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Overload clutches and designs in agritechnical engineering – from the beginnings to the present day

Agricultural implements are often used under unpredictable, extreme conditions. Moreover, they are frequently driven by tractors with a higher power than necessary for driving the implement. Appropriate precautions have to be taken to protect drivelines against overloading caused by initial starting torques, excessive charging and blockages. Overload clutches, designed to suit the respective application, have been in use ever since the power take-off was introduced.

Keywords
Overload clutches, Function systems

Abstract
Landtechnik 67 (2012), no. 5, pp. 374–384, 36 figures, 2 references

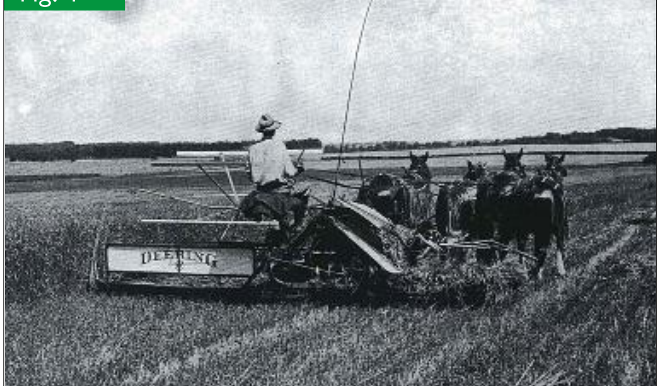
Originally, horse-drawn agricultural implements were driven via the travel wheel (**Figure 1**). The power of the horses and the traction of the travel wheel provided natural protection against overloading of the driveline. Sensitive implement components were at best protected by simple predetermined breaking points. In addition, these implements often had a pawl overrunning clutch, which ensured that the drive power could only be transmitted in one direction when reversing or turning the implement.

IHC (International Harvester Company) applied for a patent on the power take-off (PTO) shaft in 1907, installing it on the first tractor in 1917. This was an important invention that still has an impact today, as it turned the tractor from a pure „pulling machine" into a central source of propulsion (**Figure 2**).

In 1917, McCormick USA was the manufacturer to have the first „tractor binder" driven via the PTO shaft of a tractor. As a result, the above-mentioned natural overload protection of the travel wheel was no longer present. Rather, the engine's flywheel provided kinetic energy that transmitted an uncontrolled torque via the drive shaft into the implement in the event of sudden overloading/blocking. Consequently, an overload clutch was already to be found in the telescopic section of the drive shaft in 1917. It was a spring-loaded axial ratchet clutch fitted with a coil spring. Since the telescopic elements between the two universal joints rotated at unequal speeds, depending on the joint deflection angles, the joint planes between the two joints could be offset („joint plane offset") if the axial ratchet clutch had a symmetrical pitch. Even with balanced joint angles, this led to irregular rotary motion, which in turn generated rotary oscillation that caused vibration and machine damage. This relationship was very soon recognised, as a result of which the axial ratchet clutch was designed in such a way that it could only re-engage at a rotational position of 180°. Moreover, this design possessed substantial masses that had to be constantly decelerated and accelerated in the irregularly rotating telescopic section of the drive shaft. This effect likewise caused vibration that had a negative impact on the smooth running of the driveline. Furthermore, the user could vary the transmittable torque by adjusting the coil spring. This limited or even eliminated the protective effect. Despite the disadvantages described, this was the most commonly used type of overload clutch until well into the 1950s.

The power take-off spread very rapidly from about 1922, also being offered by other tractor manufacturers. Above all, this type of drive permitted the development of previously unknown machines. One example was the pick-up baler for hay and straw.

Fig. 1



Advertising IHC-Deering 1917

Acting cyclically via a crank, the ram necessitated an additional flywheel to store energy. An overload protection device that maintained the torque in the slipping phase was required in order to limit the acceleration torque of this mass. The belt drive was used for this purpose for a long time. It offered not only overload protection, but also the necessary transmission for speed adaptation. Apart from providing overload protection, this type of drive also reduced the vibration caused by the kinematics of the machine and irregularities in the driveline (**Figure 3**).

It thus became apparent that there could be a need not only for torque protection in the main driveline, but also for an additional clutch in a power take-off. Consequently, no single overload principle in the main driveline could cover the entire range of different requirements, meaning that further effective principles were needed.

Until well into the 1950s, PTO drive shafts and overload clutches were largely produced by the machine manufacturers themselves. Consequently, only secondary importance was often attached to them when designing machines, the result being that the above-mentioned, original designs of overload protection devices also remained largely unchanged. Only when supplier companies began to specialise in the early 1950s, offering driveline components specifically for agritechnical engineering, did PTO drive shafts and overload clutches acquire their eminent importance. In-house production by the machine manufacturers was abandoned within a very short space of time and taken over by the specialists (**Figure 4**). In this context, particular mention should be made of Walterscheid, Siegburg/Lohmar, later GKN Walterscheid, a company that devoted itself exclusively to the development of „agricultural PTO drive shafts and overload clutches“ from 1953 onwards. A wide range of overload clutches for the most diverse applications and output ranges was developed in just a few years. Today, Walterscheid still has a worldwide reputation as a specialist for agritechnical driveline engineering. Consequently, the development steps and their prerequisites will be presented in detail. Clutches from other manufacturers are also largely based on the same effective principles.

In addition to the above-mentioned principal deficiency of the axial ratchet clutches used in the telescopic elements, it was the large rotating mass that led to rotary oscillation (**Figure 5**). Moreover, the ratchet caused axial runout during the slipping phase, this having a damaging effect on the bearings of the tractor‘s power take-off and on the input shafts of the implements, as well as on the locking elements of the PTO drive shaft. Since the user could influence the transmittable torque, the necessary overload protection was not guaranteed. Substantial implement damage was repeatedly the consequence.

The Walterscheid company from Siegburg, which had previously been largely unknown in agritechnical engineering, presented a complete range of PTO drive shafts with integrated overload clutches and safety guards, “System Schröter”, for the first time at the DLG exhibition in Cologne in 1953 (**Figure 6**). Kurt Schröter and his revolutionary ideas were in no way unknown in

Fig. 2


International Tractor Binder

Illust. 306. 10-ft. Tractor Binder and International Tractor. The bundle carrier, power take-off on the tractor are not included with the binder in weights or prices

Latest Development

The International tractor binder is the most modern harvesting equipment made—a thoroughly up-to-date binder, designed especially for use with International gear drive tractors. It cannot be operated by horses or used with any other make of tractor.

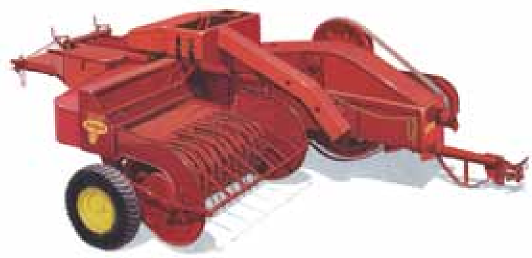
The binder mechanism is driven by power transmitted from the tractor engine through a revolving shaft. This construction insures a steady supply of power to meet the varying field and grain conditions. No choking of the binder due to slippage of the main wheel in wet and sandy soils, because the main wheel, being relieved of the work of driving the binder mechanism, simply carries the weight of the binder.

In addition to the direct power drive, the International tractor binder has a low grain elevator, 10-ft. width of cut, roller bearings throughout, heavier drive chains, stronger main frame, etc., than horse drawn binders. The control levers are in front within easy reach of the tractor seat.

With this outfit one man can harvest as much grain in a day as two men with two 7-ft. horse-drawn binders. This extraordinary capacity is due to the uniformly fast travel of the tractor and, further, to the fact that the binder is run by power from the tractor. The tractor binder mechanism, running at the same regular speed, regardless of forward speed of the tractor, minimizes clogging.

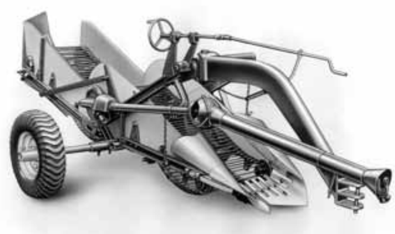
Excerpt from the 1917 advertising of the company International Harvester

Fig. 3



Baler Welger AP12

Fig 4



Potatoe Harvester Heinrich Lanz 1954

Fig. 5

Original IHC PTO drive shaft with axial ratchet clutch, ca. 1950

Fig. 6

PTO drive shaft with integrated overload clutch, brochure 1953

Fig. 7

Cam-type ratchet 400 Nm, year of construction 1953/54

Fig. 8

Mobile field measuring service, 1958

agritechnical engineering and vehicle construction at this time (overrun brake, Thümag trailers, Alpenland tractors [1;2]) .

The development of the overload clutches followed the development of the implements and will therefore also be presented in chronological order.

Clutch designs after 1953

The key feature of this overload protection was the fixed level of the transmittable torque, which could not be influenced by the user. At the same time, this clutch was part of the telescopic unit. The transmittable torque was determined by the number of spring-loaded cams that engaged the specially designed profile of the "lemon-shape profile tube" (**Figure 7**). However, this meant that the torque required for the implement also had to be known sufficiently accurately. The first "Technical Handbook" with detailed product presentations was already published in 1956, additionally discussing the theory of the kinematics of universal-joint shafts, as well as the mode of operation of overload clutches. An overview of the output powers of 26 machine groups served as a design aid, together with the first "Tractor connection table" listing the 242 tractor types produced by 30 German manufacturers in 1955. This was the first table to present the important geometrical conditions of importance for coupling implements.

This Handbook already covered 106 PTO drive shaft and clutch versions, reflecting the requirements of the individual machine types and their powers.

To be able to qualify the different requirements, a special measuring service was set up, using electronic measuring devices (strain gauges) to determine the machine characteristics and the necessary power in practical operation (**Figure 8**). Since the size of the measuring hubs familiar from general machine construction significantly affected the kinematic conditions of the universal-joint drive, a special measuring hub of short design and equipped for the different tractor and implement connections was developed specifically for agritechnical applications. This special measuring hub has since been widely used in agritechnical engineering, both by customers and by institutes. It is used in large numbers.

Since the tractor connections varied greatly, the lengths of the PTO drive shafts had to be equally variable. Telescopic tubes made it very easy to adjust the length. However, the cams could disengage if the telescoping travel was long and during length adjustment. The number of cams was determined by the length of the telescoping travel, thus limiting the torque level. It was for this reason that the combination ratchet (**Figure 9**) was developed on the basis of the cam-type ratchet, separating the

Fig. 9

Combination ratchet 440 Nm, year of construction 1954 overload clutch and the telescopic unit. To this end, the cam-type ratchet was installed in rotating fashion as a unit in an axially fixed profile tube. The outer contour of the clutch served as the inner telescopic element in this context.

The maximum torques of the combination ratchet were insufficient for high torque requirements, e.g. for soil-working implements. Therefore, an axially acting ball-type ratchet was already developed as early as 1954 (**Figure 10**). Balls, guided in the hub and loaded by Belleville springs, pressed on a plate provided with dome-shaped recesses. The balls were forced out of the recesses in the plate when the set limit load was reached. This led to very high axial forces (axial runout), and also to a strongly pulsating torque. The mode of operation of the ball-type ratchet was thus similar to that of the axial ratchet clutch. Nevertheless, this function was preferred for clearing obstacles/blockages in certain applications.

The first two clutch types discussed – cam-type ratchet and combination ratchet in the telescope – had the disadvantage of a 180° pitch. Since the maximum torque was dependent on the available length of the telescopic elements, this insight led to the development of the star ratchet clutch or radial pin clutch (**Figure 11**). Like the ball-type ratchet previously, it was located outside the telescopic elements, directly on the tractor's power take-off shaft or on the implement's PIC. It had eight grooves and thus engaged every 45°. As a result of this, and of different lengths, a higher number of cams can be installed. This permits fine setting of a higher torque, depending on the cams and springs fitted.

During the slipping phase, the previously described cam-type ratchets and star ratchets generated oscillations that could build up in very elastic drivelines. The cam-type friction clutch was developed to alleviate these rotary oscillations (**Figure 12**).

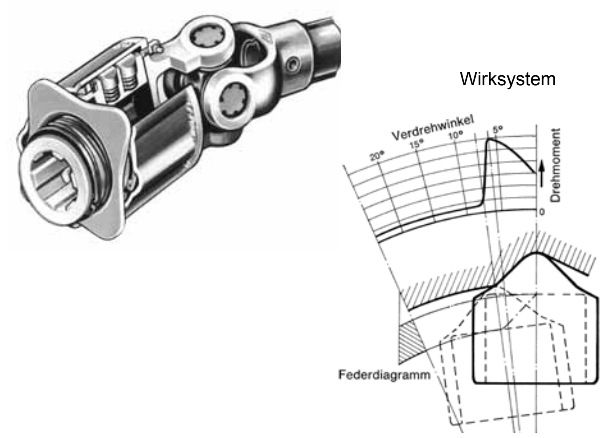
On this clutch, the locking cams acted inwards, engaging the grooves of a profiled hub. On the outside, the springs pressed against brake shoes with friction linings, these in turn acting on a smooth surface of the housing. The torque was transmitted by mixed friction, with positive locking of the locking cams in the hub and purely frictional connection via the friction linings in the housing. Owing to the different effective diameters, slip occurred at both points. The preferred application was cable winches, to prevent swinging of the cable in the slipping phase.

In agritechnical engineering, there are numerous machines with large rotating masses, e.g. high-density pick-up balers and rotary mowers. When the power take-off is deactivated, these masses continue to run, meaning that the drive shaft also rotates for an uncontrolled period and can block at large angles. This increases the accident risk. Consequently, these machines are fitted with overrunning clutches that transmit the torque in one direction only. So-called friction-based overrunning clutches, e.g. from Kessler, are familiar from general machine construction (**Figure 13**). They are based exclusively on a frictional connection, therefore requiring a large diameter and being sensitive to shock-type torque loads. Owing to the large size

Fig. 10

Ball-type ratchet 1500 Nm, year of construction 1954

Fig. 11

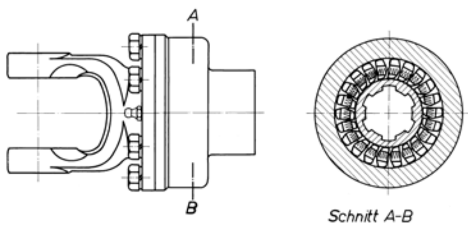


Star ratchet 1200 Nm, year of construction 1957

Fig. 12

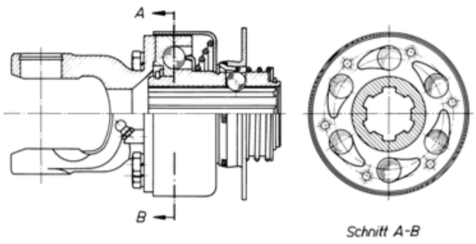
Cam-type friction clutch 1600 Nm, year of construction 1958

Fig. 13



Kessler friction-based overrunning clutch 400 Nm, year of construction 1956

Fig. 14



Ball-type overrunning clutch 1200 Nm, Year of construction 1958

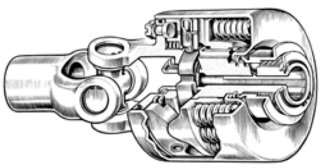
Fig. 15

Coil-spring friction clutch 550 Nm, year of construction 1958

Fig. 16

Welger flywheel clutch, year of construction 1968

Fig. 17



Automatic overload clutch 1200 Nm, year of construction 1958

Fig. 18

Star clutch 1000 Nm, year of construction 1962

inherent in the system design, they can often also not be accommodated in the limited space on the tractor or the implement.

Given this realisation, and also for reasons of cost, a ratchet-type freewheel with engaging grooves was developed that was not only insensitive to shocks, but also possessed a higher torque transmission capacity (**Figure 14**). Spring-loaded balls pressed on inclined ramps. In the event of reverse torque, the balls were displaced axially and could not transmit any torque owing to the ramp design. This overrunning clutch was characterised by a very compact design.

The use of high-density pick-up balers for collecting straw and fodder increased rapidly in the 1950s. Their design changed as a result of elimination of the belt drive as the previous form of overload protection. Thus, the flywheel mass was increasingly relocated to a position directly at the implement input. The necessary force diversion and transmission was handled by gearboxes. The overload clutch became part of the flywheel and was connected directly to the drive shaft. Since the known ratchet clutches interrupt the driveline in the slipping phase, they were not suitable for accelerating large masses. A constant-torque clutch was necessary for this purpose. Friction clutches worked in this way (**Figure 15**). The torque was determined by the number of friction linings and coil springs and their pre-tension. It was accepted in this context that the user could alter the pre-tension, and thus the torque, up to the point of blocking. Where necessary, the above-mentioned ratchet clutches were used within the baler, e. g. on the pick-up.

The Welger company from Wolfenbüttel developed a friction clutch of its own that was designed exclusively to protect the flywheel of balers. Mounted on pivot bearings, a three-arm, central leaf spring pressed on the points of contact via brake linings in a large diameter range of the flywheel. The drive shaft connection was located at the centre of the leaf spring system, meaning that the power flowed via the arms and the brake linings (**Figure 16**). This clutch had a fixed setting and could not easily be influenced by the user. The disadvantage, however, was that this clutch design was application-specific and thus did not permit universal use.

There was already a demand for cut-out clutches in the 1950s. Up to that time, the effect was achieved only by means of predetermined breaking points (shear bolt clutches). However, replacement of the shear bolt caused a lengthy stoppage. In addition to which, the clutches were not always readily accessible. These disadvantages considerably reduced their acceptance. The so-called automatic overload clutch was introduced in regular production to enable automatic re-engaging following overloading (**Figure 17**). The main applications were rotary cultivators and stump cutters in forestry. The driveline was interrupted in the event of overloading, only a minimal residual torque being preserved. Spring-loaded cam plates transmitted the torque. When overloaded, the cams disengaged against the spring tension and moved into an idle position, in which no torque was transmitted. Inertia forces prevented their return. The inertia forces were no longer present when the power take-off came to a halt, and the spring force returned the cam plates to their starting position. The clutch re-engaged. As a result, work was interrupted only briefly. However, this design was too expensive in those days, meaning that it was used exclusively for the special applications mentioned above. The time was not yet right for such a convenient solution.

The 1960s

Ratchet clutches rapidly gained great market acceptance. However, the vibration occurring during the slipping phase was a problem in some applications. Moreover, preservation of the torque was demanded, e.g. in the case of tractor front-wheel drives where an interaxle differential was dispensed with. In this instance, the clutch was merely intended to prevent torsion between the front and rear axles when cornering, in which context the torque was to be largely preserved. The star clutch met this demand (**Figure 18**). In this design, the housing of a ratchet clutch had more grooves than cams. The cams were arranged in such a way that only a predetermined number of them engaged. This resulted in gentle transitions and smoothing of the torque profile in the slipping phase. The importance of the clutch waned following the introduction of interaxle differentials in four-wheel drives.

The ball-type overrunning clutch (**Figure 14**) consisted of components that could not be used for other purposes and also proved to be sensitive to shocks. Consequently, a new, efficient overrunning clutch of modular design was developed using the star ratchet clutch housing and the PTO push-pull lock (**Figure 19**). Spring-loaded pins, guided in the hub, engaged a drive plate with inclined pockets inserted in the base. Their large-area contact ensured reliable torque transmission.

Friction clutches on the flywheel of a baler necessitated an additional overrunning clutch. This meant a friction clutch on the implement side and an overrunning clutch on the tractor side. As a result, the drive shaft continued to rotate until the flywheel came to a halt. An overrunning clutch was integrated in the friction clutch in order to eliminate this hazardous situation. With its externally profiled housing, the axial pin overrunning clutch offered ideal prerequisites for creating a combined friction and overrunning clutch (**Figure 20**). This development was widely accepted on the market. The reduced accident risk led to this combination being included in the regulations of the employer's liability insurance associations. Packer, knotter and pick-up were provided with separate internal protection.

Friction clutches gained access to further applications as a result of maintaining the torque. However, actual overload protection was not sufficiently guaranteed owing to the possibility of subsequently adjusting the coil springs. Therefore, the market called for a friction clutch with a fixed torque setting, adapted to the function and durability of the implement. A friction clutch with a specially designed Belleville spring having an almost horizontal spring characteristic was developed (**Figure 21**).

The spring force dropped again slightly after reaching its maximum. New units were installed just after the maximum,

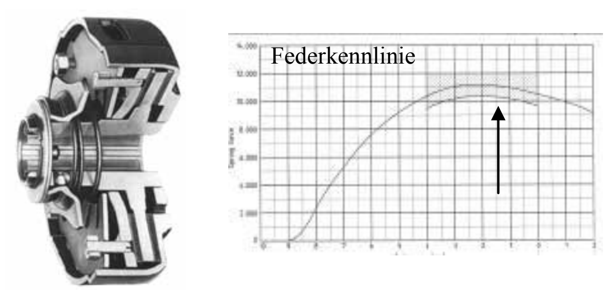
Fig. 19

Axial pin overrunning clutch 1550 Nm, year of construction 1964

Fig. 20

Combined friction and overrunning clutch 1500 Nm, year of construction 1964

Fig. 21



Belleville spring friction clutch 1350 Nm, year of construction 1964

meaning that the contact pressure increased when wear occurred on the clutch linings. Wear was generally also accompanied by a decline in the coefficient of friction. This effect, the increase in force, made it possible to maintain a virtually constant torque over the entire service life.

The torque was set by means of different Belleville springs, their stacking and by the number of clutch linings. Since friction linings tend to seize up/rust in after extended periods of time, a releasing device was provided. After lengthy stoppages, the clutch linings were relieved of the spring pressure by tightening four hexagon nuts, allowing the clutch to be turned freely. This made it possible to release seized linings again. A further advantage was that the linings were easy to replace without

Fig. 22

Friction clutch for chain drives 650 Nm, year of construction 1965

using special tools – the spring assembly could be removed in compressed state after tightening the hexagon nuts. The clutch was therefore considered to be very maintenance-friendly. This type of clutch was particularly characterised by its compact design, with hardly any projecting parts.

Self-loading bale trailers necessitated individual protection of the chain wheels. This was achieved by a friction clutch, where the chain wheel simultaneously formed the friction surfaces of the clutch (**Figure 22**). The radial forces of the chain were absorbed by a friction bearing ring. This type of bearing offered the advantage of dry running, since sintered bronze bearings could not operate without lubrication in the long term and were also unable to withstand the high surface pressures.

The torque was determined by the choice and stacking of the Belleville springs. In order to achieve a largely consistent torque via the wear on the linings, this Belleville spring likewise had a virtually horizontal characteristic. This type of clutch was used not only on self-loading bale trailers, but also on many chain drives, e. g. on pick-up drives.

The 1970s

Tractor outputs increased very rapidly in the 1960s and 1970s. Rising tractor outputs also led to a growing number of implements that used not only the 540 rpm power take-off shaft, but also the 1,000 rpm version. Driveline design was increasingly determined by torque measurements (**Figure 23**). The results of the measurements were also available to the customers. In this way, not only was the power determined, but the machine characteristic recorded at the same time, offering the design engineer important assistance in optimising the machines. Progress in measuring technology and evaluation methods was consistently exploited. More than 100 field measurements per year were performed during the above-mentioned period. They offered a good foundation for adapting product development to the wide range of different market requirements. The measurement data also served as a basis for extensive test bench studies, allowing continuous development regardless of the season.

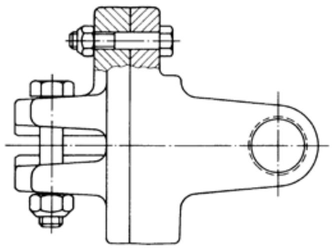
Increasing tractor outputs created the problem of overheating of overload clutches, since the power is converted into heat when the clutch is disengaged. This called for cut-out clutches that completely interrupted the torque flow.

In the simplest form, these requirements are met by a shear bolt clutch acting as a predetermined breaking point. A bolt or a shearing pin is sheared off when overloading occurs. The level of the shearing torque was determined by the shearing radius, the diameter and the strength of the shear element (**Figure 24**). Whenever the clutch tripped, the shear element had to be replaced, causing a lengthy interruption of the workflow. To keep this situation within acceptable limits, the shearing torque had to be very high. Consequently, protection was ultimately only guaranteed in the event of blocking. These clutches were often produced by machine manufacturers themselves, mainly in power take-offs.

Fig. 23

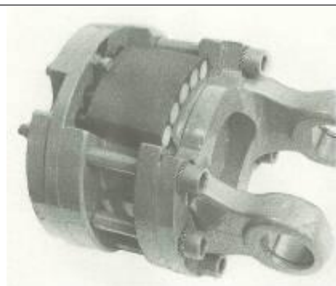
Field measurements in the 1960s/1970s

Fig. 24



Shear bolt clutch, operating principle

Abb. 25



Lely automatic shear bolt clutch, 1970s

The van der Lely company developed an automatic shear bolt clutch for rotary harrows, which tended to block on stony ground (**Figure 25**). In this design, the shear bolt was used repeatedly and advanced by spring pressure, thereby automatically replacing the shearing element. The shearing element was consumed down to a function-related stub. A set of 10 shear bolts permitted 50 overloads, and 10 spare bolts were carried in a magazine. This interesting automatic shear bolt clutch was used not only on rotary harrows, but also in various other applications.

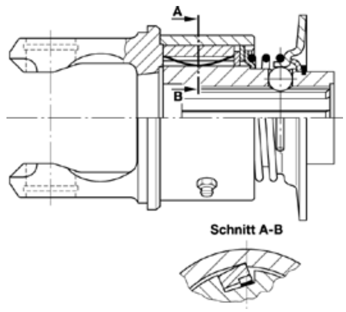
The 1980s

The increasing use of higher speeds, brought about by the 1,000 rpm power take-off shaft, led to the axial pin overrunning clutch proving to be too sluggish as a result of its mass and the friction of the pins. The pins were thus no longer able to engage reliably. There followed the development of the key-type overrunning clutch (**Figure 26**). With its low-mass, radially acting and tiltable locking keys, this clutch responded rapidly and was suitable for high speeds of up to 1,000 rpm. Centrifugal forces assisted the engaging of the locking keys, meaning that only little spring assistance was necessary. This overrunning clutch was designed to withstand high shock loads. As a rule, there were four driving grooves in the housing that were engaged by two opposite locking keys. In special versions, the housing could also be produced with just two driving grooves, thus ensuring a 180° pitch in applications in the telescopic elements between two joints (preservation of the joint planes).
Hydraulically shifted multiple-disk PTO shaft clutches of tractors, running in the gear oil, engaged very harshly, virtually independently of the torque load. This resulted in shock-like acceleration, particularly on implements with large rotating masses, but low output (shifting with shock loads). To limit these high starting torques, a simple, small friction clutch was developed as a special solution, predominantly for use on disk broadcasters (**Figure 27**). With its simple design and low heat absorption capacity, this clutch served exclusively to limit shock loads when starting.

The call for clutches with a cut-out feature and automatic restoration of the overload function thus became ever louder. The ball-type cut-out clutch (**Figure 28**) was developed on the basis of the experience acquired with the automatic overload (**Figure 17**). On this clutch, spring-loaded balls axially engage special dome-shaped recesses. If the clutch torque is exceeded, the balls are forced out of these recesses, against the spring pressure, and into specially designed recesses in a shift plate performing limited rotation. Power transmission in the driveline is thus interrupted. Because of the inertia of its mass, the shift plate cannot return to its starting position until a low speed is reached (< 200 rpm). The clutch is then re-engaged. The clutch acts in both directions.This clutch type is frequently used in forage harvesters and snowploughs.

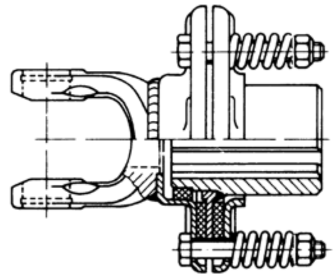
Use of the different power take-off shaft speeds also had an influence on clutch design. Subsequent adaptation was re-

Fig. 26



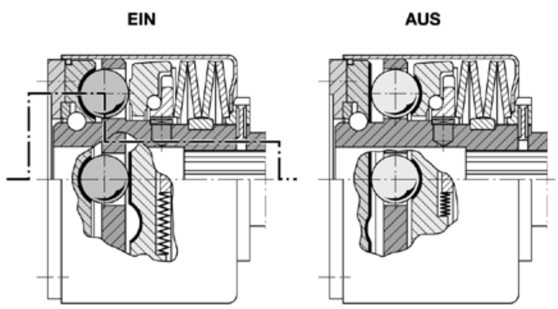
Key-type overrunning clutch 3000 Nm/1000 rpm, year of construction 1982

Fig. 27



Friction clutch 400 Nm, year of construction 1982

Fig. 28



Ball-type cut-out clutch 2500 Nm, System diagram, year of construction 1982

Fig. 29

Friction clutch and combined friction and overrunning clutch 2700 Nm, year of construction 1984 quired, predominantly in the case of friction clutches. With this clutch (**Figure 29**), this modification can be performed by dealers/users in three steps without special tools. Blocking is impossible.

The machine-specific, basic factory setting is made in fine graduations by way of the arrangement and number of the coil springs, as well as the number of friction linings and the arrangement of the locking ring (**Figure 30**). The linings are likewise changed without special tools and without dismantling the spring assembly. The selected torque setting is thus preserved. The friction linings are released/relieved by tightening the nuts on the top side of the clutch. The clutch can optionally be designed with or without an overrunning clutch. The basis is the key-type overrunning clutch (**Figure 23**). The clutch is available in two sizes, thus covering a wide range of outputs. A special flange version is available for protecting the flywheel on big balers.

The cam-type cut-out clutch (**Figure 31**) was developed to supplement the ball-type cut-out clutch. The special feature of this clutch is that the torque flow is interrupted at normal speed (540/1,000 rpm), as with the ball-type cut-out clutch, but that a pulsating torque is built up when the speed is reduced. This effect of a pulsating torque makes it possible to clear congestions or blockages at low output. The clutch re-engages when the speed drops further (< 200 rpm).

Radially acting locking cams rest on a spring system with specially arranged active surfaces at the centre. These surfaces are designed in such a way that the contact angle is approx. 45° during normal torque transmission. When the torque is exceeded, the locking cams are pressed inwards, against the spring and friction force, into a position where the active surfaces are close to self-locking. Owing to the low restoring force and the inertia of the mass of the locking cams, the cams cannot move back into the driving grooves of the housing until a sufficiently low speed has been reached (< 200 rpm). In this context, the torque pulsates in the transitional phase at low output. This type of clutch is characterised by its good response and its robustness, as a result of which it is used in a wide range of applications.

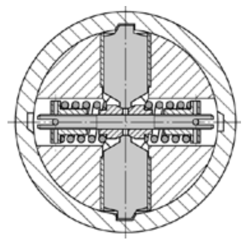
In addition to torque-interrupting clutches, there was also a demand for disengageable clutches that could specifically engage and disengage power take-offs, for example. Suitable for this purpose are friction clutches (**Figure 32**), where the spring force can be eliminated by means of appropriate measures. In the simplest design (based on the clutch in **Figure 21**), levers are arranged on an additional plate to this end. A clutch release bearing mounted on the machine frame is used to lift the pressure plate, against the spring load, thus eliminating torque transmission. The small transmission ratio of the disengaging levers results in very high actuating forces that have to be supported by the machine frame.The applications for this clutch type are limited to special cases where only occasional disengaging is necessary.

The increasing complexity of the machines and the growing number of supplementary functions meant that there was a need for a hydraulically disengageable clutch with high torques that did not introduce any supporting forces into the machine frame. A hydraulic shifting mechanism integrated in the clutch was therefore developed on the basis of the clutch in **Figure 29**.

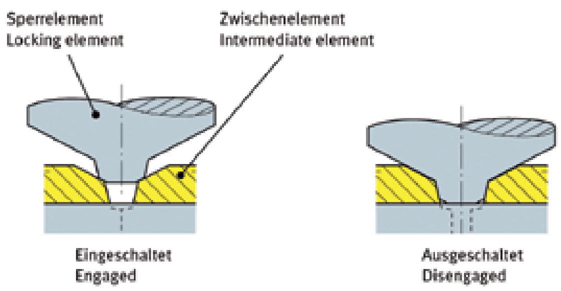
Fig. 30

Adjustment positions

Fig. 31

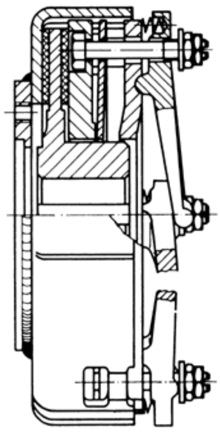


Wirksystem der Nockenschaltkupplung
Operating principle of the cam-type cut-out clutch



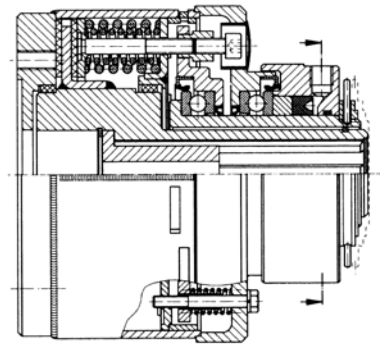
Cam-type cut-out clutch 2500 Nm, year of construction 1984

Fig. 32



Disengageable friction clutch 450 Nm, year of construction 1964

Fig. 33



Disengageable friction clutch 2500 Nm, year of construction 1990

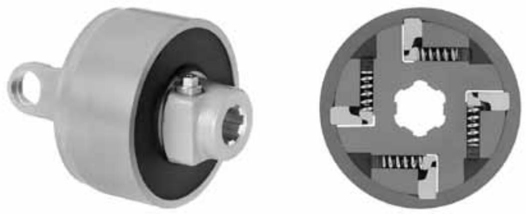
A unit with thrust bearings and a hydraulic cylinder is mounted on the basic clutch and connected to the releasing screws (**Figure 33**). If the cylinder is then pressurised, the releasing screws lift the pressure plate. This interrupts torque transmission. Because of the thrust bearings, the clutch can remain disengaged for lengthy periods of time. Consequently, the clutch is suitable for a wide range of applications.

The cam-type cut-out clutch (**Figure 31**) very soon set technical standards. However, it can only be fitted to free shaft ends. Particularly on power take-offs, there is often no free shaft end available. Consequently, the key-type cut-out clutch (**Figure 34**) with free shaft passage was developed. The cut-out function with the specially designed active surfaces corresponds to that of the cam-type cut-out clutch. The length of the locking keys, and thus the overall length of the clutch, is determined by the number of springs that can be fitted, thereby defining the torque levels. In addition, the locking elements can be arranged over a larger diameter. This modular clutch permits a large torque range of up to 15,000 Nm that cannot be covered by the clutches discussed previously.

A further clutch for high torques at low speeds was developed on the basis of the star ratchet (**Figure 11**). The larger outer diameter makes it possible to use configurations with 12 cams. Similarly, the shaft diameter can be designed for higher torque capacities (**Figure 35**). The torque is determined by the number of cams, which can be arranged in several rows. This clutch is mainly used as an in-built clutch on the feed units of choppers and the spreader units of muck spreaders.

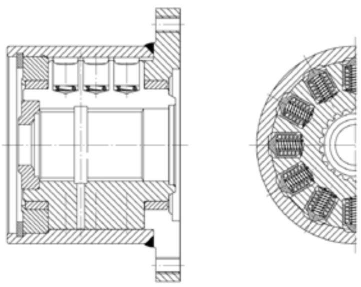
The desire for maintenance-free devices and the avoidance of severe torque shocks led to a hermetically sealed, oil-filled clutch (**Figure 36**). Its structure is similar to that of the key-type cut-out clutch. It is designed as a maintenance-free version for long slipping times and low engaging speeds. The oil contained not only offers low friction, but also reduces the engaging speed, since the locking grooves in the housing fill up with oil during the slipping phase and this oil has to be displaced by the locking keys before re-engaging. The design of the locking

Fig. 34



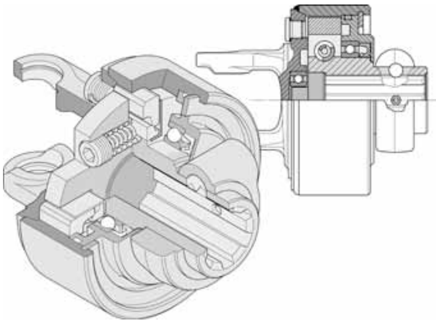
Key-type cut-out clutch15000 Nm, year of construction 1992

Fig. 35



In-built radial pin clutch 2500 Nm, year of construction 1995

Fig. 36



Hydraulic torque control 15000 Nm, year of construction 1997 elements and the degree of oil filling permit defined re-engaging speeds. This clutch is designed for high torques and high rpm speeds. Given its special features, it is primarily used as an in-built clutch to protect the flywheels and feed rotors of big balers.

Worldwide applications

From the 1970s, increasing globalisation resulted in European agritechnical equipment also being used more overseas. Consequently, manufacturers of driveline components set up branches abroad, initially in the USA and later in Asia.

In the USA, PTO drive shafts and overload clutches are predominantly designed and produced by the large agricultural machinery manufacturers themselves. Most commonly used as overload clutches are shear bolts and friction clutches, which the user can easily manipulate.

This lack of diversity offered specialised component manufacturers a wide range of applications for modern overload clutches. The increasing availability of driveline components led to in-house production largely being abandoned.

The growing interconnections of internationally operating corporations led to an increasing exchange of knowledge. Moreover, the change in the structure of farming in Europe/Eastern Europe resulted in larger farms and necessitated more powerful, larger tractors and machines with higher performance per unit of area and greater efficiency. Particularly in the case of larger machines, lightweight designs are essential to avoid damaging the ground. Driveline components that meet the respective conditions help to comply with these demands. Consequently, overload clutches with different operating principles are indispensable.

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