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Foreword

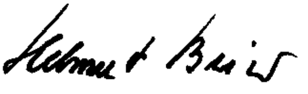
Innovations are shaping our future. Experts predict that there will be more changes in the fields of transmission, electronics and safety of vehicles over the next 15 years than there have been throughout the past 50 years. This drive for innovation is continually providing manufacturers and suppliers with new challenges and is set to significantly alter our world of mobility.

LuK is embracing these challenges. With a wealth of vision and engineering performance, our engineers are once again proving their innovative power.

This volume comprises papers from the 7th LuK Symposium and illustrates our view of technical developments.

We look forward to some interesting discussions with you.

Bühl, in April 2002



Helmut Beier

President
of the LuK Group

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New Opportunities for the Clutch?

Wolfgang Reik
Karl-Ludwig Kimmig
Rolf Meinhard
Christoph Raber

Introduction

For many years, manual transmissions have appeared to be losing ground to automatic transmissions. America and Japan are two places where modern automatic transmissions have almost completely replaced the standard shift. Europe has distanced itself from this development. Automatic transmissions have only become widespread in higher-end vehicles. Those who buy smaller cars still prefer a manual shift, whether for reasons of cost, fuel economy, performance or because they enjoy it.

In reality, automatic transmissions consume noticeably more fuel than manual transmissions, as has been mentioned repeatedly in this lecture series, and therefore do not fit in well with the scenarios outlined in the Kyoto Protocol.

For this reason, efforts are being made by all manufacturers to combine the efficiency of the spur gear transmission, as it is used in manual transmissions, with the comfort of the automatic transmission, or at least to reach an acceptable compromise between the two. And it is here that our reliable old clutch suddenly finds itself at the centre of attention – and from two different directions. The first is the effort to give the manual transmission the required ease of operation, while the second is the drive to reduce the losses of the automatic transmission, for example by replacing the torque converter with an oil-cooled start-up clutch.

We will be reporting here on new development trends in clutches that are significant for clutch automation. We have consciously decided to limit ourselves to the dry clutch, because it is the only way in which it is possible to extract the last reserves of efficiency. This is covered in detail in [5].

Development Goals for Dry Clutches

To prepare the clutch for such future applications, there are a few development goals, which must be met.

Its service life and wear reserve must be further improved, in order to ensure the vehicles service life is comparable to that of a vehicle with an automatic transmission, even if the clutch is operated with a little slip for comfort reasons. Chatter, or more precisely, any type of torque excitation that can cause a slipping clutch, must be further reduced. This applies not only to dry clutches, but also to wet clutches because of oil ageing, and represents a special challenge.

Precise controllability, meaning sufficient sensitivity at low levels of hysteresis, is required for automatic systems. The operating effort must also be as low as possible. This very property may be the key to whether comfortable clutch systems can be realised.

The smaller the operating load required to modulate a clutch, the sooner it appears possible to eliminate torque fluctuations that arise directly or indirectly from the clutch. The fact that this may be necessary in the future is indicated by efficiency-optimised drive trains, which have almost no damping themselves. Thus, torque fluctuations of only 1 Nm, elicited by the slightest fluctuations in the coefficient of friction or geometric inaccuracies, can cause chatter-like phenomena. Based on an engine torque of, for example, 400 Nm, this means a torque inaccuracy of only 0.25%. It is nearly impossible to reduce these torque fluctuations with purely mechanical measures. Much promise is shown in fast actuators, which can help eliminate these slight torque fluctuations. It is also expected that low operating loads on the clutches will reduce cost and save space for the actuators.

New actuator concepts may also be possible. The transition from hydraulic to electric motor actuation was only possible in the past because the way was prepared by actuation load reduction in the clutch, mainly by the SAC.

State of the Art

As early as eight years ago, LuK presented the self-adjusting clutch (SAC), which was able to reduce clutch operating loads for the first time [1].

Figure 1 compares the design of this SAC with that of a conventional clutch. The essential feature is a ramp ring, which the wear adjustment system uses in combination with a sensor diaphragm spring to trigger the adjustment mechanism. For an exact description, see [1-4].

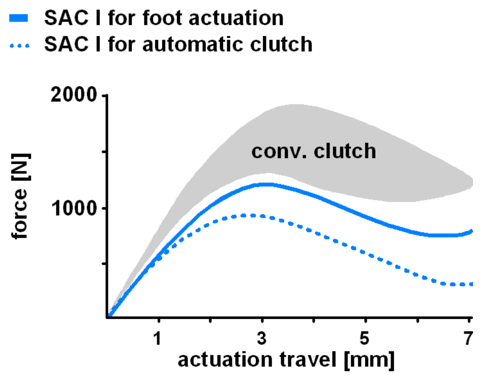

Fig. 2: *Operating Loads of Conventional and Wear-Adjusting Clutches*

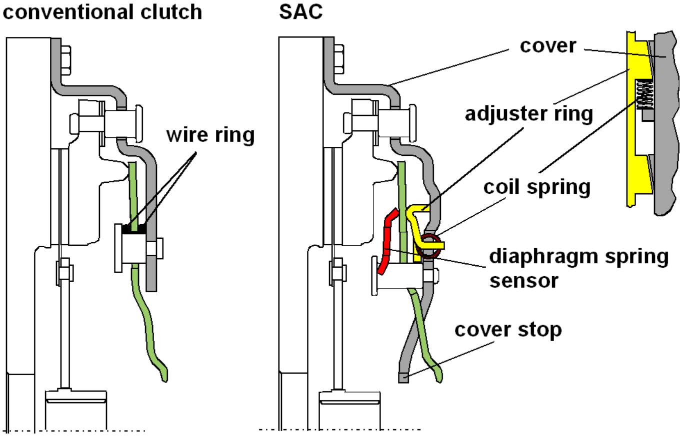

Fig. 1: *Conventional and Wear-Adjusting Clutches*

In the meantime, the SAC has become quite widespread. In 2003, it is expected to make up 40% of LuK's clutch production in Europe (figure 3). This development is supported by the rapid increase in torques, particularly in diesel engines. With a standard clutch, it would be nearly impossible to comfortably operate many modern vehicles with foot force.

Figure 2 compares the release loads of a conventional clutch, with which the operating load on the clutch pedal increases via the facing wear, with those of the SAC. Here the SAC brought decisive progress. The release loads remain approximately constant over the service life because the internal wear adjustment keeps the diaphragm spring angle constant. Lower loads can be realised even in the new condition. The force can be further reduced if the shape of the load curve can be disregarded because the driver's foot is replaced by an actuator system.

Naturally, force reduction can also occur outside the clutch, for example with an over-center spring on the clutch pedal. But the further away from the clutch the compensation occurs, the less effective it will be due to the falsification that will come into play from the elasticity and tolerance in the long chain of connecting links. For this reason, preference must always be given to reducing the force in the clutch directly. The designs to follow are also limited to this.

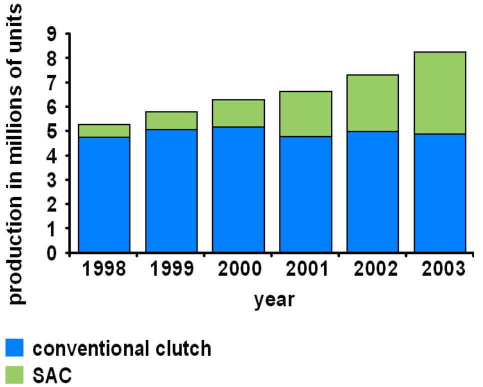

Fig. 3: *Production Volumes of Conventional and Wear-Adjusting Clutches*

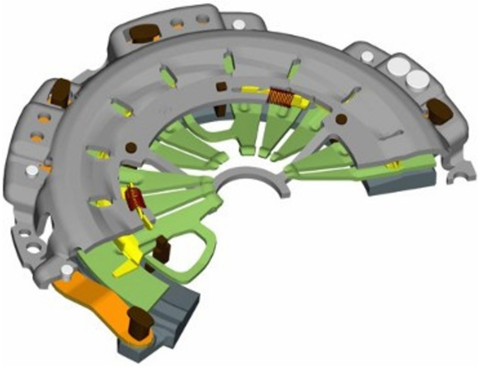

Fig. 4: *SAC II, the Latest Development in Self-Adjusting Clutches*

SAC II

Some years ago, LuK started developing the SAC II, in which the sensor force was not generated by an additional sensor diaphragm spring, but rather by the leaf springs and specially shaped fingers on the main diaphragm spring. What first looked like a cost saving program very quickly proved to be an opportunity to further reduce the operating load, as shown in figure 5. In addition, these fingers allow a flatter operating load characteristic curve, which fits in with the modulation ability of the clutch.

Without attempting to explain the SAC II in too much detail, we can say the following: The sensor force to detect facing wear is generated by the fingers that bend out from the diaphragm spring together with a special leaf spring characteristic curve (figure 4).

One factor that is beneficial in the SAC II is that the sensor force increases over the clutch's release travel. This allows flat operating load curves. With the SAC I, on the other hand, the force drops, which sometimes causes undesirable decreasing operating load curves.

The SAC II is currently undergoing endurance testing and is expected to go into production in 2002.

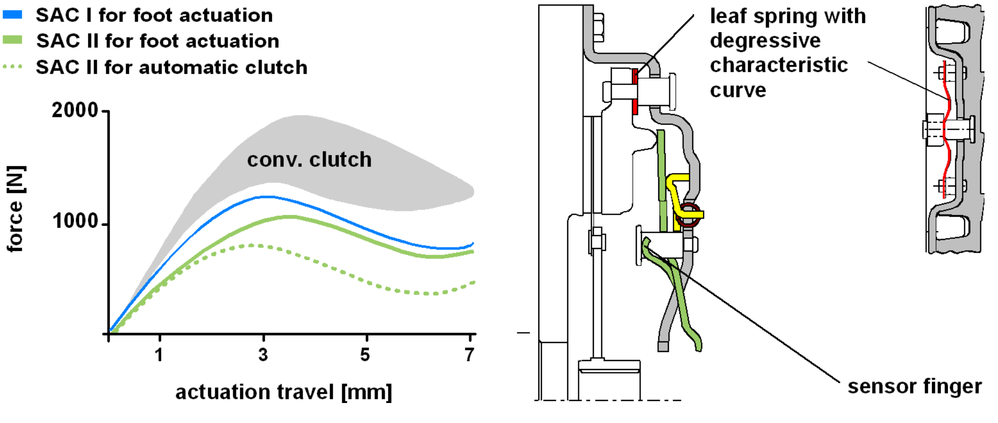

Fig. 5: *SAC II Production Design*

Other Options for Reducing Operating Load

The coefficient of friction is subject to major fluctuations, which depend on the ambient conditions and the previous load. Since a clutch must function safely under all conditions, the safety factors are increased to prevent the clutch from slipping even with low coefficients of friction.

However, this means that the clutch is oversized for normal conditions. This very factor is the starting point for the observations to follow.

Figure 6 shows the occurrence distribution of the clutch torques over the safety factor S (= transferable torque/maximum engine torque). All of the parameters that influence the coefficient of friction, such as temperature, service life and wear, are taken into consideration here. According to this, the clutch can generally transfer 1.5 to 2 times the engine torque.

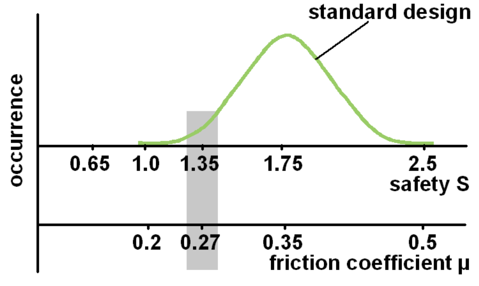

Fig. 6: Occurrence Distribution of the Clutch Torques over the Safety Factor S. Corresponding Coefficients of Friction μ can be Approximately Assigned to the Safety Factor.

The unit is designed to guarantee torque transferability even in the most unfavourable cases. The occurrence distribution thus must not extend below a guaranteed transferability of S = 1.0. In a properly designed clutch, the occurrence distribution curve will thus “end” at S = 1.0, as shown in figure 6.

Since the variation in the clutch torque depends mainly on the changing coefficients of friction, an approximate scale can be created with coefficients of friction. We now can see that even at a coefficient of friction of 0.2, the clutch still transfers the engine torque. At a coefficient of friction of 0.27, 1.35 times the engine torque is transferred. This happens to correspond to the usual design criteria: A design coefficient of friction of 0.27 requires a slip safety of 1.35.

And here begins the hypothesis:

If we did not have to worry about the seldom-occurring slip safety of < 1.35, the occurrence distribution could be shifted 0.35 to the left (figure 7), and the coefficient of friction scale with it.

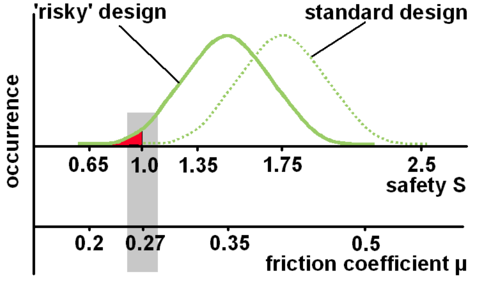

Fig. 7: Occurrence Distribution Curve Shifted S = 0.35 to the Left. The Red-Shaded Area Indicates Insufficient Transfer Safety.

This results in a clutch design with a clamp load that is reduced by a factor of 1.35 and an operating load reduced by nearly 30%, but at the cost of insufficient transferability for the relatively rare cases of a low coefficient of friction.

When such a case occurs, the clamp load can be increased by pulling on the diaphragm spring fingers. The operating load curve is thus extended toward the left into negative loads (figure 8). A clutch such as this is obviously unsuitable for foot operation. However, electric motor actuators can in principle apply load in both directions. The size of the motor essentially depends on the maximum load, and would be 1.35 times smaller for a clutch like the one we are currently discussing. In this way, we can achieve an effective load compensation within the clutch. Special attention must then be given to changing the direction from push to pull without any play.

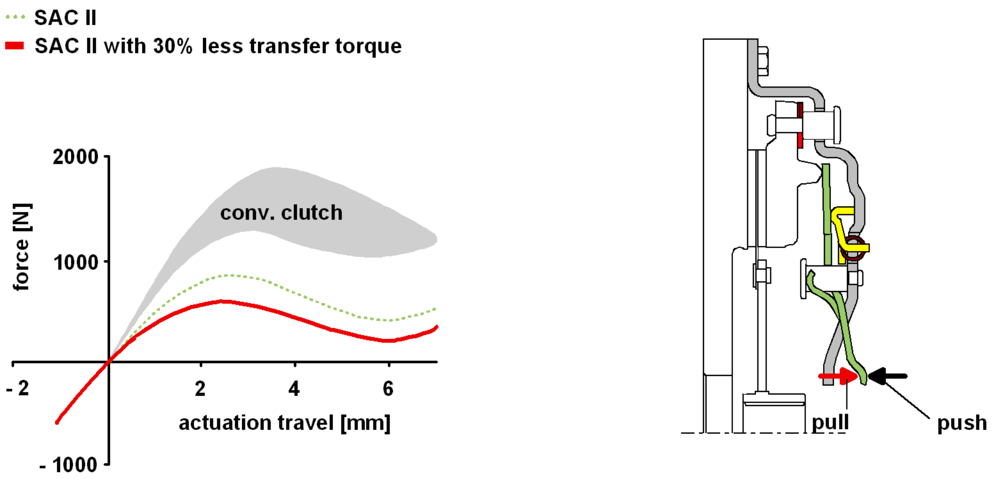

Fig. 8: *SAC II with 30% Reduced Clamp Load. With Unfavourably Small Coefficients of Friction, the Clamp Load is Increased by Pulling on the Diaphragm Spring Fingers (Red Arrow).*

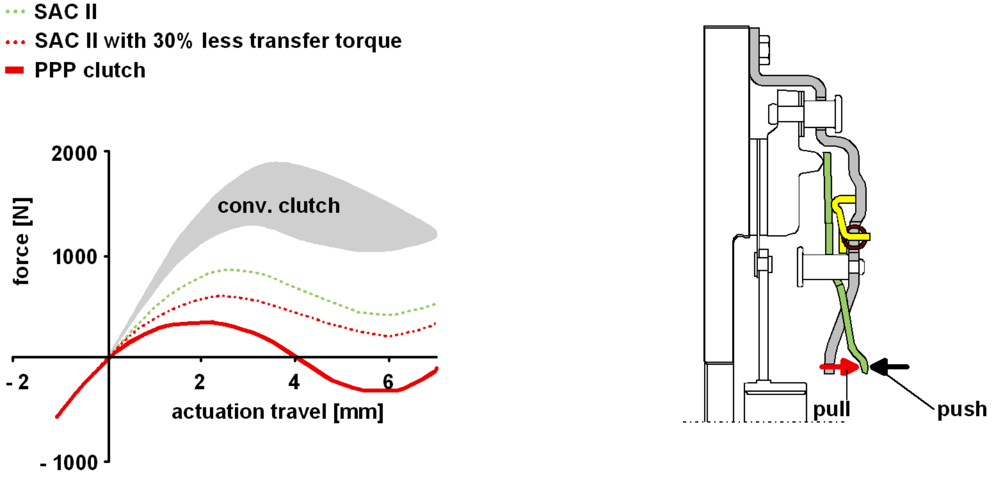

Fig. 9: *Pull - Push - Pull (PPP) Clutch for Maximum Reduction of the Operating Load*

Once we have come to terms with the idea of reversing the operating load, we can achieve a further load reduction by dropping the minimum point of the release load characteristic curve below the zero line (figure 9). A second part of the operating load characteristic curve then falls below the zero line, this one occurring outside the modulation range in the ventilation phase of the clutch (figure 10). The entire operating travel now goes through the following phases: **P**ull - **P**ush - **P**ull. Hence the name PPP clutch.

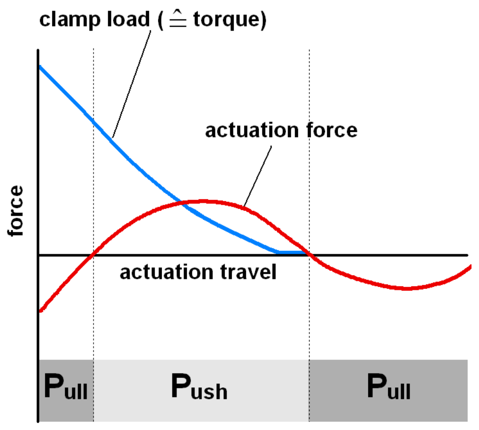

Fig. 10: Operating and Clamp Load Curve over the Operating Travel for a PPP Clutch

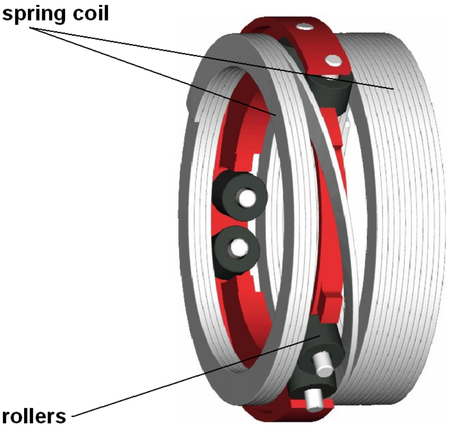

Fig. 11: The Coil Transmission is a Threaded Drive with Slight Feed and Good Efficiency

This construction is also a load-controlled, self-adjusting clutch, but one which now requires a very low sensor force. This can be generated using the leaf springs. The sensor diaphragm spring can now be eliminated. The main diaphragm spring also no longer requires the complex-shaped sensor fingers of the SAC II. From the standpoint of its mechanical design, the PPP clutch is thus as simple as possible, yet still requires the least possible operating load.

However, greater effort must be put into the actuation system, to ensure that it is largely free of play when switching from pushing to pulling operation. Actuation systems that are made up of a long chain of individual components are therefore generally unsuitable. The most direct connection possible between the electrical actuator and the release bearing must be found.

An ideal design is an electrical concentric actuator (ECA), a hollow shaft motor with coil transmission. The coil transmission consists of a thin coiled spring band (figure 11). Rollers, which run between the coils of the spring, generate an axial feed, which corresponds to the thickness of the band. Its function is thus similar to that of a screw, but with very small feed and also with the lowest friction, due to the rollers mounted on roller bearings.

This transmission is therefore perfectly suited in combination with a hollow shaft motor, in which high motor revolutions must be converted into a comparatively small axial travel (figure 12).

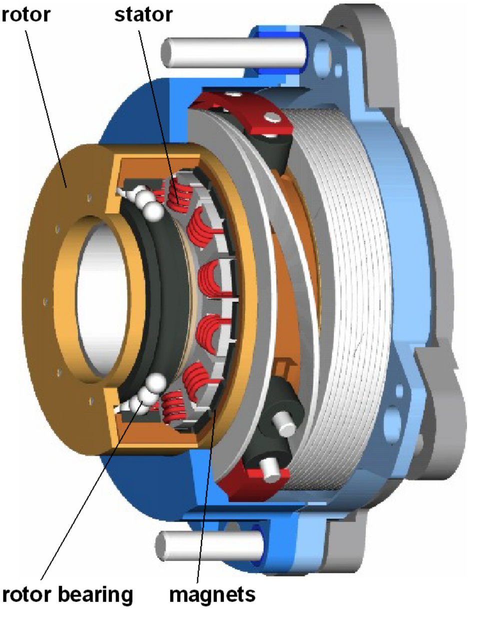

Fig. 12: Electrical Concentric Actuator – a Combination of a Coil Transmission with a Hollow Shaft Motor

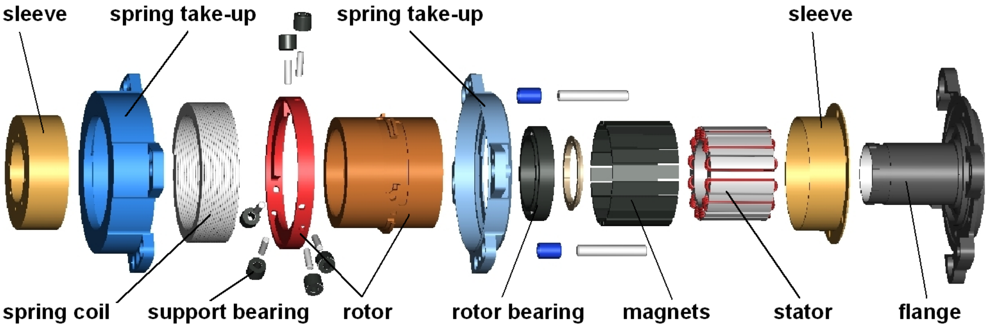

Fig. 13: Components of the Electrical Concentric Actuator

The hollow shaft motor consists of a rotor with magnets and a stator with the coil. The brushless motor is electronically commutated (figure 13).

One especially advantageous feature of this concept is that the entire electrical concentric actuator requires no more space than a hydraulic central release system (figure 14).

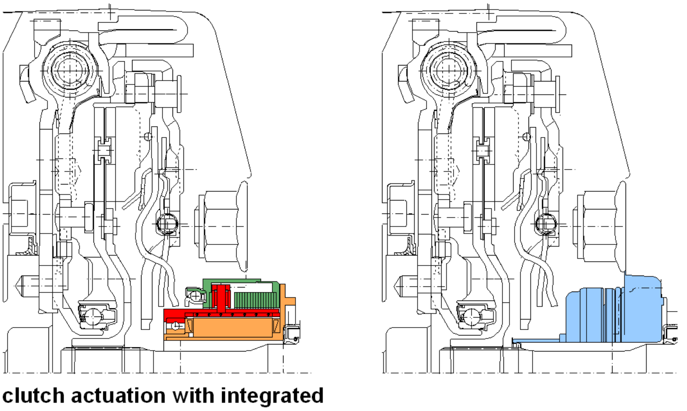

Fig. 14: A Comparison of the Space Required for an electrical concentric actuator (ECA) and a Hydraulic Concentric Central Release System

This actuator operates the release bearing, which must also be designed for push-pull operation. A hollow shaft motor will have a higher mass moment of inertia than a smaller conventional electric motor. This has a negative effect on the adjustment dynamics because some of the energy is needed to accelerate the rotor. The hollow shaft motor is therefore designed for the lowest possible speeds to counteract this effect.

Also conceivable is a 'sleeve release system' (figure 15). The release bearing is screwed forward and back by turning the sleeve. The sleeve is turned using a lever, which can be moved using a spindle drive, for example.

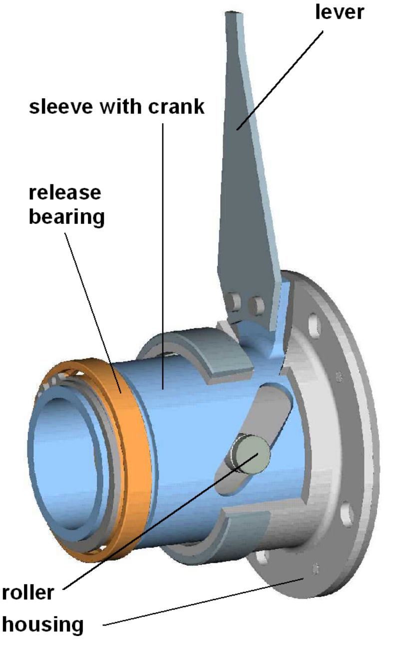

Fig. 15: Sleeve Release System for Pushing and Pulling

Twin Clutches

One type of transmission that is strongly favoured for the future is the parallel shift gearbox (PSG) or twin clutch gearbox, which is described in more detail in a later article [5]. Briefly, the transmission has two input shafts, which must be connected to the engine independently of one another with a clutch for each one (figure 16).

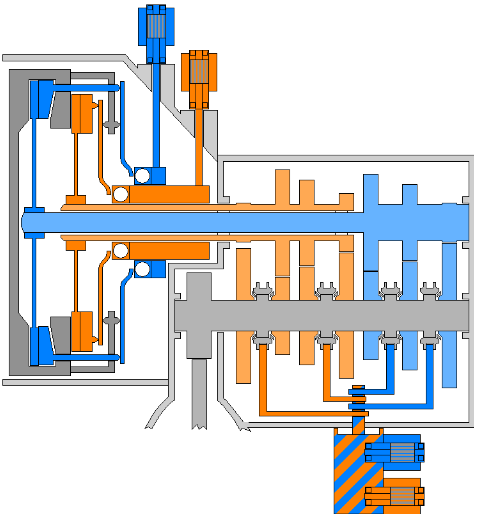

Fig. 16: *The Principle of the Parallel Shift Gearbox (PSG)*

This type of function is not new. There is a version for farm tractors with one independent power take-off shaft, which also uses twin clutches (figure 17). Since the operating load is not as critical for such applications, both partial clutches can even be operated with the same diaphragm spring.

Twin clutches that are suitable for twin clutch gearboxes must have a very compact design and be able to be operated using very small actuators. Therefore, the findings used for load reduction in single clutches should also be used for these.

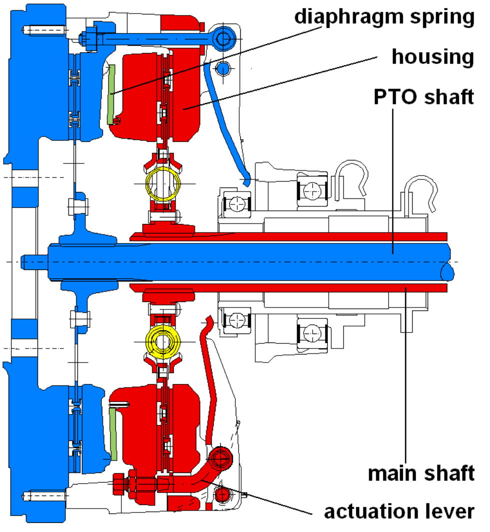

Fig. 17: *Twin Clutch for Tractor for PTO Shaft (Left Clutch) and Master Clutch Operation (Right Clutch)*

Figure 18 shows a twin clutch in connection with a dual mass flywheel. The two clutches are somewhat radially interlaced for reasons of compact design. The larger of the two clutches is logically used as the start-up clutch and connected with gears 1, 3 and 5.

A more compact unit can be achieved by generating the vibrational isolation with slip control [6]. In connection with two torsion-damped clutch discs, it replaces the DMFW in automatic clutches (figure 19). This likewise reduces the mass moment of inertia, thus reducing fuel consumption, which approximately compensates for the increased consumption due to the slip.

So that it is possible to test the PSG completely prior to installation in the vehicle, the twin clutch should be an integral part of the gearbox. It can then be mounted on the engine using a flex-plate, as is done with torque converter transmissions.

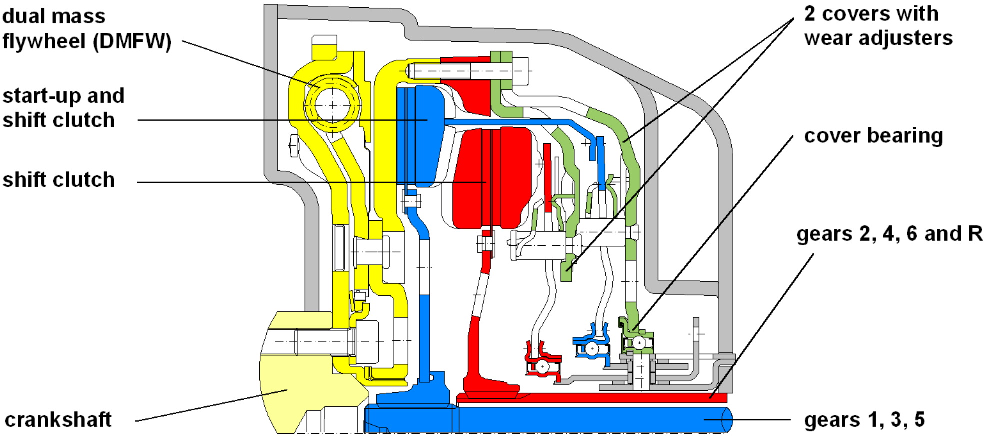

Fig. 18: Twin Clutch with DMFW

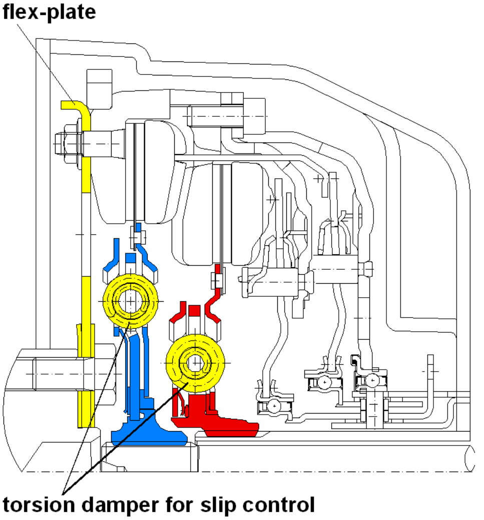

Fig. 19: Twin Clutch in Connection with Slip Control

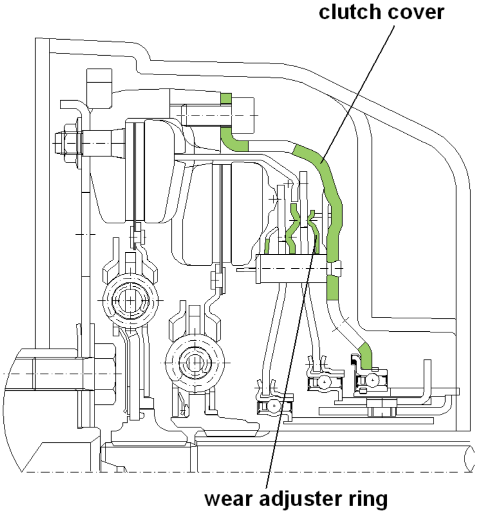

Fig. 20: Twin Clutch with One Cover and One Common Wear Adjuster

For especially tight spaces, it is also possible to mount both clutches with only one cover and one wear adjuster (figure 20). The two clutches must then wear at approximately the same rate. To ensure similar wear rates in the two clutches, the second clutch can be used during start-up. Since the extra wear is generated only in the lower-wearing clutch, this does not shorten the service life of the entire system.

Yet another variant is shown in figure 21. It has two separate wear adjusters, which are placed on either side of one clutch cover. In this arrangement, only three cast friction surfaces are required because the middle mass is used for both clutches. This mass must be thicker than usual because of the poorer heat dissipation and therefore requires internal ventilation.

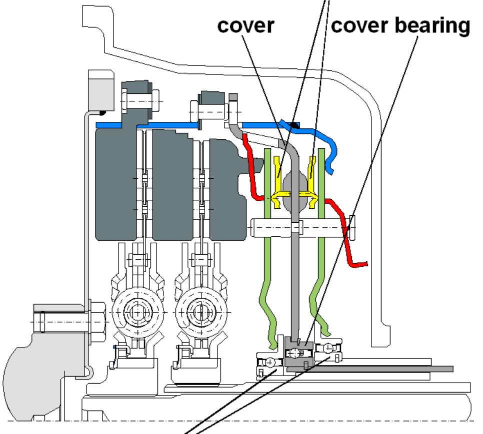

Fig. 21: Twin Clutch with Shared Cover but Two Separate Wear Adjusters

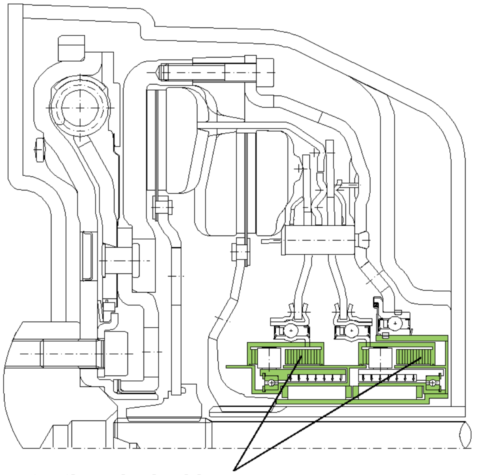

Fig. 22: Vision of a Double Electrical Concentric Actuator for Twin Clutches

All of the twin clutch concepts shown here can be combined with the load reducing measures described before.

Figure 22 shows a variant with a double electrical concentric actuator and push/pull operation, similar to figures 11 to 14. It is an extremely compact unit, which requires the lowest possible operating loads and thus all conceivable load reduction measures. Only then will a smaller hollow shaft motor be able to operate the clutch with a coil transmission.

Another alternative is with a double sleeve release system, which makes fewer demands on the clutch, but requires two externally placed actuators (figure 23).

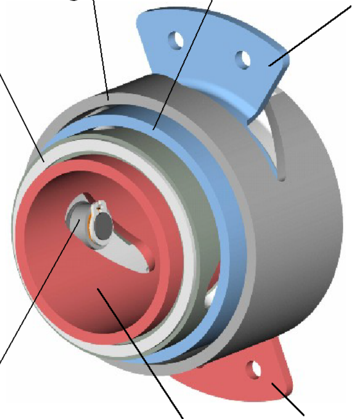

Fig. 23: Double Sleeve Release System

Summary

New transmission concepts whose main purpose is to save fuel lean strongly towards dry clutches, which promise the best efficiency. Demands for less expensive actuators and faster control require clutches with the smallest possible operating loads. LuK is finding new ways to reach these goals.

Development will be particularly focused on the area of dry twin clutches. We have presented several different design options.

Which concept will be given preference will be strongly influenced by space considerations in future. What appears ideal for one project can be unsuitable for the next application.

For this reason, LuK is developing custom solutions for various twin clutch projects.

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