receive a torque of 20 Nm.

From this point on, the two springs (13) of damping stage 2 are also engaged. They cause the torsional characteristic curve to climb more steeply up to the stop, at a torsion angle of 8 degrees and 140 Nm.

The torsion dampers are designed so that the maximum torque is well above the engine torque.

If the vehicle is decelerating, the 1st damping stage (12) acts up to an angle of 7 degrees and a torque of 40 Nm. From this point on, the 2nd damper stage (13) acts up to a torsion angle of 8 degrees, corresponding to a torque of 65 Nm.

Two-stage torsion damper, separate idle damper.

The principles previously described also apply to the design of two-stage torsion dampers with separate idle dampers, as shown in the centre of the illustration. Added is the separate idle damper (10, 11). It was previously used primarily for diesel vehicles. Thanks to the lighter construction and the resulting high level of vibration elimination, this design is also increasingly being used for petrol engines.

The torsion damper diagram is clearly different from the preceding one. The torsional characteristic curve stays flat around the zero position, to prevent the transmission gears from "rattling" especially with diesel engines at idle speeds. The 1st main stage (12) is applied at barely 10 degrees torsional angle and very low torque.

With this type of clutch plate, the idle damper (10, 11), which produce the flat characteristic curve at zero, is separately inserted in the retainer plate and the idle damper cover plate (20) riveted to it. The idle damper flange (18) is connected to the hub. The idle damper stage must therefore always be turned to the maximum, until the mechanism of the main damper stages (12, 13) described above is applied.

This clutch plate has a friction washer (8) between the hub flange (17) and the cover plate (16). The frictional force is produced by two spring washers located between the hub and retainer plate and between the hub flange (17) and the retainer plate (15).

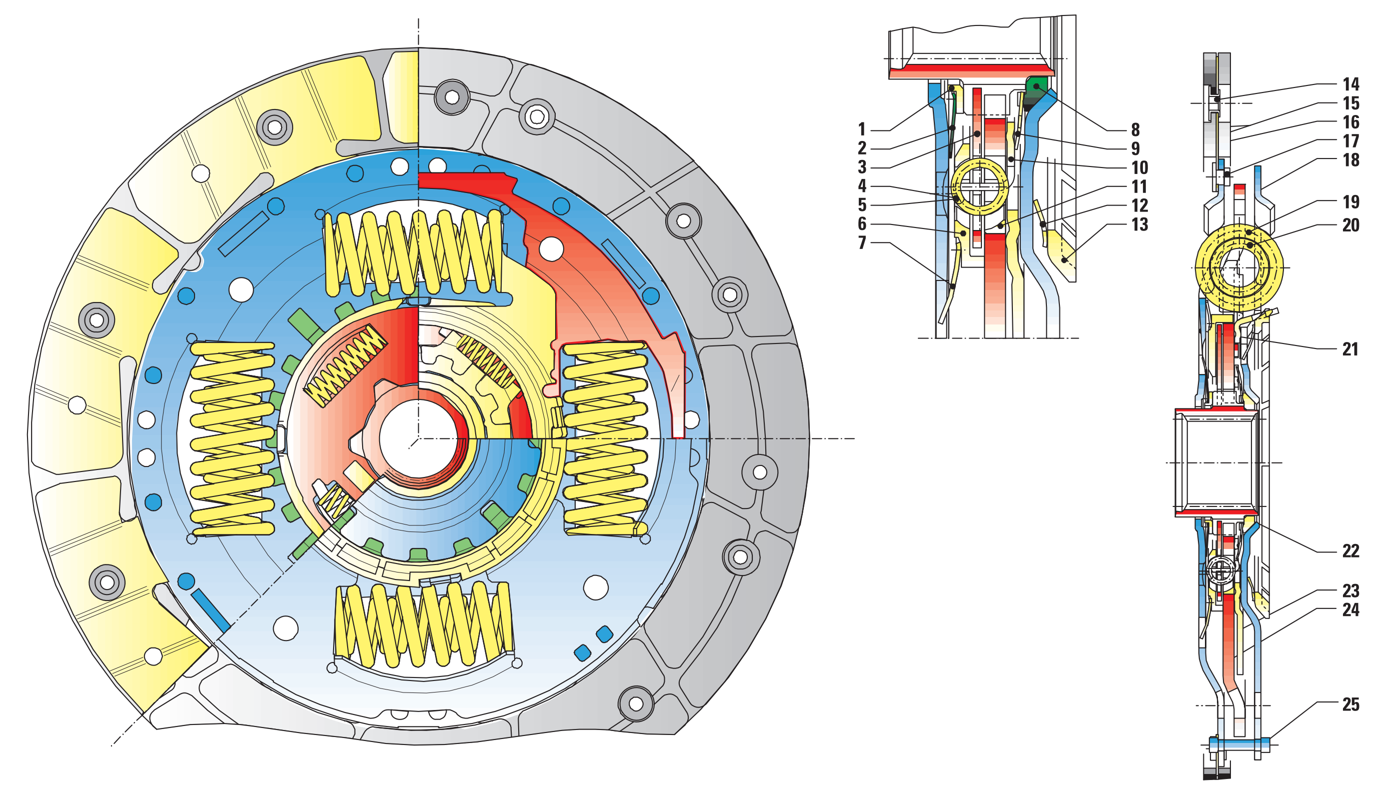
Two-stage torsion damper, integrated idle damper, variable friction control.

In the version illustrated on the right, the idle damper springs (10, 11) are located not in the clutch plate, but in the spring windows.

Whereas the friction was constant in the preceding versions, here it is made variable by the two separate friction washers (8), with two corresponding spring washers (7). One operates in the first main damper stage and the other in the 2nd main damper stage. They take effect only after the corresponding torsion angle is achieved (5 degrees and 8.5 degrees, respectively, on the 'drive' side and 1 degree and 7 degrees, respectively, on the 'coast' side).

The torsional characteristic curve and friction damping cannot be predetermined for a vehicle type. Extensive tuning, together with vibration computations for the vehicle are indispensable for optimising the torsional characteristic curves and friction damping.

Clutch driven plate – Bauarten, Torsionsdämpfungs



1 Idle damper friction washer
2 Idle damper diaphragm spring
3 Idle damper hub plate
4/5 Idle damper springs
6 Idle damper cage
7 Main damper first stage diaphragm spring
8 Aligning taper
9 Idle damper second stage diaphragm spring
10 Idle damper friction washer
11 Idle damper retainer plate
12 Main damper second stage diaphragm spring
13 Main damper friction washer
14 Facing rivet
15 Facing material
16 Spring cushion segment
17 Segment rivet
18 Main damper retainer plate
19/20 Main damper springs
21 Drive washer
22 Splined hub
23 Main damper friction washer
24 Main damper hub plate
25 Stop rivet

Fig. 1

The optimum for every application

When space or financial constraints prevent the fitting of a LuK Dual Mass Flywheel (DMF) or Damped Flywheel Clutch (DFC), the LuK clutch driven plate with an integrated torsion damper provides the ideal solution.

Every LuK clutch driven plate has the facing material fixed onto spring cushion segments. These offer improved comfort on take up of drive in addition to an ergonomically adjusted clutch pedal effort curve.

Multiple spring steel segments ensure that effective support of the facing material is achieved.

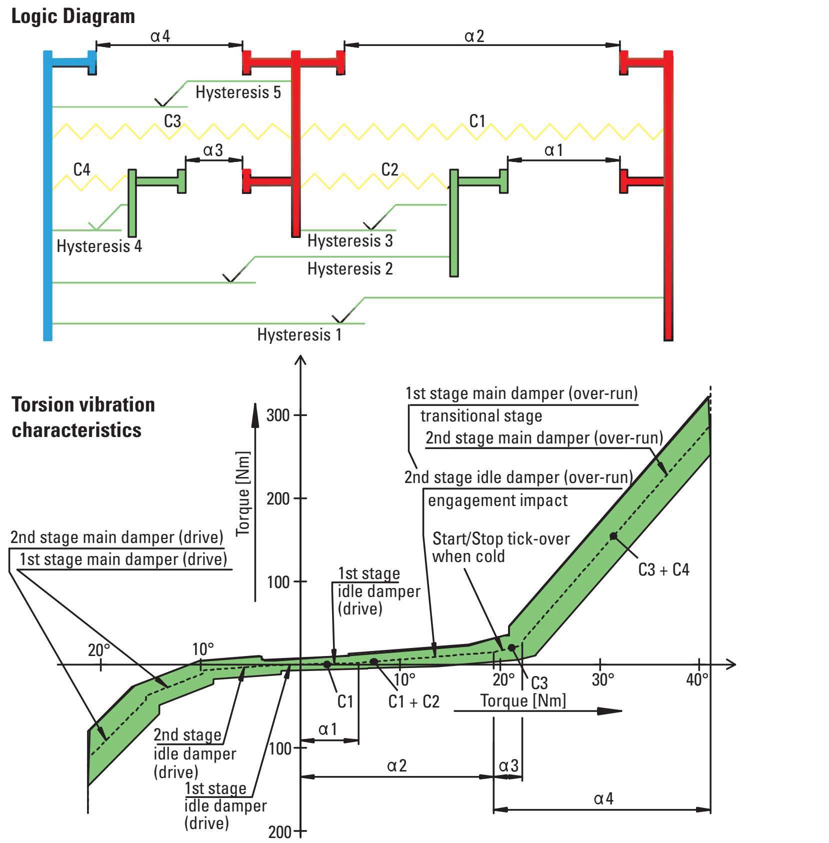


Fig. 2

Clutch driven plate with integrated torsion vibration damper

A driven plate with an integrated vibration damper is the most economical, yet space saving solution to torsional vibration within the vehicle drive train.

Driven plates are specifically engineered to each vehicle in order to comply with the size and weight constraints imposed by vehicle manufacturers. These constraints are set to assist in obtaining optimum fuel consumption and exhaust emission levels, in addition to delivering the level of comfort demanded by todays' driver.

Torsion vibration damping needs to be effective during all operating states and load levels, there is therefore a necessity to define friction damping (hysteresis) to take account of these different states and levels (see figure 2).

Torsion vibration damping can be tailored to meet the individual customers requirements. This can range from simple cost effective, single stage torsion vibration dampers, to complex four or five stage dampers engineered to provide optimum damping throughout all anticipated load conditions.

Self aligning-driven plate splined hubs have been developed by LuK to compensate for any possible misalignment between the engine and gearbox first motion shaft. This will ensure that the torsion vibration idle damper operates correctly during engine tick over.

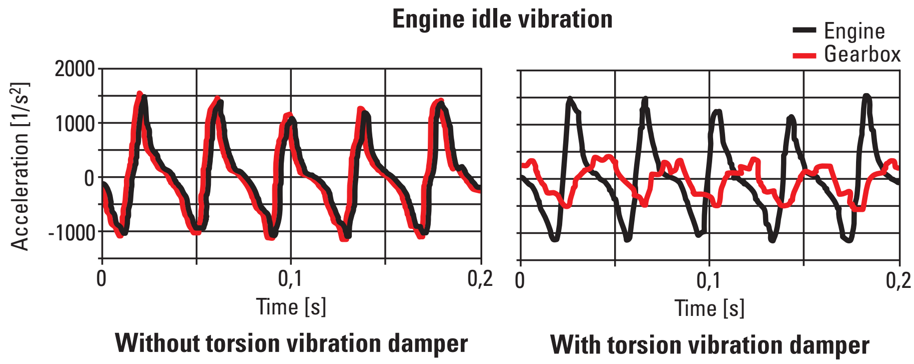


Fig. 3

Clutch driven plate without torsionvibration damper

A driven plate without torsion a vibration damper is used in conjunction with the most effective vibration damper system LuK has to offer, the Dual Mass Flywheel (DMF) or its derivative the Damped Flywheel Clutch (DFC).

Misalignment correcting clutch driven plate

As a result of the tolerances permitted during manufacture, particularly in the gearbox first motion shaft bearings, a slight misalignment condition occurs between the crankshaft and the first motion shaft.

When this misalignment occurs in conjunction with a rigid clutch driven plate as fitted with DMF or DFC it is possible that noise can be generated and in critical cases wear can become accelerated.

This problem is rectified by the use of LuKs' misalignment correcting driven plate. This allows a certain amount of driven plate hub deflection at engine idle and during low speed usage, thus avoiding any potential radial force on the hub (see figure 4).

This also ensures the DMF or DFC can operate efficiently even with a misalignment condition present during idle.

Adjustment and simulation

The most advanced technology available is used to assist engineers with extensive experience in refining the characteristics of each individual driven plate to the vehicle manufacturers requirements.

Vehicles that require a clutch to meet specific criteria are fitted with special driveline sensors. Readings from these sensors are taken and a sample drive plate is created, this sample is then compared with the measurements taken from the driveline (figure 3).

After theoretical parameter variations to determine the best possible characteristics, constructive examination of the units functionality and trials in prototype vehicles, a torsion vibration damper specifically produced to meet the customers target requirements becomes available.

Specifically adjusted idle dampers also offer a good level of vibration suppression which allows a lower engine idle speed to be achieved, this contributes to lower fuel consumption and emission levels to be achieved.

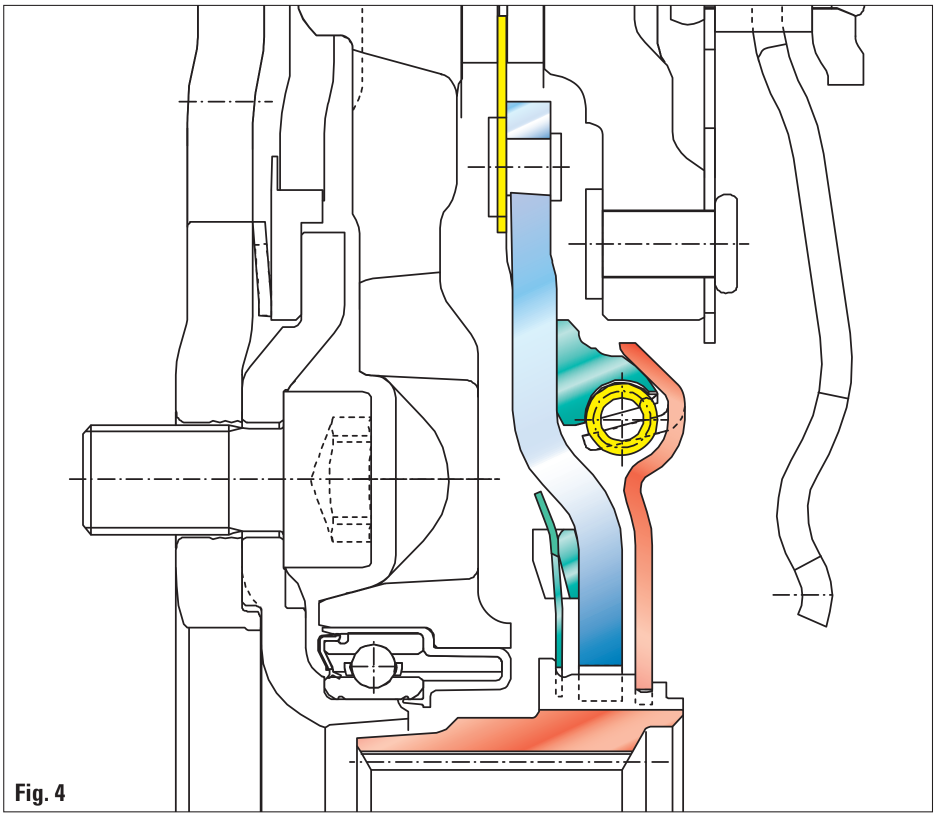
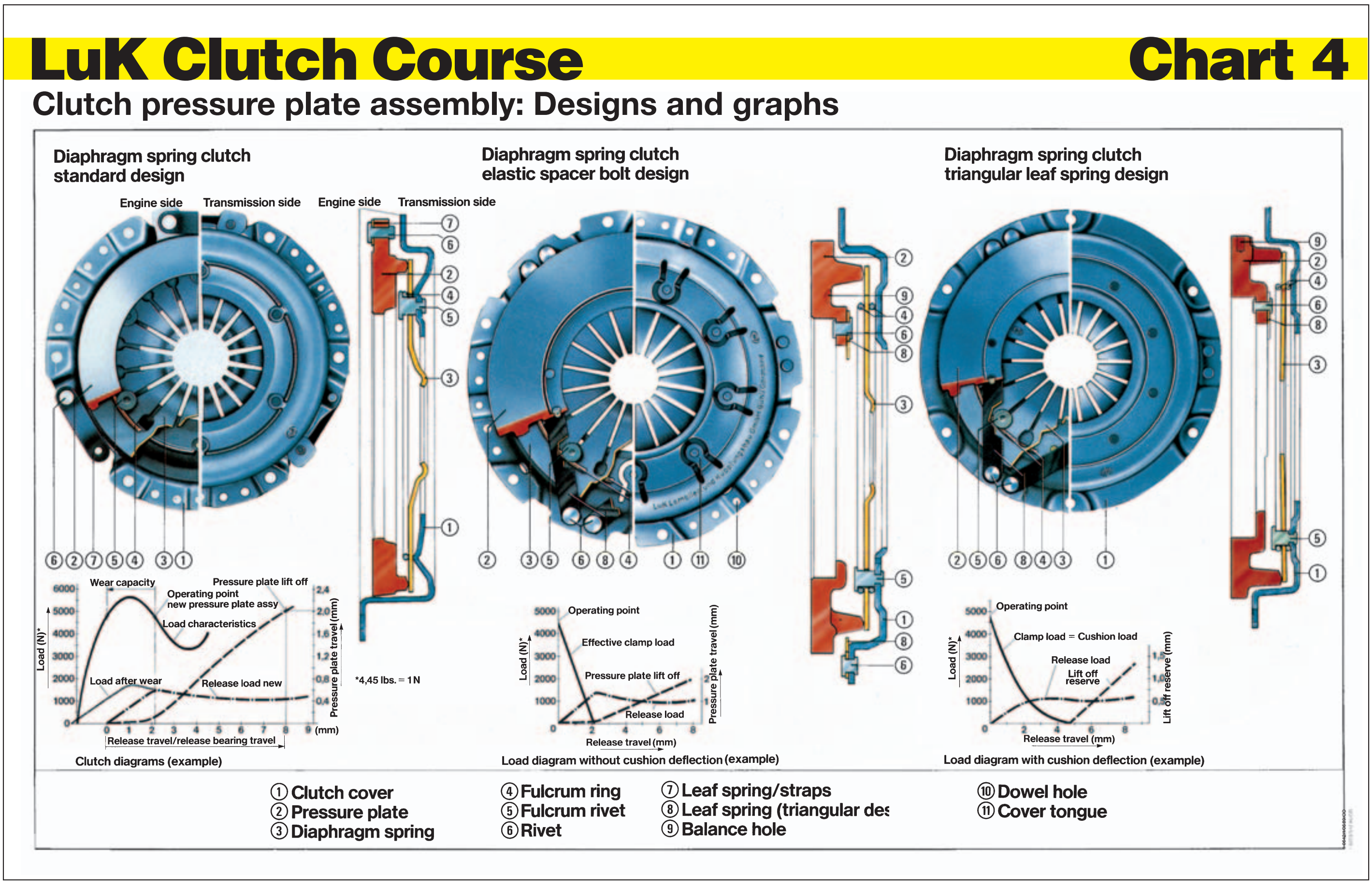


Fig. 4

Clutch cover assembly – Designs and graphs



Tasks

The clutch cover assembly forms a frictional system together with the flywheel and the clutch plate. It is secured to the flywheel via the bolts of the cover pressing and acts to transmit the engine torque via the clutch plate 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 previously 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 by fulcrum rivets (5) and/or fulcrum rings (4) on the cover. The diaphragm spring (3) presses the pressure plate (2) against the clutch facing. Tangential leaf springs (7) or triangular leaf springs (8) form a connection between the cover (1) and the pressure plate (2).

The balance hole (9) is made to compensate for the imbalance of the pressure plate (2). Dowel holes (10) assist in aligning the cover (1) to the flywheel.

The diaphragm spring

A central component of all these designs is the diaphragm spring. It is made flatter and lighter than the coil springs. Especially important is the characteristic curve of the diaphragm spring, which clearly differs from the linear curve of a coil spring.

Precise design of the diaphragm inside and outside diameters, thickness, opening angle and material hardness allow a characteristic curve to be produced as represented by the continuous curve in the left-hand diagram in chart 4.

While the clamp load with a coil spring clutch linearly decreases with decreasing facing thickness due to wear, here it initially increases and then drops again. The design is selected so that the clutch begins to slip before the wear limit of the facing is reached. The necessity of a clutch change is thus signalled in due time, so that further damage, e.g. by an ingress of the facing rivets, is avoided. Moreover, because of the diaphragm spring characteristic curve the requisite pedal force is less than with coil spring clutches.

Designs

Diaphragm spring clutch, standard design.

The left-hand diagram in illustration 4 shows the standard version of a diaphragm spring clutch. The clutch cover (1) encloses the diaphragm spring (3) and pressure plate (2). The pressure plate (2) is connected to the clutch cover via tangential leaf springs (7). They are riveted to the pressure plate (2) at three lugs. The tangential leaf springs (7) perform three basic functions:

- lifting the pressure plate during disengagement
- transmitting the engine torque from the cover to the pressure plate
- aligning 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 plate between the flywheel and pressure plate (1). In doing so, it is supported by a rib in the clutch cover (1) and by a fulcrum ring (4). The outside diameter of the diaphragm sits on the pressure plate (2). If the clutch is actuated, the release bearing presses on the ends of the diaphragm spring fingers (3) – the pressure plate (2) is lifted and the clutch plate is disengaged

Diaphragm spring clutch with a triangular leaf spring design.

First, observe the version presented on the right in illustration 4. It essentially differs from the standard design by having a different type of connection between the clutch cover (1) and the pressure plate (2). As the design does not allow lugs to be attached to the pressure plate (2), because of a pot or recessed flywheel, a triangular leaf spring arrangement was chosen.

The leaf springs are riveted at both ends to the clutch cover (1), with the pressure plate fastened to the centre of each leaf spring.

Instead of the cover rib as a support and fulcrum ring for the diaphragm spring (3), an additional steel fulcrum ring (4) is used here.

Diaphragm spring clutch with a keyhole cover design.

The latest design is the diaphragm spring clutch with keyhole tabs shown in the centre of illustration 4. The keyhole tabs are formed so as to pull the rivets (5) outwards. As a result, the diaphragm spring (3) is constantly held in position despite wear which will occur at the diaphragm spring seats. The advantage; uniform lift throughout the entire life of the clutch.

Clutch characteristic curves and load diagrams.

The lower part of illustration 4 shows some examples of clutch characteristic curves and load 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 abscissa represents the release travel – in the diagram on the left shows the release bearing travel as well – and the y-axis on the right gives the lift of the pressure plate.

The continuous curve in the diagram on the left represents the development of clamp load. With a newly installed clutch plate. (operating point: new pressure plate assy).As the facing begins to wear the clamp load increases to it's optimum level.

The clutch plate thickness decreases by about 1.5 to 2 mm during its lifetime. The clamp load is calculated so that the clutch begins to slip shortly before the rivets of the clutch facing run against the pressure plate or the flywheel and thereby avoiding additional damage.

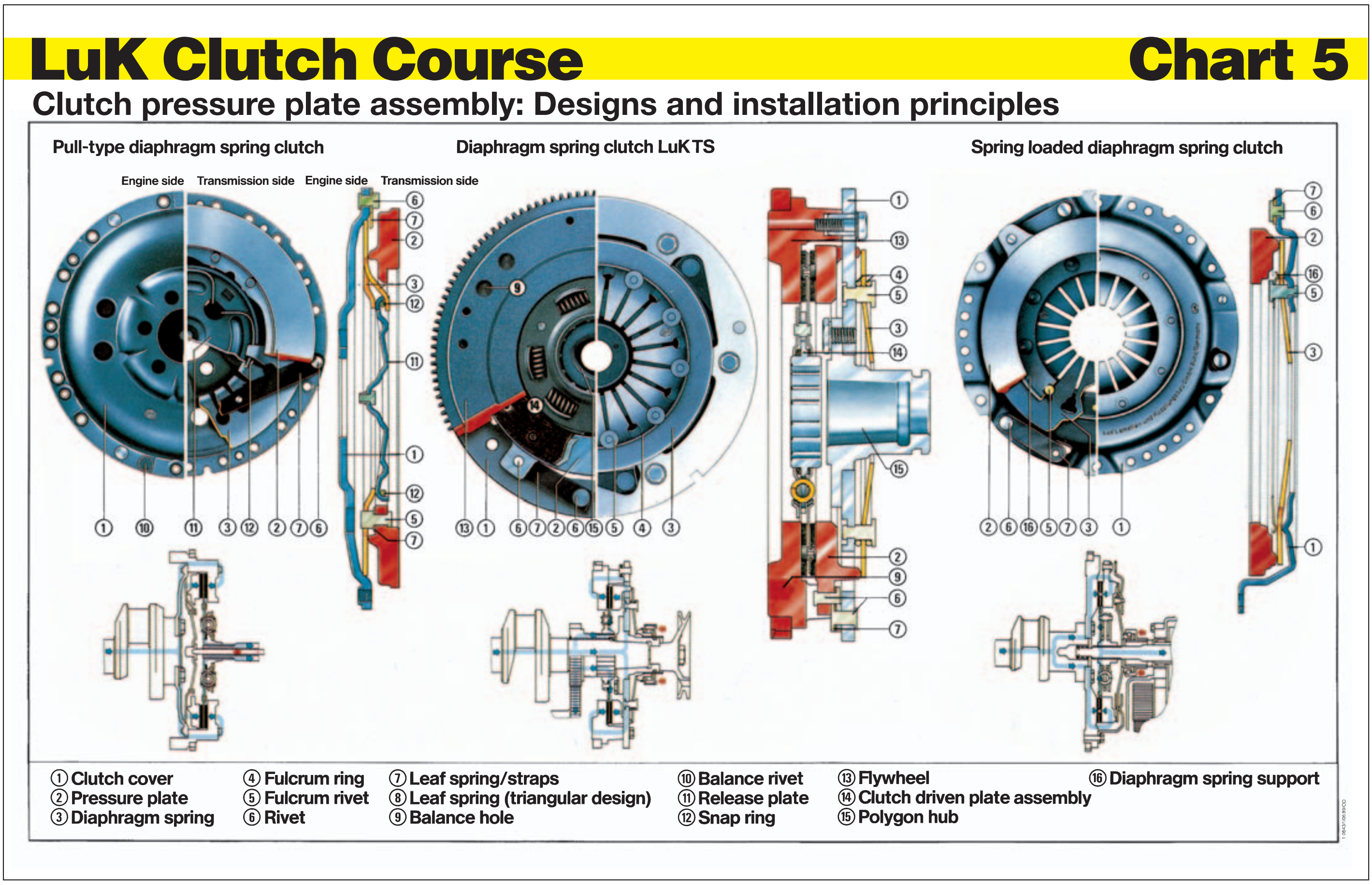
The dashed/dotted line shows the development of the release load, i.e. the load required to actuate the new clutch and (compare the dotted line) that load after facing wear. The release load initially increases until the operating point is reached, and then slowly drops again. The curve for the release load with facing wear has been moved to the left for better representation of the ratio of clamp load to release load. The larger the clamp load at the operating point the greater the release load.

The dashed line shows the lifting of the pressure plate above the release bearing travel. In this case the lever ratio in the clutch is clear: 8 mm release travel corresponds to 2 mm lift, i.e. to a ratio of 4:1 (not considering the elasticity of the clutch). This also applies to the ratio between clamp load and release load.

In the centre and right-hand diagrams, the measurements are contrasted for clutches with and without considering the spring compression of the clutch plate cushion springs. The commentary to illustration 2 already noted the advantages of cushion springs, such as gentle clutch engagement and more favourable wear characteristics. Without the cushion springs, the effective clamp load (solid line) falls off 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 load, along which the clamp load diminishes, is about twice as great. On the other hand, with the clutch engaged, the clamp load slowly increases along a curve, as the cushion spring must first be compressed. Thanks to the relatively gentle climb of the clamp load curve (solid line), the pronounced load peak at the required release load is reduced. As long as the pressure plate (2) still makes contact with the clutch plate, the clamp load and cushion spring correspond to one another.

Clutch cover assembly – Designs and installation



Diaphragm spring clutches are used almost exclusively in automotive construction today. The coil spring clutches used so frequently in the past have practically disappeared, owing to a number of disadvantages, but especially because of their considerably larger installation space requirement and greater weight.

The most important advantages of the diaphragm spring clutch compared to the coil spring version are:

- insensitivity to high engine speeds
- despite the small space, clamp loads are achieved at low release loads
- the diaphragm spring fingers also function as release levers
- fewer wearing parts

The diaphragm spring also makes a big difference for the driver, as the lower release load means that only low pedal efforts need be applied.

Depending on the design and type of clutch actuation, we distinguish between:

- pulled diaphragm spring clutch
- spring loaded diaphragm spring clutch

Pulled diaphragm spring clutch

The left clutch in chart 5 was specially designed for the VW Golf and Jetta. With respect to the support position of the diaphragm springs, one speaks of a pulled clutch; of course, because of the reversed installation compared with the usual design, actuation is only through pressing. Usually the drive proceeds from the crankshaft to the flywheel and then to the clutch and transmission. Here, however, the clutch is initially bolted to the crankshaft. The flywheel is attached after the clutch plate is fixed and then connected to the clutch.

This construction constrains the design of the clutch in the following way: the outer edge of the diaphragm spring (3) is supported by the clutch cover (1) and the inner edge by the pressure plate.

A reversal of the diaphragm springs, as with standard clutches, does not occur during disengagement. The diaphragm spring (3) is simply lifted via the thrust plate (11), which is inserted in the diaphragm spring points. The thrust plate is actuated via a push rod supported by bearings in the hollow transmission input shaft and extending to the rear of the transmission end, where the release bearing and release arm are located.

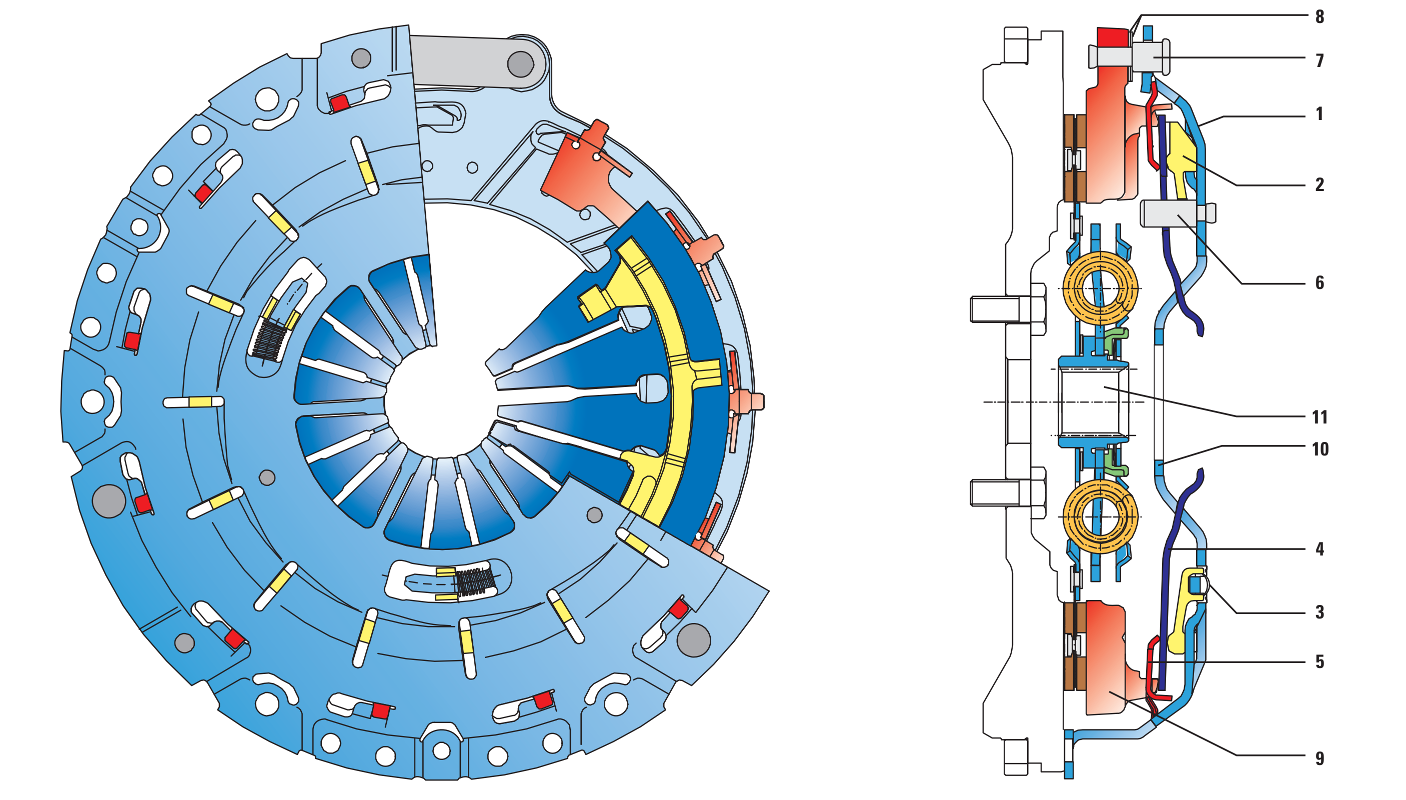
Diaphragm spring clutch LuK TS

The diaphragm spring clutch LuK TS is a pressed steel clutch. It is recognised by the integration of clutch and flywheel. The polygonal hub (15) of the clutch is bolted together with the V-belt pulley to the crankshaft, which has the corresponding matching taper.

The drive line is through the clutch cover (1) to the flywheel fastened to it. The pressure plate (2) is seated between the clutch cover (1) and clutch plate (14). It is connected via tangential leaf springs (7) to the clutch pressure plate (2). The seats of the pressure plate (2) protrude through the openings in the clutch housing (1). The outer diaphragm spring is supported by these seats, and is mounted at the housing by the fulcrum rivets (5) and fulcrum rings (4). The release bearing is mounted so as to slide on the outside diameter of the polygonal hub. The torque is transmitted via the clutch plate (14) to the transmission input shaft, formed as a hollow shaft and positioned on the crankshaft end – between the clutch and the engine. In this way, the transmission could be integrated into the lower half of the engine.

Spring loaded diaphragm spring clutch

The diaphragm spring clutch with second steel pressing is a special version. Here the fulcrum rings have been completely replaced by a rib in the clutch cover (1) and by a sprung second steel pressing (16) formed as a further fulcrum. In this way, any wear can be compensated for and automatic re-adjustment is achieved. Otherwise this design does not differ from those shown in illustration 4.



As has already been mentioned on page 18, the diaphragm spring clutch is almost exclusive to modern vehicle construction.

Further technological development on this component has been pushed ahead vigorously in the past few years (e.g. diaphragm spring key hole cover; for description, see pages 17 and 18), and has now evolved into the current new development, the SAC clutch.

The abbreviation SAC stands for Self-Adjusting Clutch.

Higher performance engines of the type gaining ground nowadays also need clutches with higher-transmission moments, this means that pedal force has also increased. It may have been possible to keep this increase within certain limits by means of various measures (such as improved release systems), but, despite this, the demands are steadily growing for clutches with reduced actuation force.

The most important advantages of this type of design in relation to previous formats are:

- Low disengagement forces, which remain constant throughout the entire service life,
- Resulting in high levels of driving comfort throughout the entire service life,
- Increased wear reserves, and therefore longer service life thanks to automatic wear adjustment,
- The over-travel of the release bearing is limited by the diaphragm spring stop.

This in turn results in a whole range of secondary advantages:

- No further need for servo systems (on utility vehicles)
- Simpler release systems
- Shorter pedal travel
- Uniform pedal forces across the entire range of engines
- New possibilities for reducing clutch diameters (torque transfer)
- Shorter release bearing travel throughout the service life

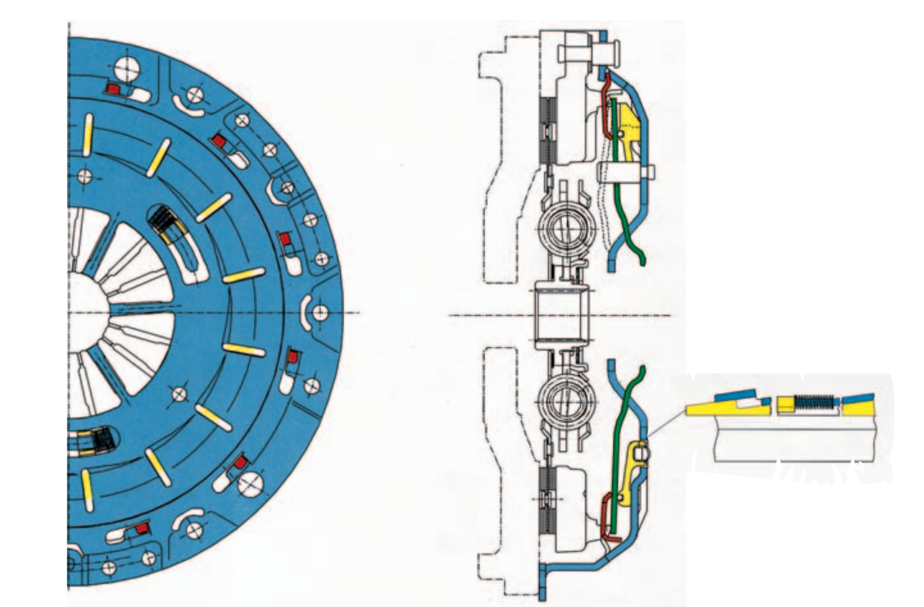


Figure 2: self adjusting clutch (SAC)

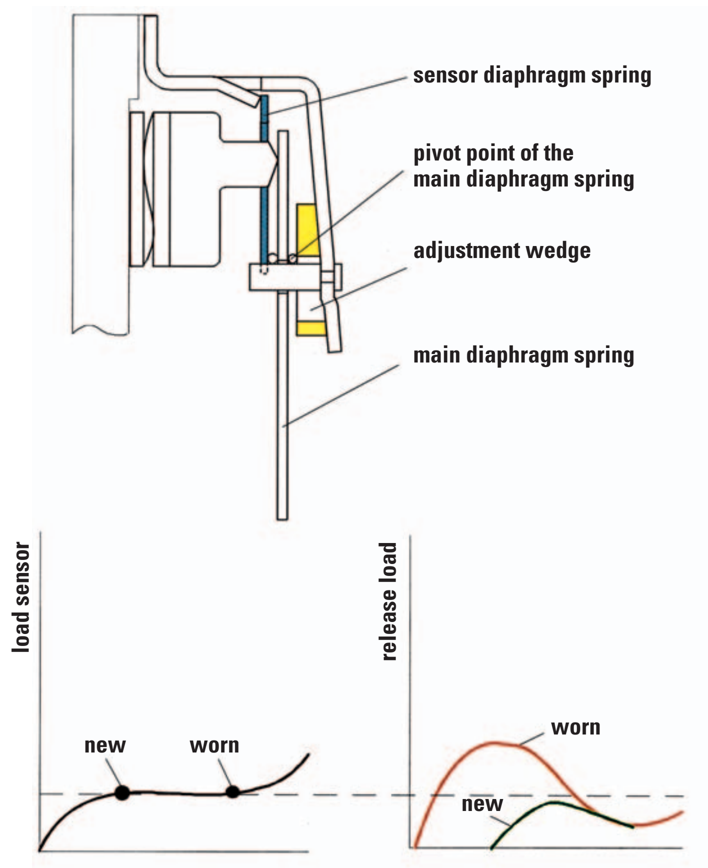


Figure 1: principle of the self adjusting clutch (SAC)

Principle of the self adjusting clutch (SAC):

The LuK wear adjustment feature works on the following principle: the load sensor determines the increased release load due to wear and correctly compensates for the reduction in facing thickness.

Figure 1 shows a schematic representation of the SAC. As opposed to 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 targeted release load. The pivot point of the main diaphragm spring remains stationary as long as the release load is smaller than the load of the sensor spring.

When facing wear increases the release load increases, the opposing load 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. Graphically, this means that the intersection point between the two curves has returned to its original location. When the sensor spring deflects, a gap develops between pivot point and cover, which can be compensated for by introducing a wedge-shaped component.

Design of a wear-sdjusting clutch with a load sensor.

The load sensor with the thickness adjustment wedge can be realized in a simple and elegant manner. Figure 2 shows such a design. 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).

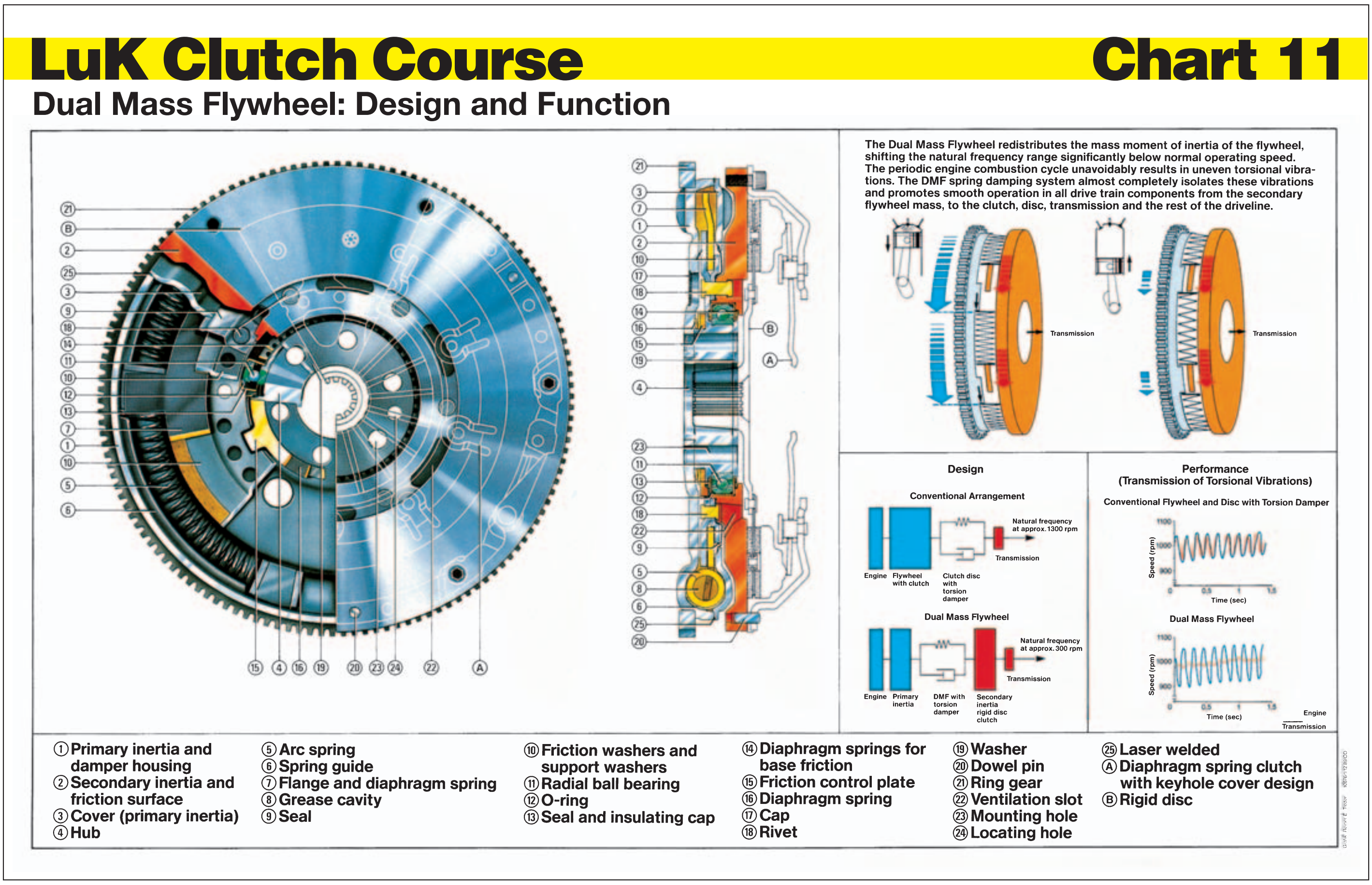
The sensor diaphragm spring is suspended in the cover. Its inside fingers position the main diaphragm spring. Because of centrifugal forces, the wedges that provide the actual adjustment are positioned circularly instead of radially. A plastic ring with twelve ramps moves on opposing ramps in the cover.

The plastic ring (adjustment or ramp ring) is circularly preloaded with three small coil springs which force the ring to fill the gap between the diaphragm spring and the cover when the sensor spring moves.

Figure 3 shows the release load curves for a conventional clutch in new and worn facing condition. In contrast, compare the 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 wear capacity, 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 4 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.

Dual-mass flywheel (DMF) – Design and function



The increase in noise sources owing to inadequate natural damping is a noticeable feature of modern automotive construction. The causes lie in the reduced weight of the vehicles and in the wind-tunnel optimised bodies, whose low wind noise now makes other noise sources perceptible. But also sleek body designs and extremely low-revving engines, as well as 5-speed transmissions and thin oils, contribute to this phenomenon.

The periodical combustion process of the reciprocating engine induces torsional vibrations in the drive train, which as transmission rattle and body noise interfere with driving comfort.

Design:

The division of the conventional flywheel into two parts results in a primary flywheel mass (1) with starter ring gear (21) on the engine side, and a secondary flywheel mass (2) with vent slots (22) for heat transfer, which increases the angular momentum on the transmission side.

The two masses are connected to one another via a spring/damping system and supported by a radial ball bearing (11) to allow free rotation. Sealing is provided by the O-ring (12) and the seal (13).

Two moulded sheet metal parts (1, 3) laser-welded to the outer edge (25) form the ring-shaped grease cavity (8), in which the curved compression springs (5) with spring guides (6) are located. Sealing is provided by the diaphragm (9).

The flange (7) in the form of a diaphragm spring engages with its lugs between the curved compression springs (5). It lies frictionally engaged between the friction and support rings (10) riveted on the secondary side. The diaphragm spring load is designed so that the friction torque is greater than the maximum engine torque.

An additional friction device (15, 16), bearing-floated on the hub (4), is carried by one of the retaining plates.

As the spring/damping system is integrated in the dual-mass flywheel, a rigid version of the clutch plate (B) is used without a torsion damper.

Usually a diaphragm spring clutch with a keyhole cover positioned via dowel pins (20) is used as a clutch cover (A).

Function:

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

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

As shown in the "function" diagram, with the DMF the angular moment is decreased in front of the torsion damper and increased behind it. The angular moment of the engine is now assigned to the primary mass of the DMF, while that of the transmission is assigned to the secondary mass including the clutch driven plate and the clutch pressure plate. In this way, the resonance speed is shifted from approx. 1300 rpm to about 300 rpm and can no longer interfere with driving comfort, as the engine is not operated in this speed range.

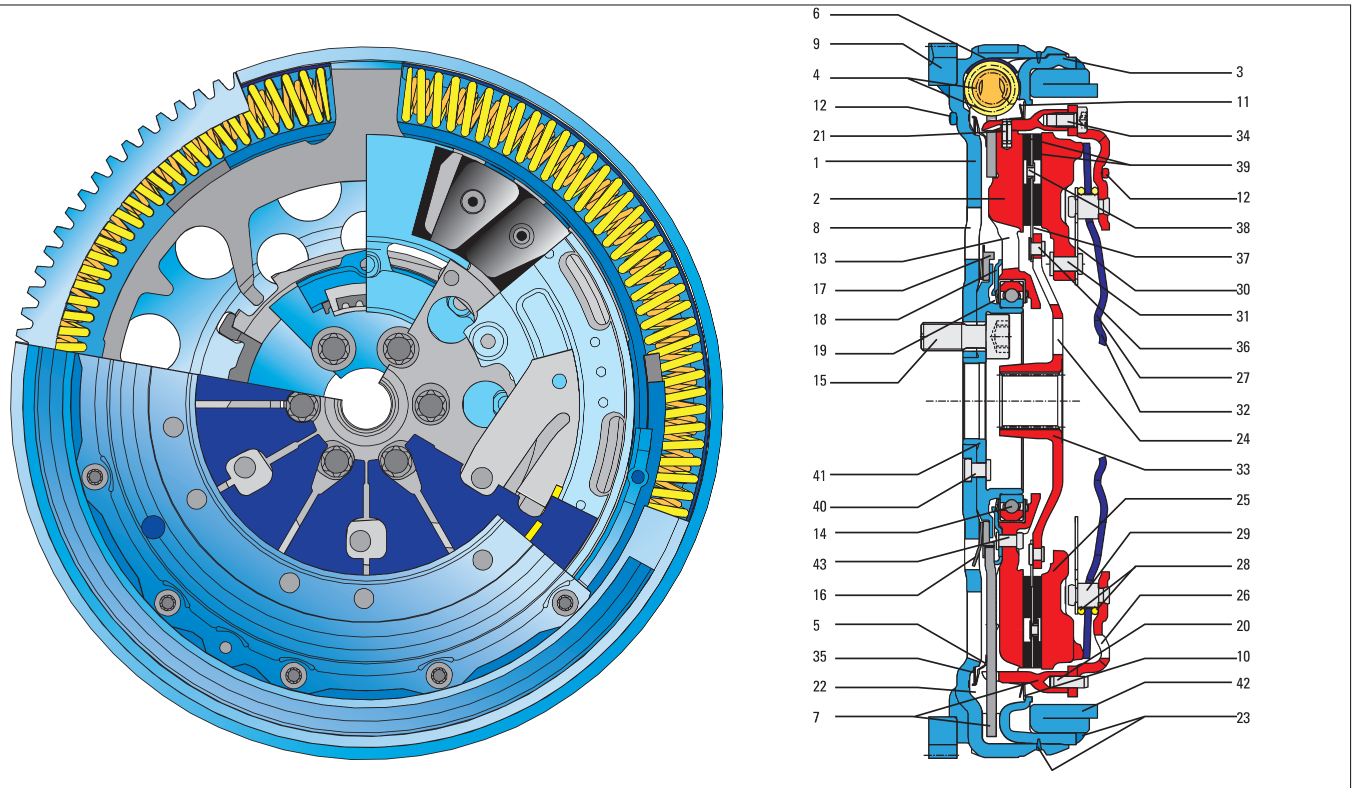
An added positive effect is provided by the reduced angular momentum on the engine side: Gear changing is improved thanks to the lower mass to be synchronised, and the synchromesh units are subject to less wear.
The effects on torsional vibrations can be seen from the diagram With previous conventional flywheels and torsion-damped clutch plates, the torsional vibrations in the idle speed range were transmitted to the transmission with the least possible filtering, causing the teeth of the transmission gears to strike against one another (transmission rattle).

The use of a dual-mass flywheel however, filters out the torsional vibrations of the engine by the complex construction of the torsion damper, preventing vibration from affecting the transmission components – rattling does not occur, and driver comfort is fully ensured.

The advantages of the LuK dual-mass flywheel at a glance:

- first-class driving comfort
- absorption of vibrations
- noise insulation
- fuel saving due to lower engine speeds
- increased gearchanging comfort
- less synchronisation wear
- overload protection for the drive train

DFC – Damped Flywheel Clutch – (Compact-DMF)



1) primary flywheelmass and damper housing
2) secondary flywheelmass with friction surface
3) cover (primary flywheelmass)
4) arc coil spring
5,10) gland seal
6) arc spring guide race
7) cover ring and flange
8,13,26) ventilation slot
9) starter ring gear
11) support plate
12) balance weight
14) deep grooved ball bearing
15) socket head cap screw
16,19,27,35) diaphragm spring
17) load friction plate
18) retainer plate
20) dowel
21) dowel pin
22) grease cavity
23) laser welding
24,32) access opening for removal/installation tool
25) pressure plate with friction surface
28) fulcrum ring
29) stretch bolt
30) leaf spring
31,40,43) rivet
33) hub
36) segment rivet
37) driven plate segment
38) lining rivet
39) linings
42) inertia ring

Layout and function

The dual-mass flywheel provides a highly efficient system for the damping of torsional vibrations in the drive train. It has proved its worth in the prestige vehicle sector.

The middle range sector and those referred to as compact vehicles, with transverse engines are steadily growing in demand. The demands for used engines and those with reduced pollutant emissions is also growing, but at the same time this is leading to increased irregularities in engine performance, especially in the direct-injection Diesel engine sector. LuK has now developed the DFC, to provide top-of-the-range driving comfort for the middle range sector.

There were two basic problems which needed to be overcome in this context:

1. The installation space in front transverse engine vehicles is very restricted.
2. The price structure on which this class of vehicle is based makes cost-optimised solutions essential.

The DFC is already effectively cutting out engine vibrations when idling; in other words, there is no more gearbox rattle, and the unpleasant drumming of the bodywork at certain engine speed ranges has become a thing of the past.

With regard to environmental protection positive results are also becoming apparent.

- Thanks to the excellent noise reduction at low engine speeds, there are fewer gear changes, as well as the average operating speeds of the engine dropping.
- The degree of efficiency of the system as a whole is boosted, and the fuel consumption and emissions of pollutants usually associated with this are reduced.

The DFC is an integrated assembly of dual-mass flywheel, cover assembly, and driven plate.

It is supplied complete to the vehicle manufacturer, including the crankshaft bolts, and can then be fitted as a unit in one operation on the vehicle assembly line. The crankshaft bolts can be tightened through drillholes in the driven plate and the cover assembly.

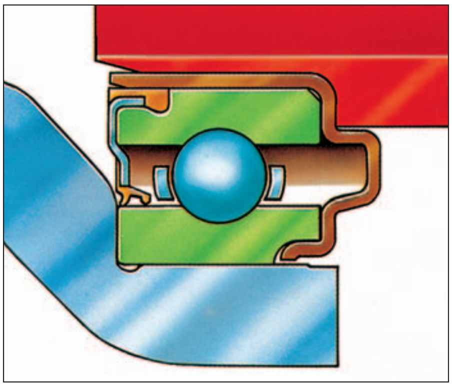
Other advantages in relation to conventional systems:

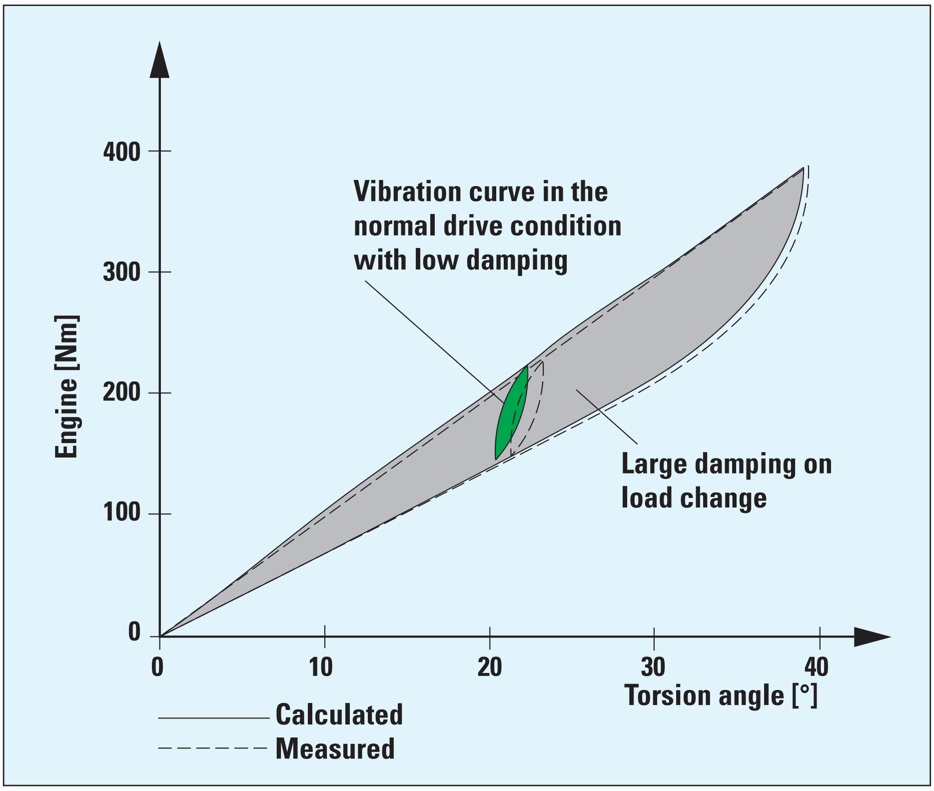
- Lower weight
- Less imbalance of the system as a whole
- Reduced set up height tolerances for the diaphragm spring.

Total technology

Clutch cover:

The pot-shaped primary mass (1) which forms the damper housing for the arc springs, the cover (3) that belongs to it, the secondary mass (2), and the cover ring (7) are deep-drawn from one single blank.
The secondary mass (2) and the pressure plate (25) are made of grey cast iron, which has extremely good thermal conductivity properties. The carefully designed ventilation and air-cooling system provides excellent cooling effect for the flywheel and the pressure plate.





Damper springs:

The arc spring damper, already in use in the dual-mass flywheel, is integrated in the DFC unit. The spring damping system is required to fulfil two contradictory requirements.

1. Under normal operating conditions, the irregular shape of the engine vibration curve means that there are only low working angles in the damper. In this operating range, low spring rates associated with low damping qualities are required to attain optimum vibration damping.

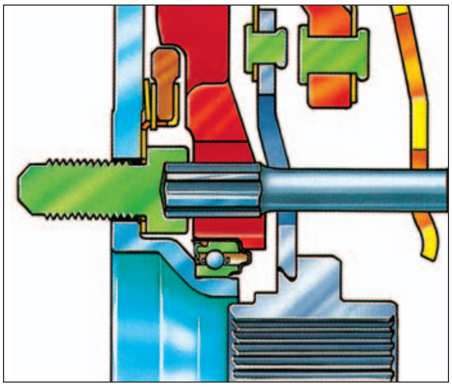
2. With the typical load changes (e. g. full acceleration), the engine vibration curve then changes, which to a considerable degree leads to the creation of noise. This effect can only be offset by a torsion damper, which has an extremely low spring rate and, at the same time, a high damping quality.

The arc spring resolves this contradiction, at high operating angles it provides a high damping effect at very low spring rates. At normal operating angles it provides a low damping effect at low spring rates, providing perfect isolation against vibration.

Bearings:

A special design of bearing makes it possible for it to be arranged inside the crankshaft bolt housing. The bearing (14) is subjected constantly to the rotational vibrations of the engine, with no further relative motion occurring between the inner and outer ring. It must also withstand peak operating temperatures. These operating conditions impose an extraordinarily high burden on the bearing.

The solution is an integrated bearing concept with special seals, which guarantees lubrication throughout the entire service life. A thermal insulation cap also resists even the most extreme operating temperatures.



Installation and logistics:

The DFC drastically reduces the effort and expenditure needed for installation during vehicle manufacture, as well as in the workshop.

The cover assembly can be removed from the unit, in order to replace the driven plate if necessary (oil contamination etc.).

A further advantage is the arrangement of the components: All the parts are matched to one another, and are, as a rule, delivered and replaced complete.

The number and variety of types can be reduced dramatically, which accordingly cuts down on logistics and overall expenditure (parts stocks, catalogue applications, etc.).

ECM – Electronic clutch Management

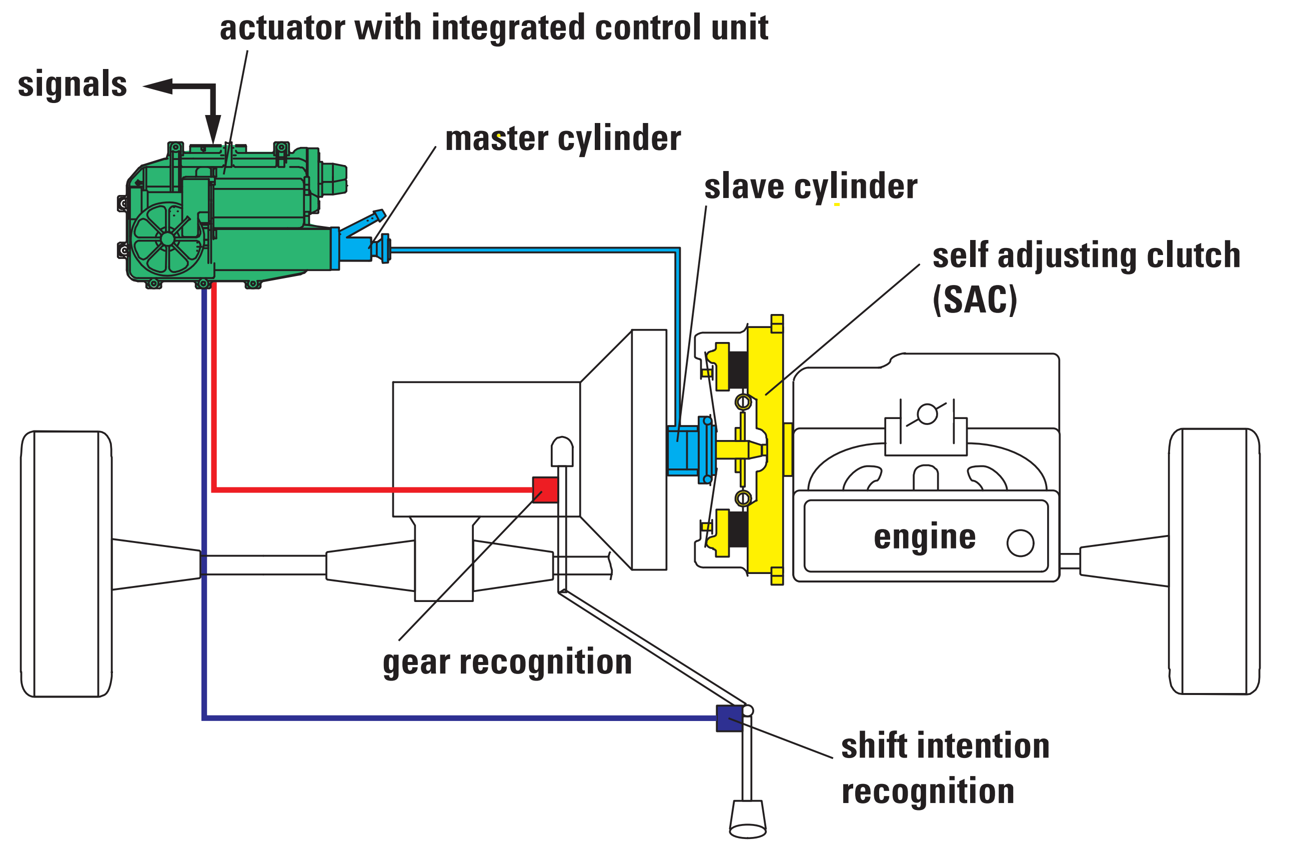


Fig. 1

With the growing traffic density and increasing comfort requirements the automation of the drive train will gain importance in vehicles. One milestone in this direction is the Electronic Clutch Management (ECM) system from LuK. With the aim of setting a new standard, LuK developed an ECM for mass production which is at present the most compact and most comfortable system world wide. During the development more than 4 million kilometres of test drive and more than 30,000 hours on test benches were conducted to ensure maximum reliability and functionality. The system is now in mass production since 1997.

The LuK-ECM-System is designed as an add-on system. In co-operation with the car manufacturer it can be easily adapted to the vehicle.

Function of the ECM

With an ECM the driver can shift as usual but doesn`t have to operate a clutch pedal. The actuation of the clutch during starting, shifting and stopping is done by means of an electronic actuator in an optimum manner. This means more comfort and increased safety due to the relief of the driver, and it also creates more pleasure in driving with a manual transmission.

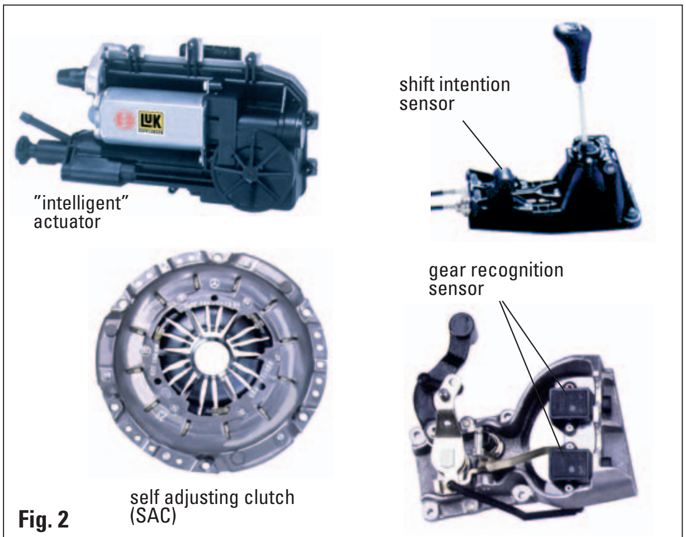


Fig. 2

Components of the LuK ECM

Assembly and special features of the LuK-ECM:

Figure 1 shows the principle layout of the ECM.

The basis of this system is the self adjusting clutch (SAC), developed by LuK. At the same torque capacity the release load of the SAC is about 30% lower compared with a conventional clutch, and due to the self adjustment during wear and tear, this release load remains constant over lifetime.

The SAC in combination with intelligent control strategies like the so called **"torque tracking strategy"** makes it possible to use a very small electric motor for the clutch actuation. Because of the low heat generation of this small electric motor the actuator and the control unit can be combined into an "intelligent actuator" **(figure 3)**.

A very important target during the development of the ECM was to minimise the effort for the car manufacturer. The system had to be a pure add-on-system and changes on the transmission and shifting mechanism had to be avoided. Under these preconditions, a clutch position sensor and a speed sensor for the transmission input shaft could not be considered (see also **figure 4**). Consequently modifications to the transmission, the release-system and the corresponding wiring are not necessary. Only one potentiometer for shift intention recognition and two non-contact sensors for gear position recognition are required. Other signals like engine speed are usually available in the vehicle. The target of component reduction has been realised by means of intelligent control software.

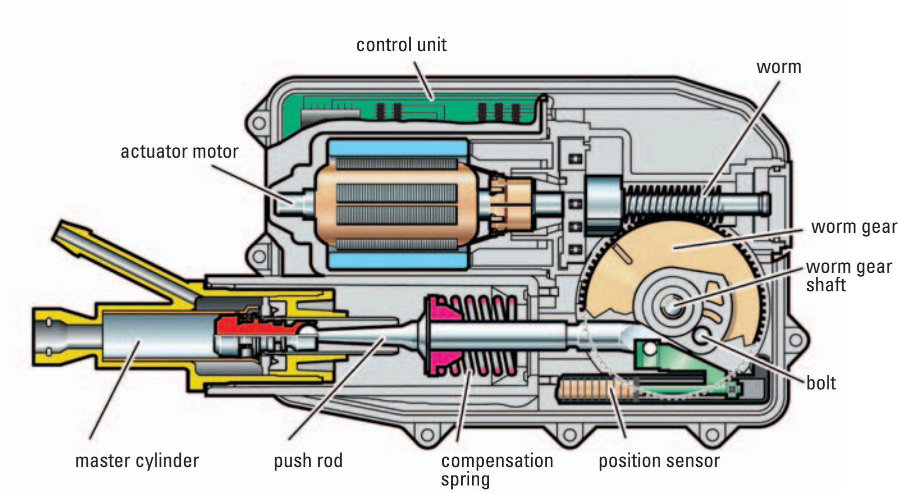


Fig. 3

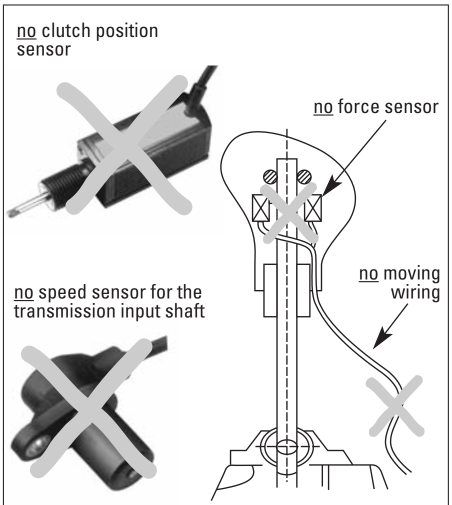


Fig. 4

The LuK ECM: optimisation of the system due to component reduction

Torque tracking strategy

The basic solution to support fast shifting with the small electric motor and to improve tip-in/back-out performance is the previously mentioned torque tracking strategy – see also the illustration in **figure 5**.

Usually a clutch is designed to transmit about 1.5 to 2.5 times the maximum engine torque. Torque tracking is based on the principle that the set clutch torque will be the current engine torque plus a certain safety margin. When the driver releases the accelerator pedal before shifting, the clutch torque will be reduced simultaneously. When the shift intention is detected, the clutch is almost completely open. The remaining time to open the clutch fully is very short and therefore allows fast gear changes.

One further advantage of torque tracking is the improved tip-in/back-out performance. A full throttle acceleration generates torque peaks which cause jerking oscillations in the drive train. In this case due to torque tracking a very short slip in the clutch damps the oscillation. This results in an improved comfort and protects the drive train from torque impacts. The minimal slip is not relevant in terms of fuel consumption and wear and tear of the clutch.

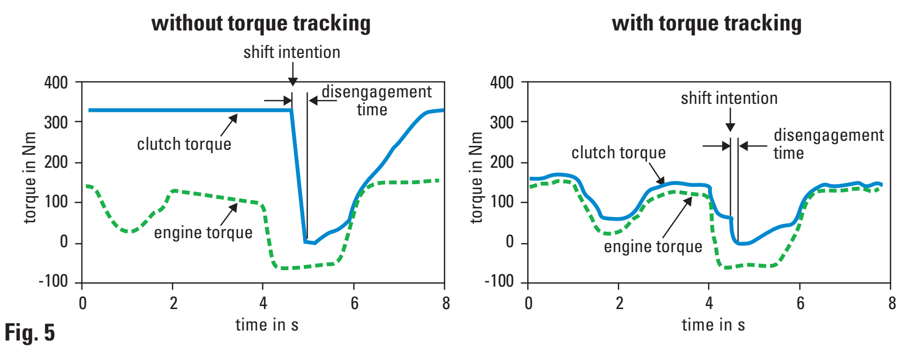


Fig. 5

Benefits of the LuK ECM:

Increased road traffic safety
The ECM system relieves drivers of having to concentrate on operating the vehicle and thus allows them to turn their attention to the actual traffic situation. With the ECM it is impossible to stall the engine.

Improved manoeuvrability
LuK has developed a strategy whereby the vehicle can creep forward when in 1st, 2nd or reverse gear even if the driver is not pushing on the gas pedal. This “creep function”, much appreciated on automatic transmissions, makes it easier for the driver to inch forward when manoeuvring. Depressing the brake pedal automatically reduces the creep torque to zero – thus optimizing fuel efficiency.

Increased comfort in stop-and-go traffic
Driving comfort in stop-and-go traffic is considerably improved. Here, too, the "creep function" has proved very valuable.
Improved tip-in/back-out performance
As previously explained, torque tracking allows the driver to change gears quickly. In addition, tip-in/back-out responses are significantly improved.

Protection of engine and transmission from operating errors
If the engine speed exceeds the admissible limits when gearing down, the clutch will be engaged only to a degree protecting the engine from being damaged. Damage to the transmission is avoided by closing the clutch only when the next gear is fully engaged.

Tow start/tow off
In an emergency situation, the vehicle can be towed off analogous to a conventional clutch. Tow-starting the engine is also possible.

Optional features
Optional features of the ECM system help to further reduce fuel consumption:

Start/stop function
Using the ECM system, it is possible to switch off the engine where advisable without compromising on driving comfort. To restart the engine the driver must do nothing more than e.g. put in the first gear.

Gear display
A display can be integrated in the dashboard showing the driver which gear is actually selected without requiring a look on the gearshift lever.

Gearshift recommendation
The gearshift control in the dashboard informs the driver of the gear recommended to achieve optimum fuel efficiency. A recommendation drivers won’t hesitate to follow as changing gears is made easy with the ECM system.

Hydraulic clutch release systems

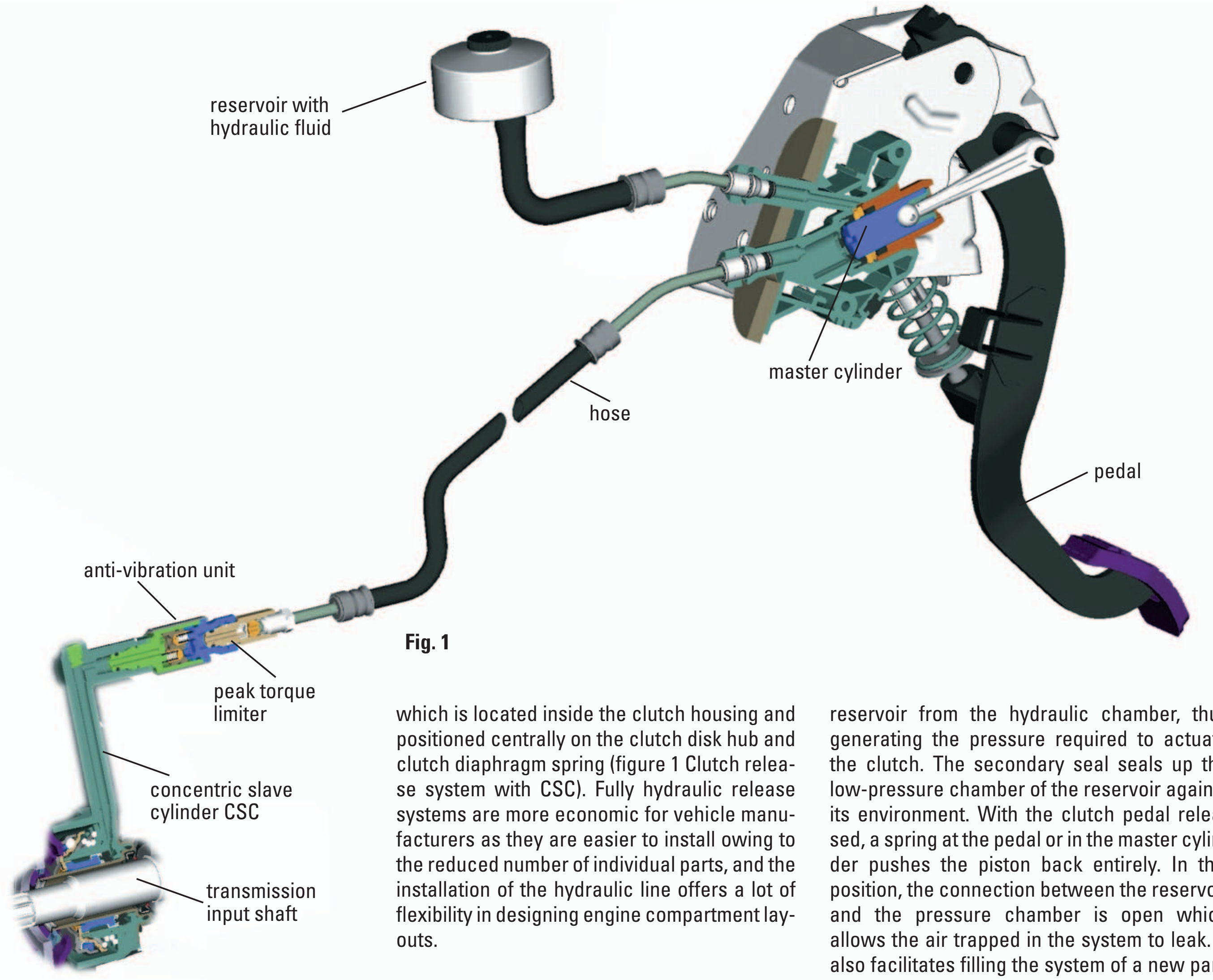


Fig. 1

Function

On vehicles with foot actuated single disk 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 disk hub and clutch diaphragm spring (figure 1 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 2) 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 connected 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.

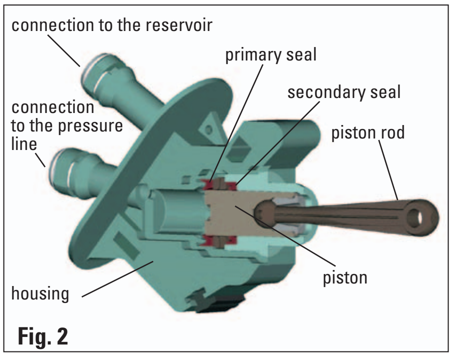


Fig. 2

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 3) with two non-return valves acting in opposite direction.

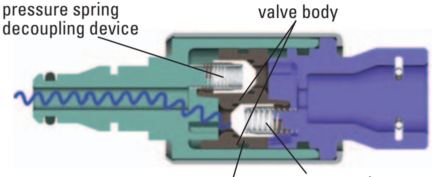
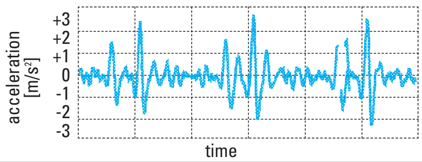


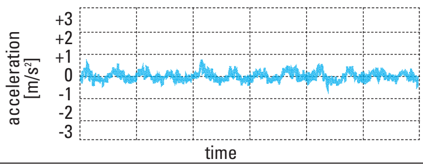
Fig. 3

Vibration at the clutch pedal

without anti-vibration unit



with anti-vibration unit



Peak torque limiter

Peak torque limiters 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.

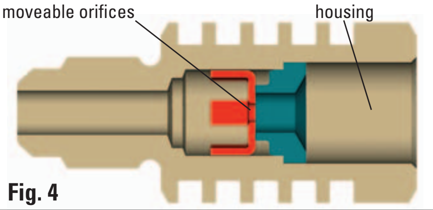
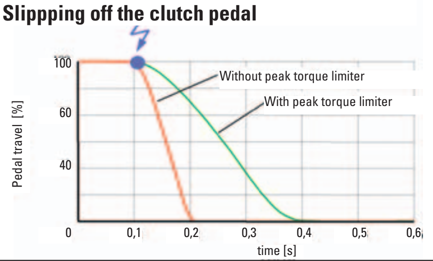


Fig. 4

Slipping off the clutch pedal



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 pre-charge spring and a bleed screw. The

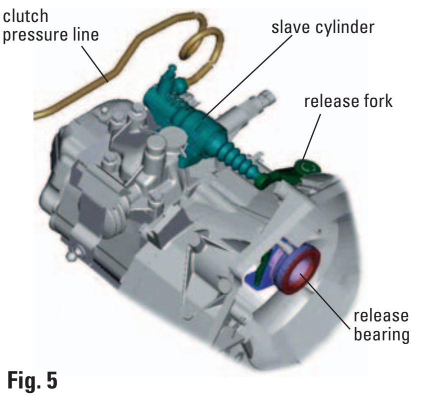


Fig. 5

pre-charge spring applies constant pre-load 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 6), the release bearing is directly connected to the piston and tensioned against the diaphragm spring tips of the clutch by the integrated pre-charge 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.

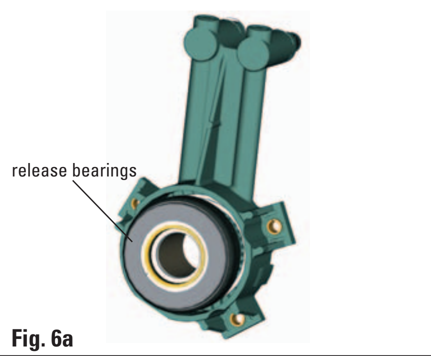


Fig. 6a

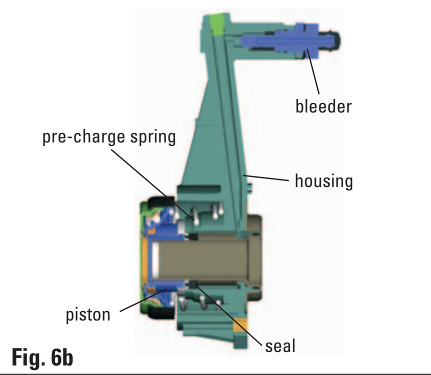


Fig. 6b

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 at least every two to three years. 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 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.

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 the diaphragm 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**

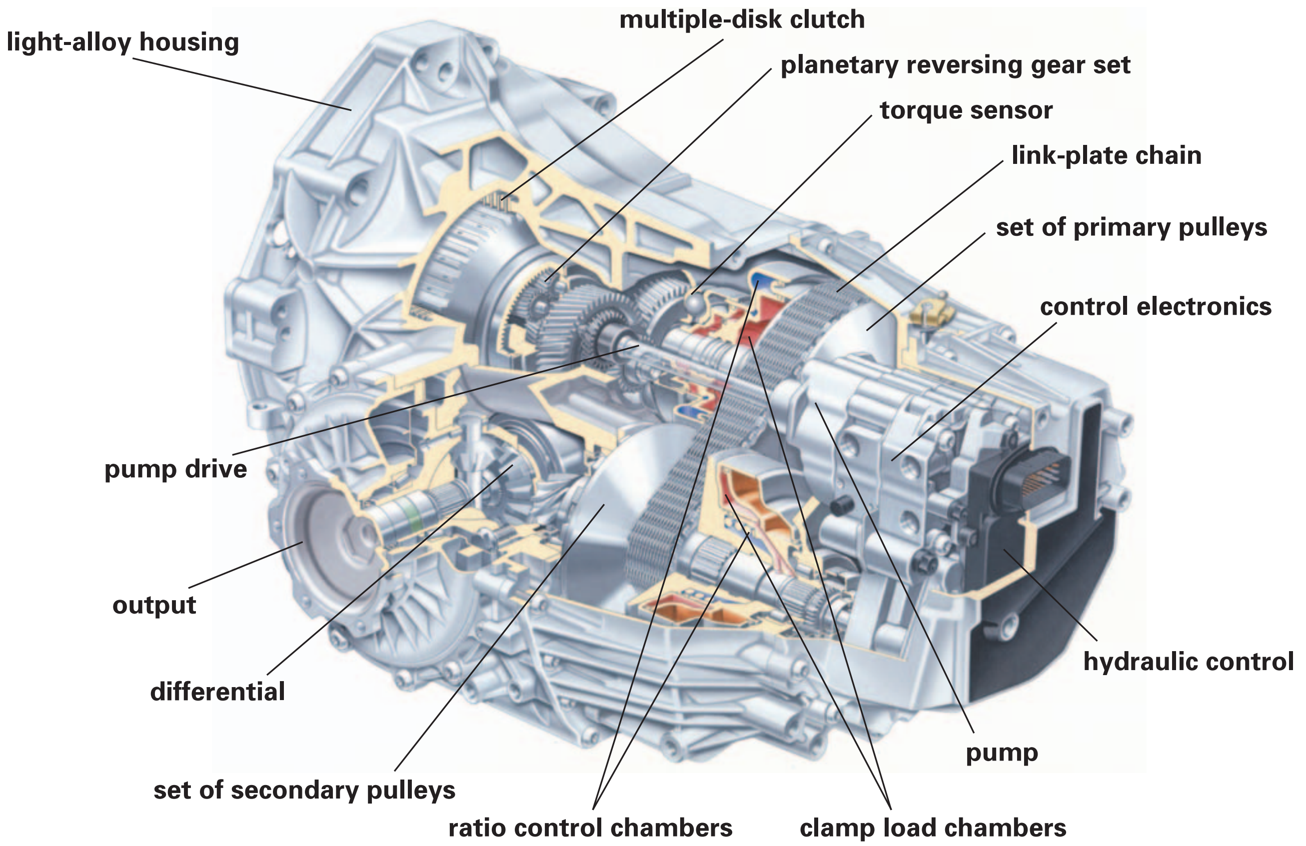


Fig. 2 Audi multitronic with LuK CVT components

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 300Nm 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 wheel side without slip of the linking element. The clamping load and ratio control of the pulley sets are regulated hydraulically.

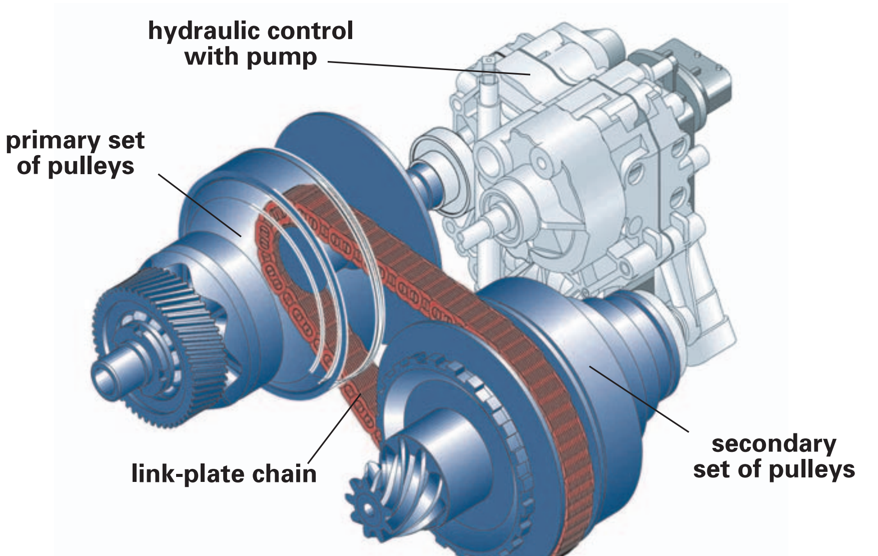


Fig. 1 LuK CVT components

Design of a CVT

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

Figure 2 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 load and ratio control, these functions are also ensured by hydraulically actuating the relevant clutches. The hydraulic system, in turn, is regulated by the control electronics.

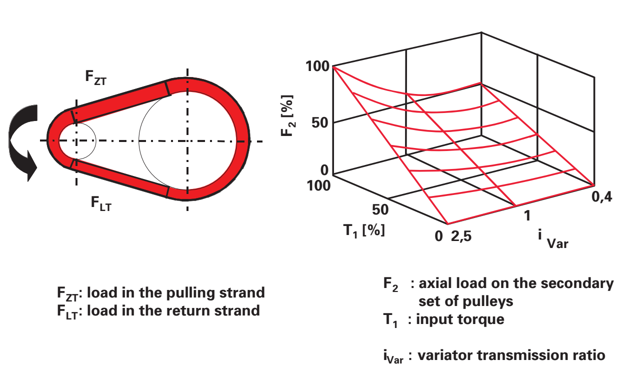


Fig. 3 Continuously variable power transmission by means of frictional engagement

The multitronic® uses a wet multi-disk 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. The figure on the right-hand side 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 3 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 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, beyond 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.

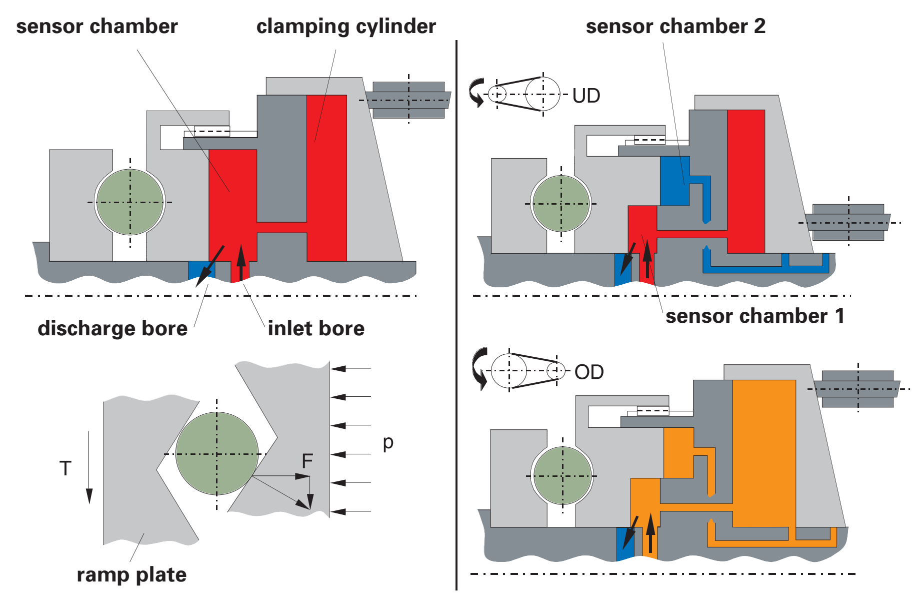


Fig. 4 Single and dual-stage torque sensors

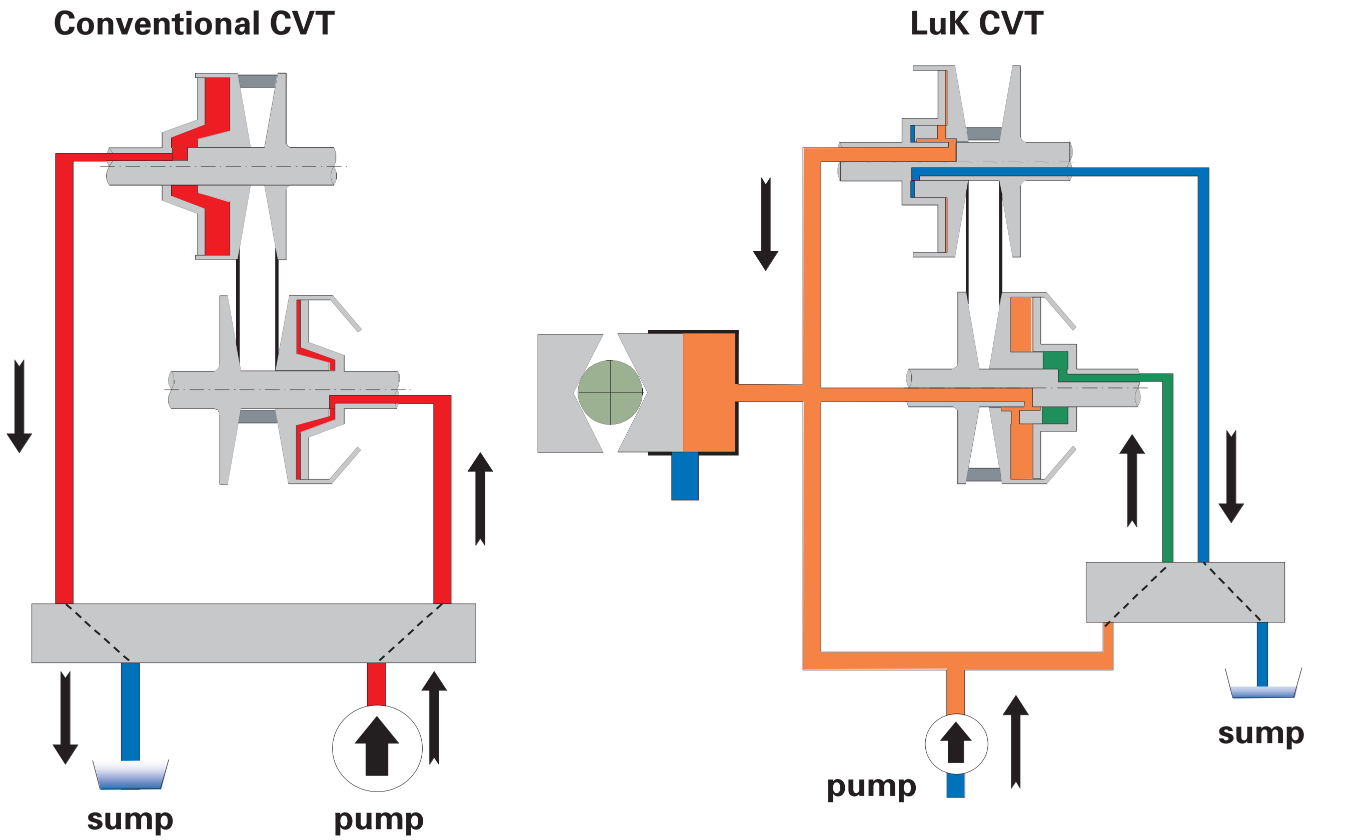


Fig. 5 Comparison between conventional vs. LuK hydraulics

This system is currently undergoing further development. Alternatively, the said dual-stage torque sensor is also available as continuously variable system by adjusting 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 5, 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 control functions 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 necessary to fulfil the system's dynamic requirements, 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 pulley 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 6 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.

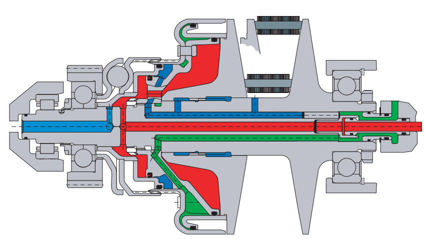


Fig. 6 Primary set of pulleys with integrated torque sensor

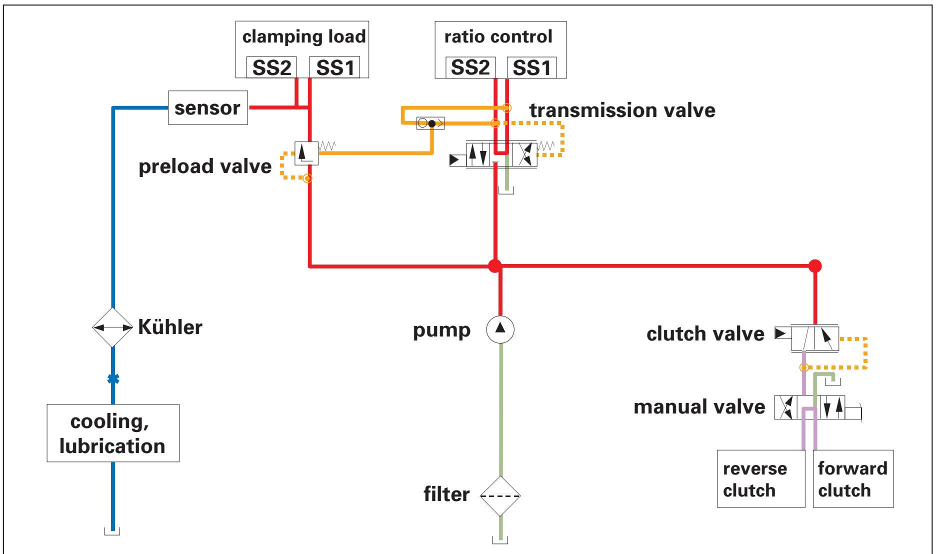


Bild 7 Vereinfachtes Hydraulikschema

Simplified hydraulics schematic – CVT with lock-up clutch

Figure 7 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 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 lightweight. 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 8 illustrates the CVT chain for applications producing torque up to 300Nm. 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.

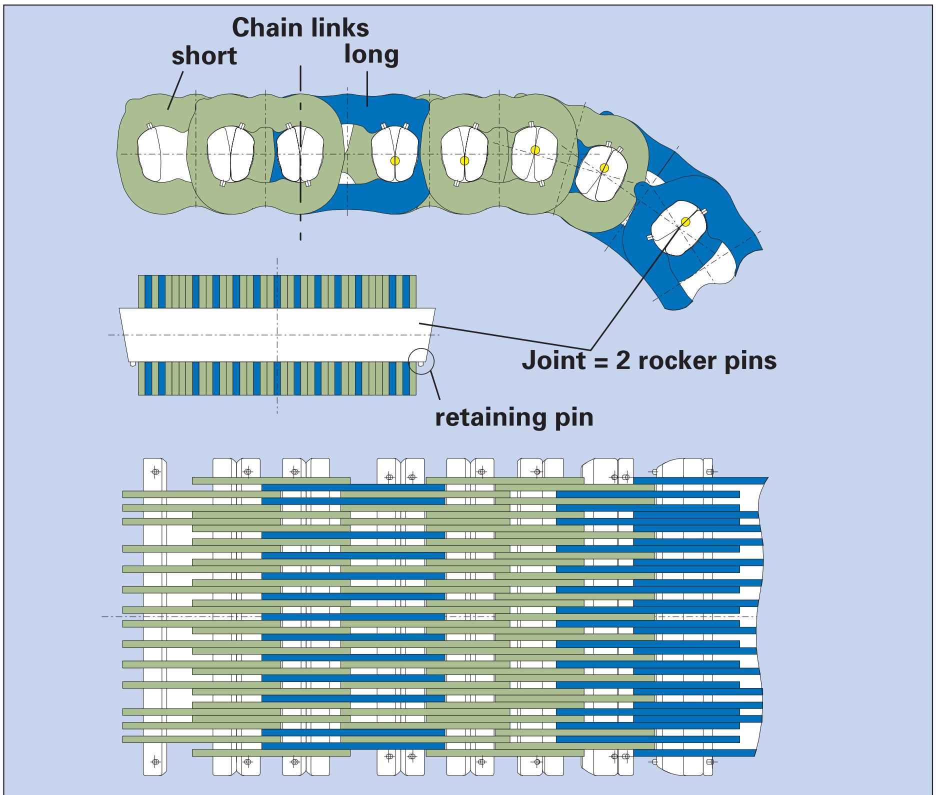
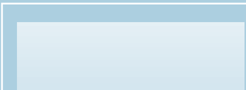
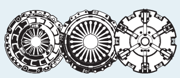
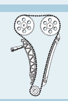
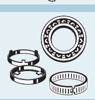
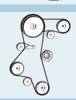
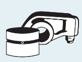
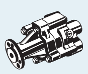
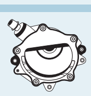
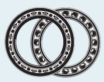


Fig. 8 Design and components of the LuK chain

					
	✓	✓	✓	✓	✓
	✓	✓			
	✓	✓	✓	✓	
	✓	✓	✓	✓	
	✓	✓			
	✓	✓	✓	✓	
	✓	✓			
	✓	✓	✓	✓	

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