Electromotoric actuators for double clutch transmissions –

Best efficiency by itself

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Introduction

The development of double clutch transmissions is currently being paid considerable attention by almost all vehicle manufacturers, with the prospects of combining the comfort of a stepped automatic transmission and the high basic efficiency of a manual transmission. In addition to the power-transmitting components such as the double clutch and the gear set, automatic actuation of the clutch and gearshift elements in the transmission is especially important, since this has a very significant influence on the aforementioned criteria of comfort and efficiency. In the selection of suitable actuators, consideration is currently being given to a wide variety of competing concepts. This reflects the fact that double clutch transmissions are still relatively new in the market and have yet to go through a process of maturing and development [1]. Initial experience in possible actuator designs has been derived from the automatic transmission sector as well as from automated manual transmissions [2].

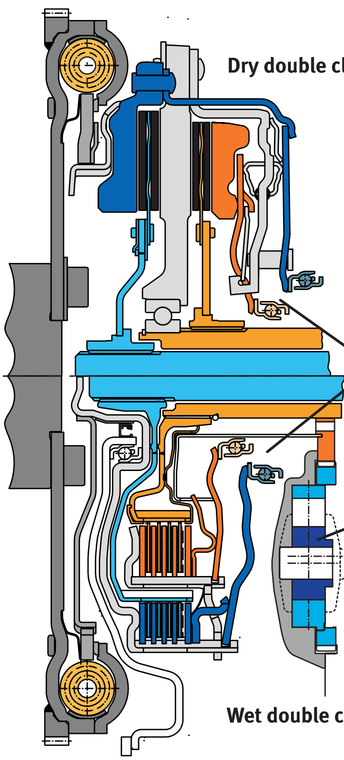


Figure 1 Modular clutch concept for actuation by means of clutch engagement bearing

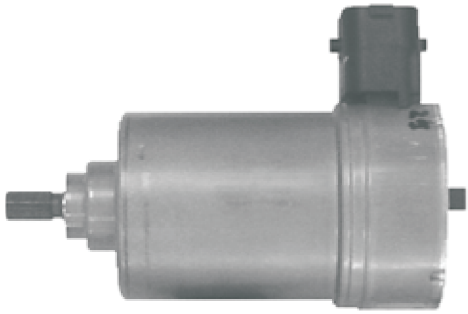
With the modular concept of wet and dry double clutches (Figure 1), LuK has created the conditions for standardisation not only of the basic transmission but also for its automation. As a result, possible synergy benefits have been opened up in terms of control systems and actuators for both variants.

In order to combine the best characteristics of both manual and stepped automatic transmissions into a concept for automation, the actuators for the double clutch transmission must fulfil the following technical requirements:

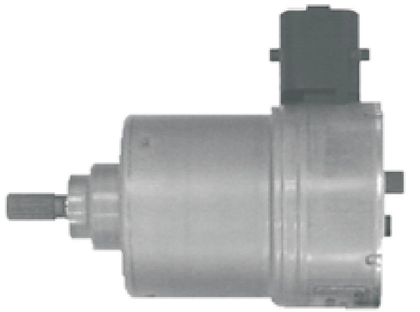
1. Function
    - Highly dynamic positioning
    - Precise controllability
    - Defined failure mode behaviour
2. Operating life
    - Vehicle life 240 000 km and over
    - Free maintenance
    - Robust under all environmental conditions (temperature, vibration, contamination)
3. Integration and packaging
    - Smallest possible packaging with highest possible integration into the transmission
    - Compact components
    - Simple assembly processes
4. Energy requirements
    - Lowest possible actuator energy requirements and thus lowest possible additional fuel consumption
5. Additional requirements to form a hybrid system
    - Energy source independent of the internal combustion engine

The possible choices are restricted particularly by the last two requirements for a minimal influence on fuel consumption and independence of the energy source from the internal combustion engine. As reported at the LuK Symposium 2002 [3] and based on current perspectives following the market launch of the first double clutch transmissions [1], this requirement is best fulfilled by electric motors. LuK has therefore developed, in partnership with a manufacturer of electric motors, a modular range of electronically commutated (EC) motors for driving the clutch and transmission actuators (Figure 2). The dimensions of these electric motors are planned such that they fulfil the performance requirements for dynamic positioning in the various functions of clutch and gearshift actuation.

The use of these electric motors is not restricted to double clutch transmissions. Based on the actuators shown in this paper, actuation elements can equally be derived for automated manual transmissions, transfer cases or hybrid clutches.



max. 170 W



max. 110 W

Figure 2 EC motor range for automated transmissions

Lever actuator for clutch actuation

Initial concept

After the initial decision in favour of electrically driven actuators, the next highest priority is the criterion of integration and design envelope. Solutions are sought that, where possible, use space close to the actuation points in order to avoid unnecessary enlargement of the complete transmission.

Based on these considerations, an actuation concept for the double clutch was developed in which levers in the transmission housing act on the clutch by means of a clutch engagement bearing and the electric motors are mounted directly on the housing. Figure 3 shows a schematic of this lever actuator.

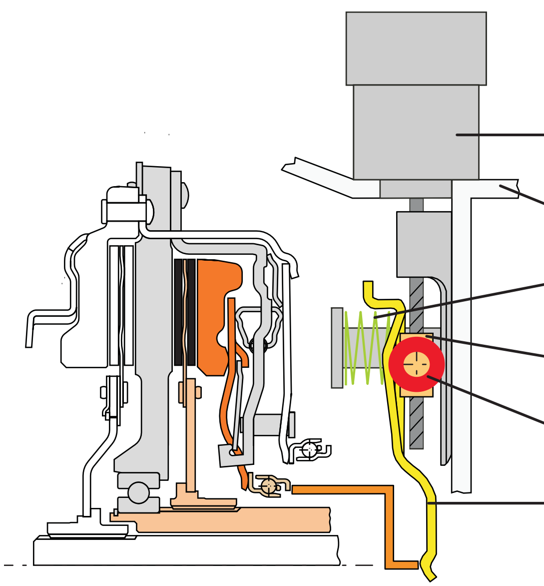


Figure 3 Lever actuator for clutch actuation – principle

The engagement force for the clutch is applied by a preloaded spring in the actuator. This acts at the other end of the lever. Between the two is a movable support point, whose longitudinal motion is generated via a ball screw drive by rotation of the electric motor.

Lever equilibrium

The mechanism of the lever actuator can be explained by means of a simple lever model (Figure 4). The preload of the spring (F_{spring}) and the lever ratio resulting from the adjustment position (x) determine the engagement force of the clutch (F_{clutch}).

An important requirement for actuators in double clutches is passive opening if power is lost (“normally open”) [4]. In this respect, the double clutch transmission differs from automated manual transmissions, where the permissible “freezing” of the actuators (“normally stay”) is the default mode. In the double clutch transmission the “freezing” could lead to an internal locking of both clutches with high energy dissipation and uncontrollable negative wheel torque at the output.

The requirement for a self-opening double clutch system means that the mechanical individual ratios in the actuators must not be self-locking, which means in turn that the electric motor must actively hold the clutch closed during normal operation through continuous current flow.

$$F_{clutch} = F_{spring} \bullet \frac{x}{L - x}$$

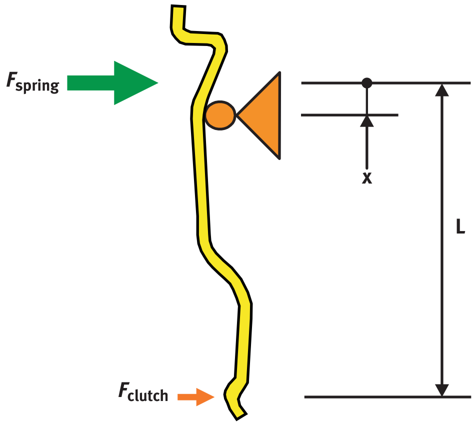


Figure 4 Lever principle

Figure 6 shows the design features that can be used in the lever actuator to ensure this self-opening function. The contact point between the actuator lever and the axially movable roller unit must always have a positive contact angle α. The total of the spring and clutch forces acting, multiplied by this angle α, gives a restoring force ($F_{spindle}$) on the ball screw drive. As a function of the spindle pitch, this in turn creates a reverse torque ($M_{spindle}$) on the electric motor, allowing the support point for the level to be

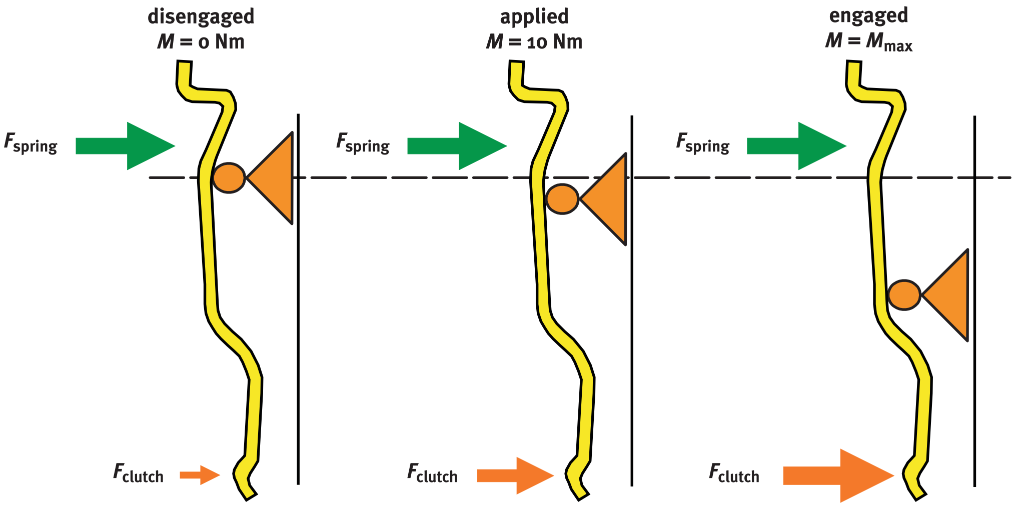


Figure 5 Lever actuator – engagement by displacement of support point

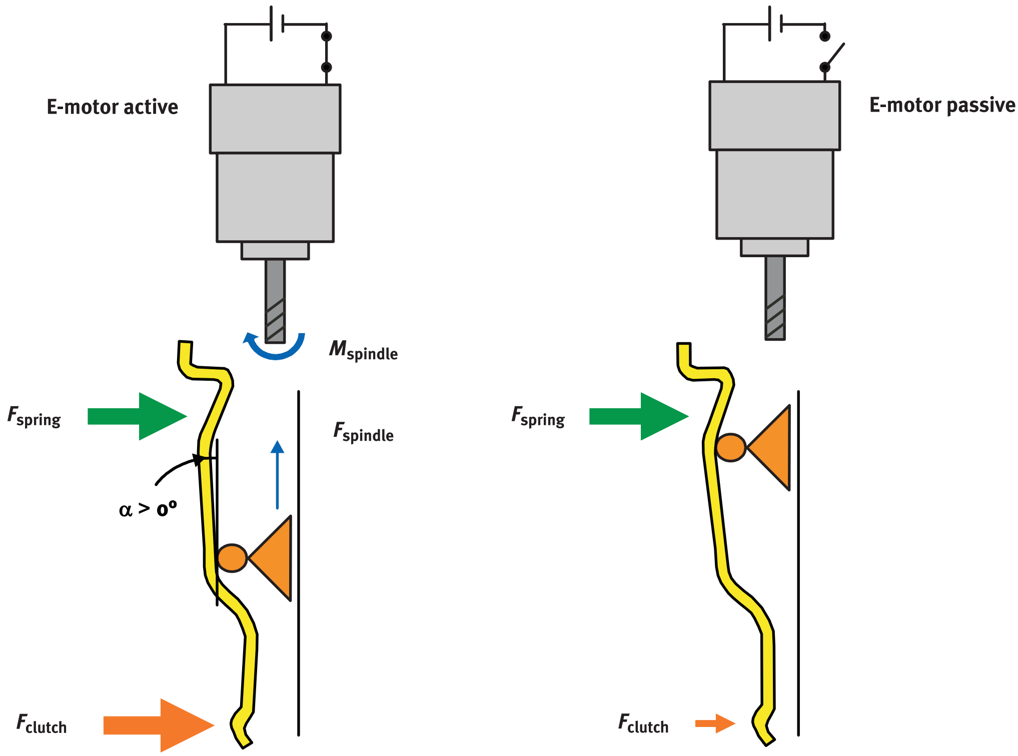


Figure 6 Passive opening under loss of power

pushed back into its original position close to the motor. In order that the electric motor does not act against this reverse torque, the windings must be open in the passive state (Figure 6, right side).

Continuous power fed to the electric motor while the clutch is engaged must not lead to thermal overload of the electric motor. The limit for continuous loading is approx. 20 watts of electrical energy input. The profile of the actuator lever curve (rocker curve) must be designed such that the contact angle α is always positive in order to ensure passive self-opening and is always small enough to require only a low holding torque from the electric motor.

These conditions apply to all tolerances in the clutch curve taking account of fitting of the clutch and actuators in the transmission. Overall, there may be a curve displacement of the order of approx. 50 % of the engagement travel of the clutch in the new condition. Figure 7 shows the influence of curve tolerances on the operating point of the clutch and actuator.

In order to keep the operating point of the clutch as constant as possible, the spring in the lever actuator should be preloaded but with a low stiffness as soft as possible. In relation to the travel of the engagement bearing ($s_{engagement}$), a given lever ratio (constant adjustment position x) will give a straight line (green broken line in Figure 7) as the effective spring curve and the stable operating point is the intersection of these straight lines with the relevant clutch curve.

The following tolerance situations are shown in Figure 7:

a) The clutch with the smallest air clearance and the steepest curve (orange curve) across all tolerances is set to the maximum transmissible torque. The roller unit in the actuator is set to position xa. The angle is then large enough that the electric motor is not overloaded.

b) The actuator in the same adjustment position ($x_b = x_a$) is then matched to the clutch with the largest air clearance and the curve with the smallest pitch (red curve). Since the counterpressure due to the engagement force at the lower end of the lever is smaller, the lever rotates clockwise with a constant adjustment position x and follows the clutch. This theoretically gives the stable operating point c), in which the contact pressure force and thus the transmissible torque are only slightly smaller than in operating point a) due to the soft spring characteristic.

   In practice, however, spreading of the curve under the arithmetical boundary position of the tolerances is normally so great that the minimum angle min permissible for return of the actuator is not achieved at this point. In order to avoid this, the actuator has a lever stop that restricts rotation at the upper end of the lever to ensure a positive angle at all times. The force content of the spring supported against this stop is not sufficient to create the necessary engagement force in the clutch. The torque capacity is not quite sufficient and operating point b) is adopted.

c) In order to achieve the necessary torque capacity in the clutch, the support point in the actuator must be positioned slightly further towards engagement (position x_c). Due to the modified lever ratio, the resulting preload and rigidity of the spring increases in relation to the engagement travel and the clutch is pressed on more.

The system is designed such that the stop is only active with arithmetic tolerance calculation at the most extreme limits of the possible curves. The actuator normally operates without this stop. In this operating mode, the self-setting force equilibrium normally allows self-regulation of approx. 2/3 of a curve displacement of the clutch. The lever system thus operates as a mechanical positional controller subject to the electronic clutch control with only approx. 30 % residual deviation.

Layout and installation

Figure 8 shows a draft design of the lever actuator. The mechanism is mounted using a backplate and two screws on the base of the transmission housing. The electric motor is radially located on the housing from outside and engages with the spindle via a hole in the housing and a centring flange on the actuator mechanism. The springs are arranged concentrically to the screw connections and act on the lever at the lower end. The roller unit driven by the ball screw drive has pairs of rollers that run on the backplate and the lever, providing the movable support point.

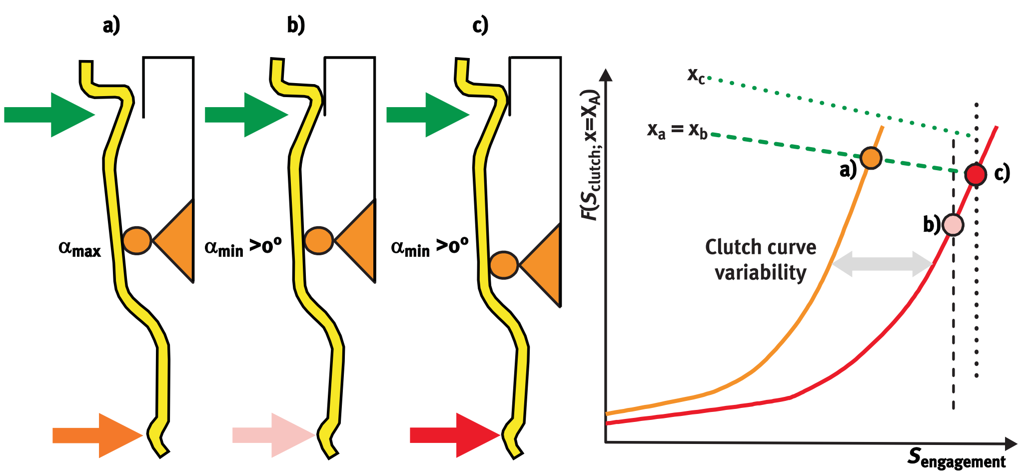


Figure 7 Lever actuator – robustness to tolerances

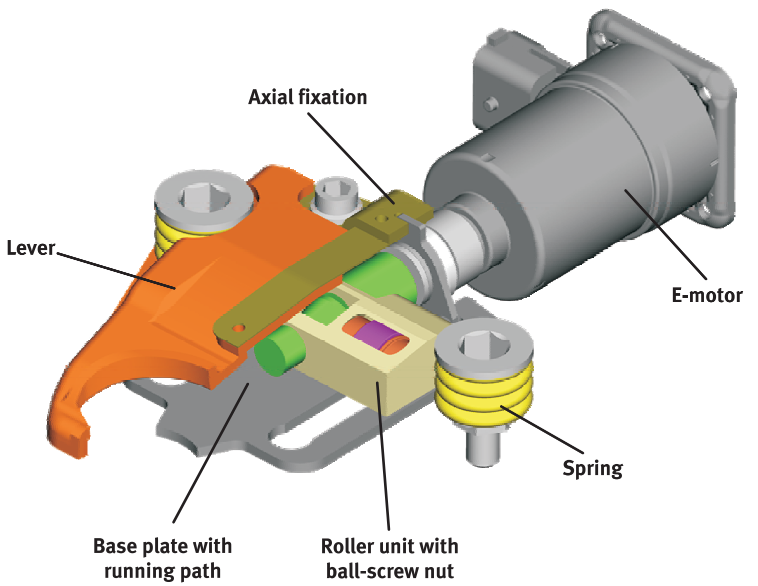


Figure 8 Model of lever actuator

On the basis of the common core elements of electric motor, ball screw drive and roller unit, actuators of the same type can be matched to different design envelopes. Figure 9 shows the examples of a clutch housing with a dry clutch (left) and with a wet clutch (right).

The key performance data required of these clutch actuators are considerable. In order to achieve adequate accuracy for comfortable clutch control, the adjustment position of the roller unit must be controllable to less than 1/10 mm. The actuator must also completely close or open the clutch in 100 – 120 ms with nominal clutch curves. The maximum engagement forces of the clutches may be significantly more than 3000 N, with more than twice these forces applied to the roller unit due to the lever mechanism. There are demands not only on the mechanism but on the electric motors too. They must be able to continuously apply the maximum holding force in critical load spectra at flange temperatures up to 125°C.

With this performance profile, the lever actuator has an ideal combination of controllability, dynamics and efficiency for the automation of dry and wet double clutches, while requiring a minimum of additional space since it is partially mounted inside the clutch housing.

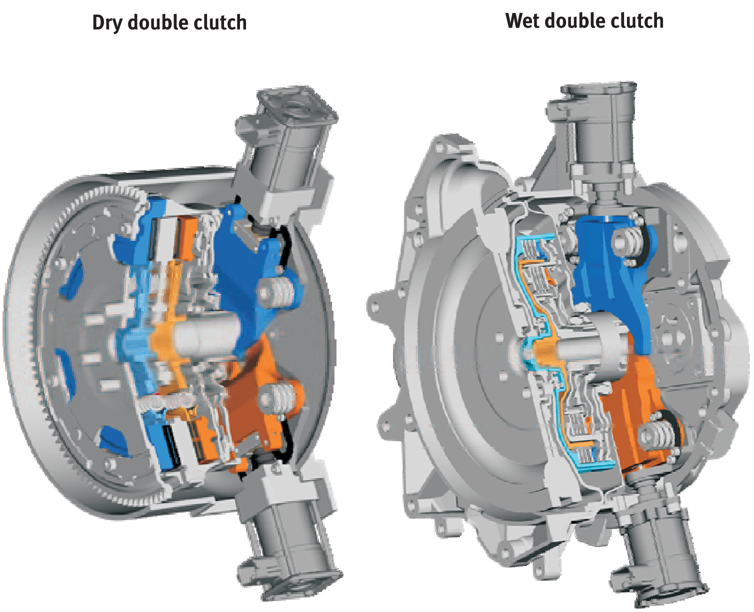


Figure 9 Possible applications of lever actuator

Electric motors for transmission actuators

Active Interlock

Creating an automation system for a double clutch transmission that is driven completely by electric motors requires not only the actuators previously described for a double clutch but also the actuators for operating the shifting elements in the transmission. This can be achieved using electric motors from the same modular concept (Figure 2). In order to minimise the complexity and design envelope for actuating the transmission, LuK has developed the Active Interlock System [5, 6]. It is thus possible to pre-select gears in both sub-transmissions in any combination required using a shared actuator.

The Active Interlock transmission actuator essentially comprises two modules:

1. Shift/select shaft with shift finger unit
2. Drive unit

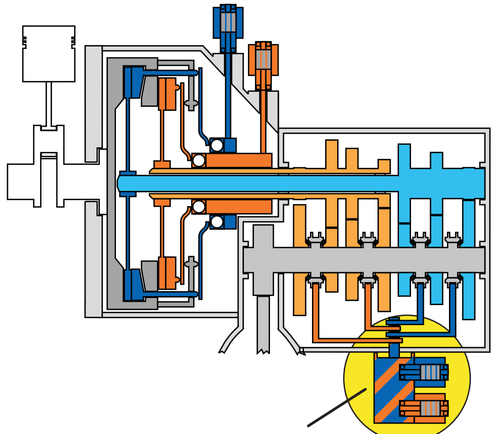


Figure 10 Active Interlock – one actuator for both sub-transmissions

Shift/select shaft with shift finger unit

The shift finger unit, which includes the shift finger as well as the interlock and disengagement elements, forms the interface with the internal selection system of the transmission. It interacts with the shift rails in order to actuate the shift hub assemblies (Figure 11).

The gears are entered by means of the shift finger in a similar way to the actuation of manual transmissions. The special feature of the Active Interlock is the significantly wider mouths on the shift rails in comparison

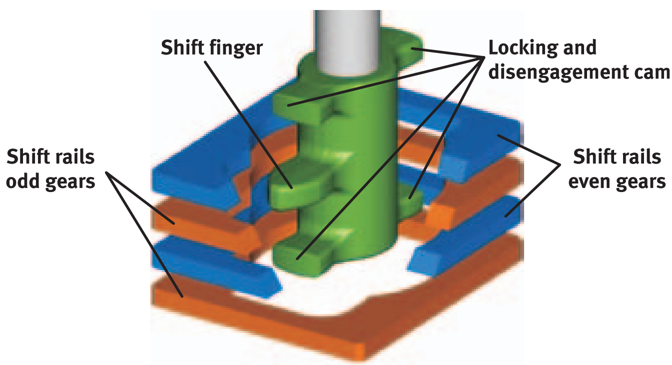


Figure 11 Integration of shift/select shaft with shift finger unit and shift rails

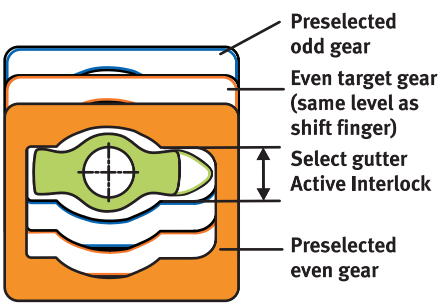


Figure 12 Active Interlock – select gate with engaged gears

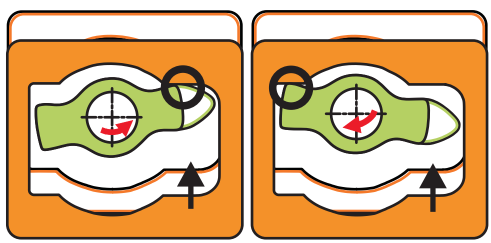


Figure 13 Active Interlock – disengagement of engaged gears

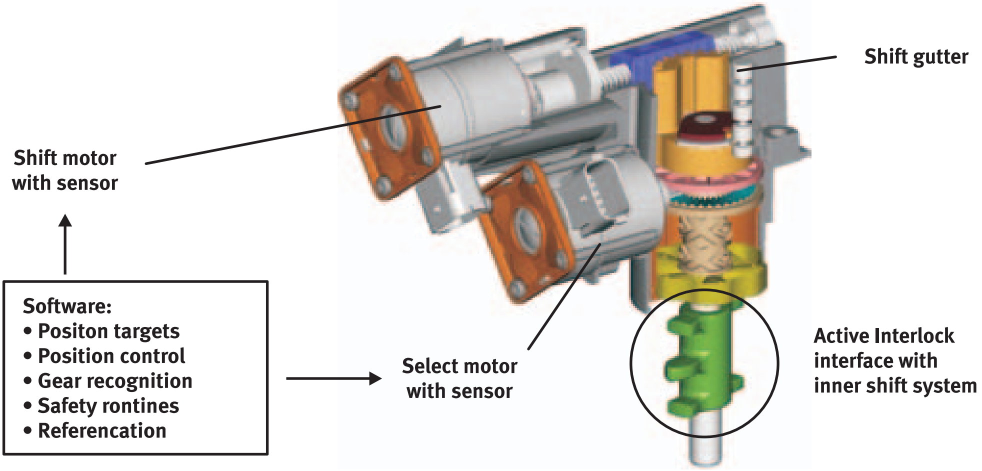


Figure 14 Transmission actuator with shift and select motors

with the width of the shift finger. This makes it possible to reverse the shift shaft even if a gear is engaged and to select a different shift rail with the shift finger (Figure 12).

If a new gear is to be pre-selected, the interlock and disengagement elements simultaneously disengage any previously engaged gear in the same sub-transmission, whereby it is immaterial in which direction the shift shaft rotates or the shift finger moves (Figure 13).

Drive unit

For the shift motion and select motion, one electric motor each is used (Figure 14). The shift motor engages and disengages the gears and moves the shift finger back into the central select position (rotary motion of the central shift shaft). The select motor is responsible for positioning the shift finger in the required gate (axial motion of the shift shaft).

The common base elements of the transmission actuator such as the electric motors and functional elements of the Active Interlock System, can be positioned differently to allow flexibility in the arrangement around the complete transmission. This gives the transmission designer flexibility in design of the internal shifting system and in the arrangement of shift rails. Figure 15 shows two different designs for mounting of the transmission actuator on the transmission.

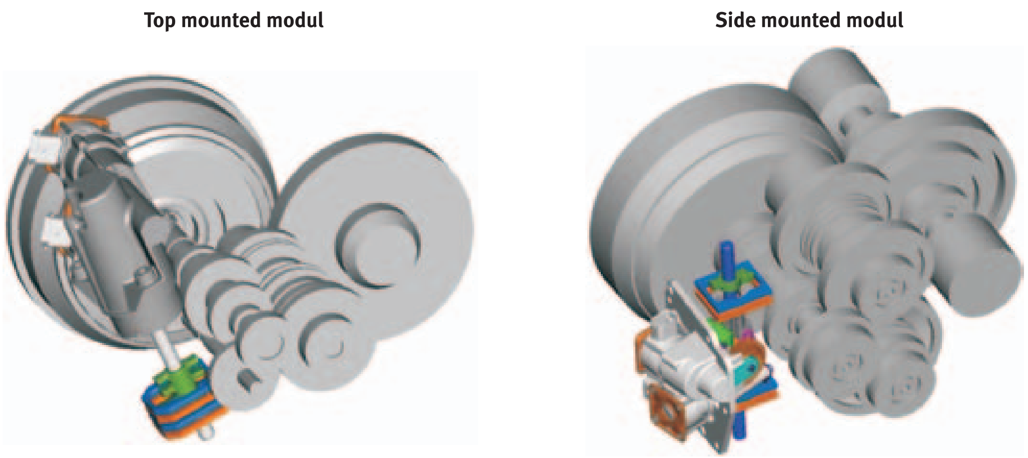


Figure 15 Transmission actuator – flexibility

Single motor transmission actuator

The single motor transmission actuator is a further development of the drive unit for Active Interlock transmission actuation using a new drive unit with the standard shift/select shaft and shift finger unit. The different sub-functions of shift and select are enabled not by separate motors but by rotating a single electric motor in different directions (Figure 16).

Direction of rotation 1:

If the electric motor rotates in direction 1, the shift finger unit is moved (rotated) in the shift direction (red direction in the figures) in order to engage the target gear. In the first motion step, the gear previously engaged in the same sub-transmission is disengaged by means of the shift finger unit.

Direction of rotation 2:

If the electric motor rotates in the opposing direction, the shift finger unit is first brought back to the central position. Once it reaches this position, it is then moved in the select direction, i.e. the shift finger and the disengagement elements are moved along the individual gear gates.

Figure 17 shows the internal construction of a single motor transmission actuator of this type. The various motion phases are shown in Figure 18. The core elements for the shift function in this actuator mechanism are a cam plate with a pin on its circumference to induce lifting and lowering of the shift shaft and roller clutches that fix the currently selected gate position when changing from direction 2 to direction 1 (select to shift) by preventing reverse motion of the cam plate. Shift motion is applied by means of two contra-rotating spindle nuts that, when the drive rotates in direction 1, move away from each other on a spindle with two sections having different

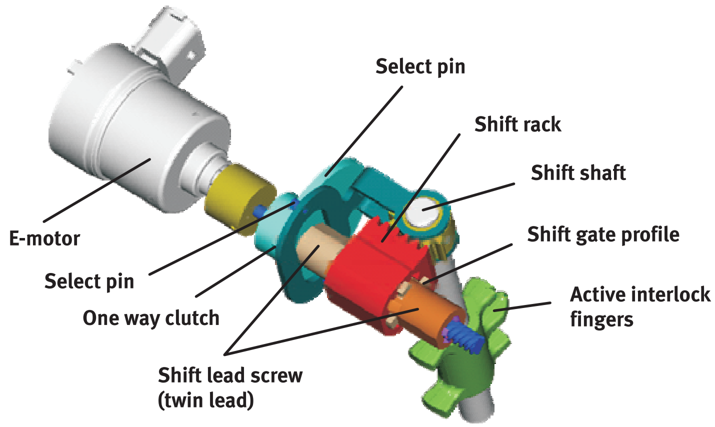


Figure 17 Kinematics of single motor transmission actuator

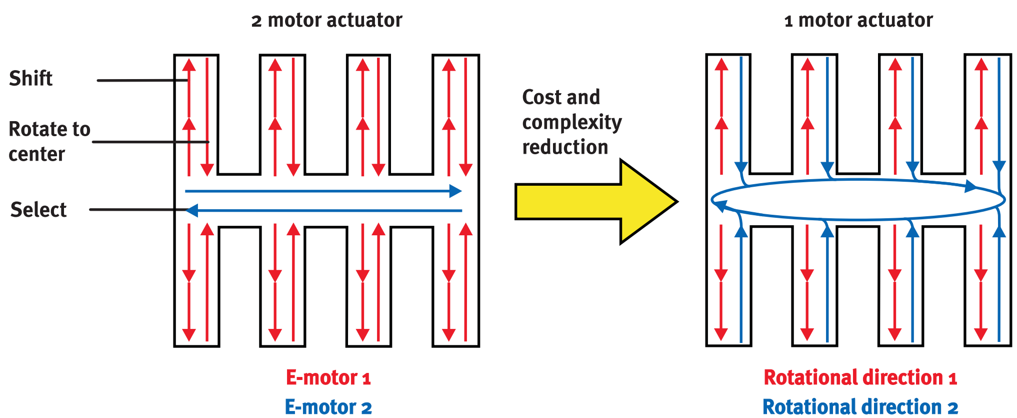


Figure 16 Single motor transmission actuator: basic concept

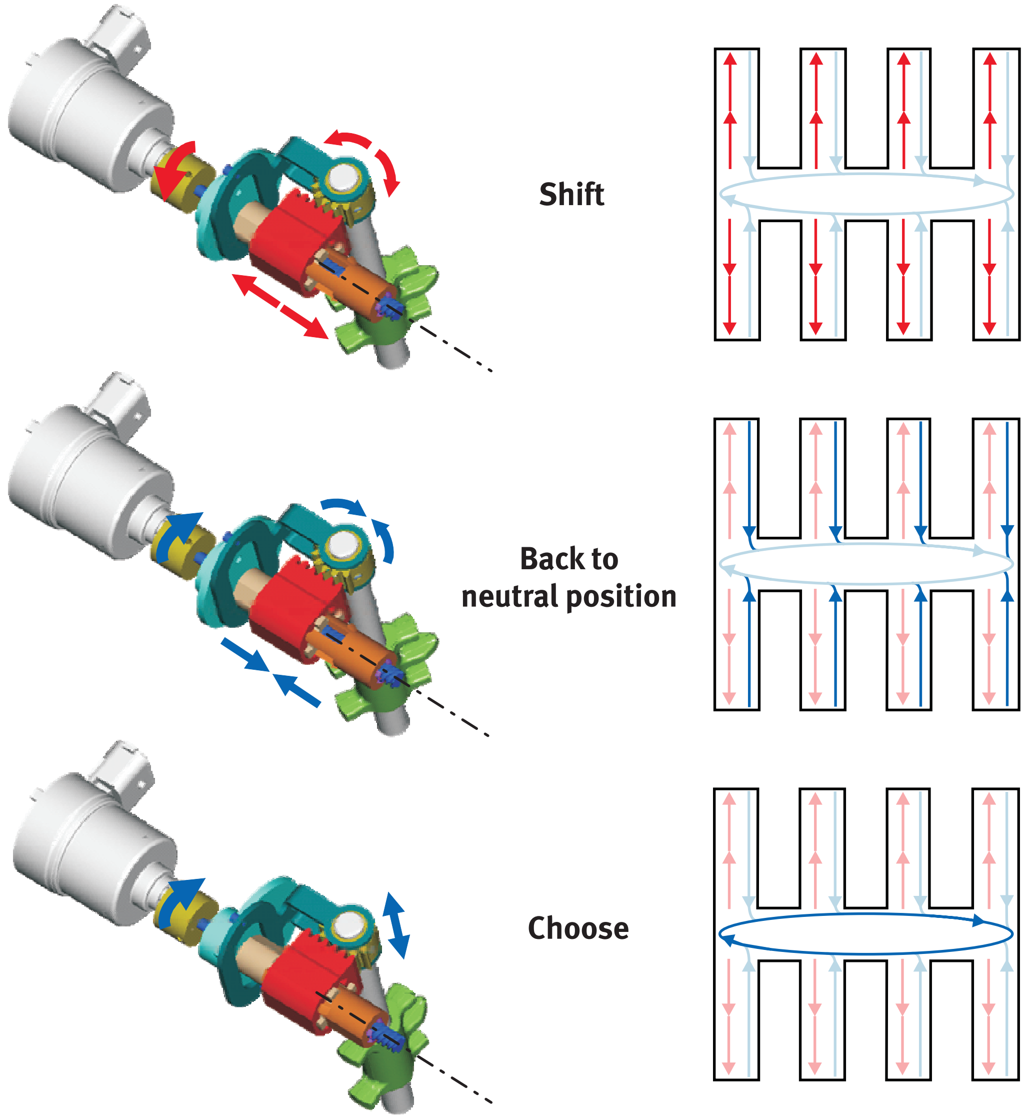


Figure 18 Single motor transmission actuator: function

thread directions. Complementary cam arrangements on the two nuts are arranged such that the motion of one nut is transmitted to one dog while the cams on the other nut run in slots in the dog. This cam/slot combination represents the gear gate profile. The toothed rack on the dog causes rotary motion of the shift shaft and the shift finger is thus turned from its central position in the shift direction.

In Figure 18 the top view shows the shift motion of the single motor transmission actuator. In the central image, the direction of rotation has changed and the shift finger moves back into the central position. This motion stops when both contra-rotating spindle nuts have moved together until they are in contact. The only remaining degree of freedom is rotation and the cam plate is thus driven. The shift shaft rises and falls in the select direction due to the cam plate (Figure 18, bottom view).

Electro-hydraulic power pack

Optimised system approach

Electric motors were initially selected as the drive means for double clutch transmission systems. That is why these purely electromechanical solutions have been presented first. However, electro-hydraulic solutions also meet the requirement for transmission operation that is independent of the internal combustion engine. An electro-hydraulic power pack must therefore be considered in the competition between the various actuation systems for double clutch transmissions.

While purely electromechanical actuation systems have proved more economical for automated clutches and automated manual transmissions, this consideration must be reviewed in the double clutch transmission, which has a further active element, the second clutch. Using the experience gained with add-on electro-hydraulic ASG systems, LuK sees the following potential for optimisation:

1. Integration of the electro-hydraulic power pack in a unit without expensive connection parts such as hoses and cables
2. Integration of an economical pressure accumulator instead of expensive gas accumulators that are often of poor durability
3. Simplifications in valve designs by using valve seats directly in the control plate instead of cartridge valves
4. Rotary slide valve for the shift function in the transmission actuator instead of several linear valves

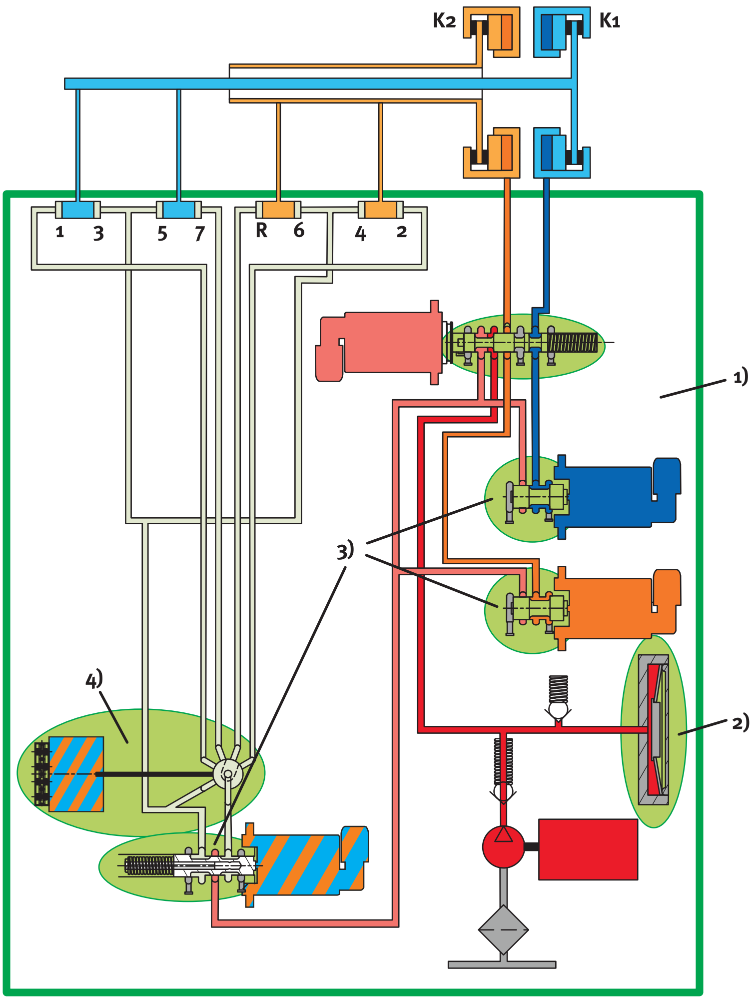


Figure 19 Hydraulic layout of power pack for double clutch transmission

Figure 19 shows the hydraulic layout of an electro-hydraulic power pack in a double clutch transmission and highlights the optimisation potential mentioned in accordance with the numbering above.

As a further possibility for optimisation, the use of ATF instead of hydraulic oil as an actuation fluid is being investigated, potentially allowing a shared oil circuit with the transmission.

Rotary slide valves for the select function

The rotary slide valve for the select function is rotated by means of an electric motor. In the individual positions of the rotary slide valve, each shift cylinder in the transmission is connected to the shift pressure valve such that the hydraulic pressure controlled by the shift pressure valve leads to the motion of one of the shift rails.

Figure 20, top diagram, shows engagement of 1st gear. The rotary slide valve is then used to select the shifting element for 2nd gear and the corresponding gear is engaged. In the bottom diagram in Figure 20, 1st gear is changed to 3rd.

This valve concept for pre-selection gearshifts in the transmission minimises the effort involved and is considerably simpler than comparable existing concepts. In terms of dynamics and failure mode behaviour, it is absolutely equivalent to electromechanical transmission actuation with Active Interlock.

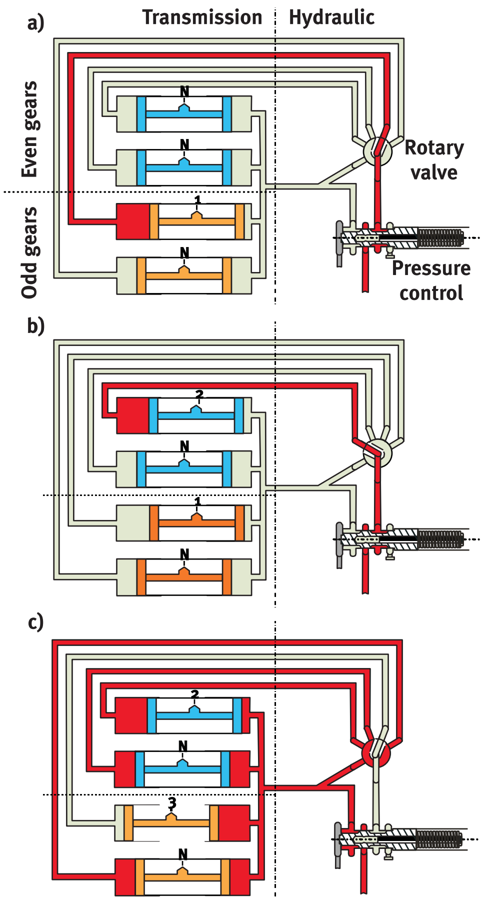


Figure 20 Rotary slide valve for select function

Diaphragm spring accumulator in the power pack

Gas pressure accumulators in electro-hydraulic power packs are often a weak point and are replaced during the life of the vehicle. LuK is working on a diaphragm spring accumulator that can store hydraulic oil by means of two diaphragm springs on both sides of the valve body, Figure 21.

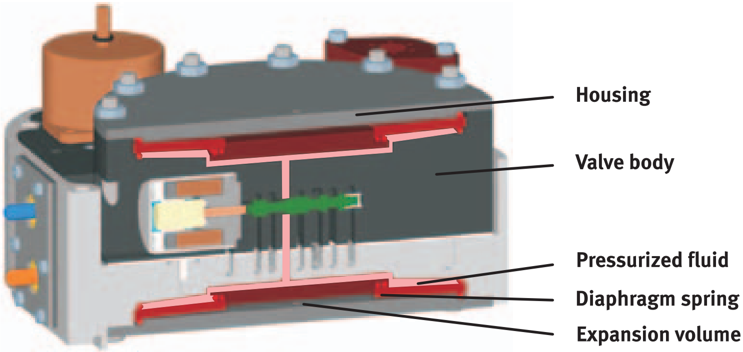


Figure 21 Power Pack with diaphragm spring accumulator

Due to the characteristic curves of the diaphragm springs used, it is possible to achieve a relatively large constant pressure range, allowing a reduction in peak pressure to 25 bar. In comparison with high pressure power packs, this means either reduced leakage with comparable

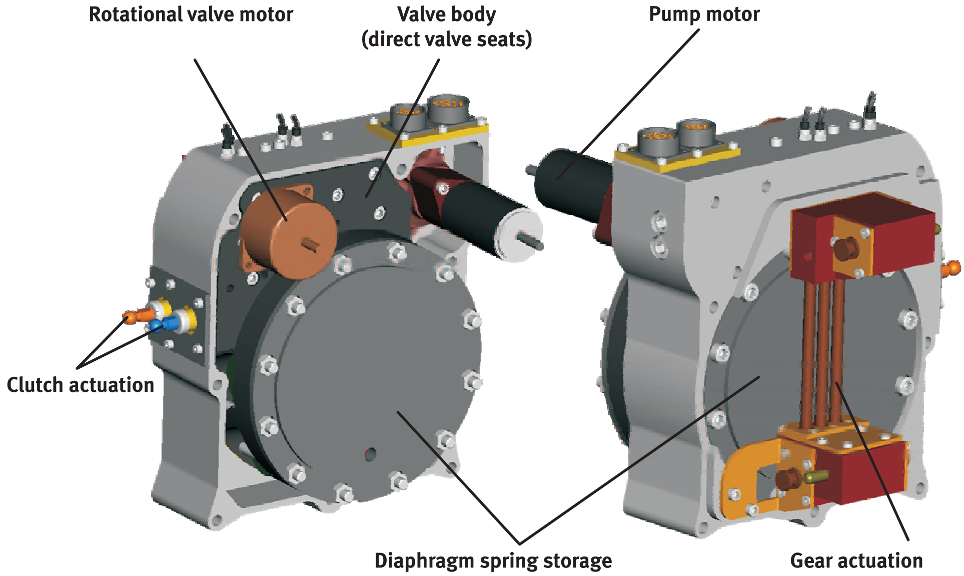


Figure 22 Power pack assembly

valve gap width values or comparable leakage with expanded valve gap width values. In the low pressure power pack shown here, this advantage is exploited such that considerably more economical valve seats located directly in the valve body can be used with a comparable leakage balance.

In addition to the stated characteristics of the low pressure accumulator such as low pressure and life capability, it also has the advantage that it is very well matched to the design envelope of the power pack (Figure 21).

Complete integration

Figure 22 shows the complete design of a power pack with the aforementioned features:

- integration of all functions in a single module
- durable diaphragm spring accumulator
- valve seats located directly in the valve body
- rotary slide valve for the select function.

Initial validation on test rigs has been successfully completed.

Wear adjustment in the clutch engagement system

In the previous paper on the double clutch, wear adjustment mechanisms were also presented for dry double clutches. In optimisation of the com-

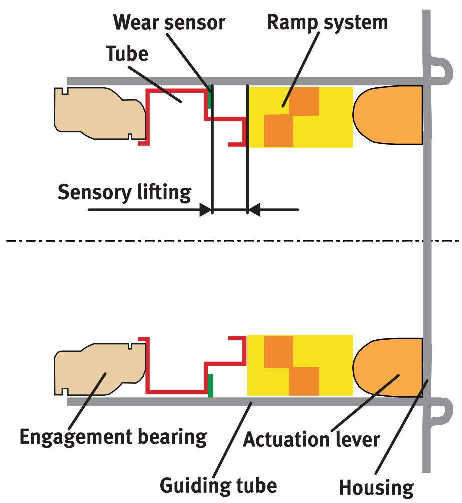


Figure 23 Wear adjustment in the clutch engagement system – schematic

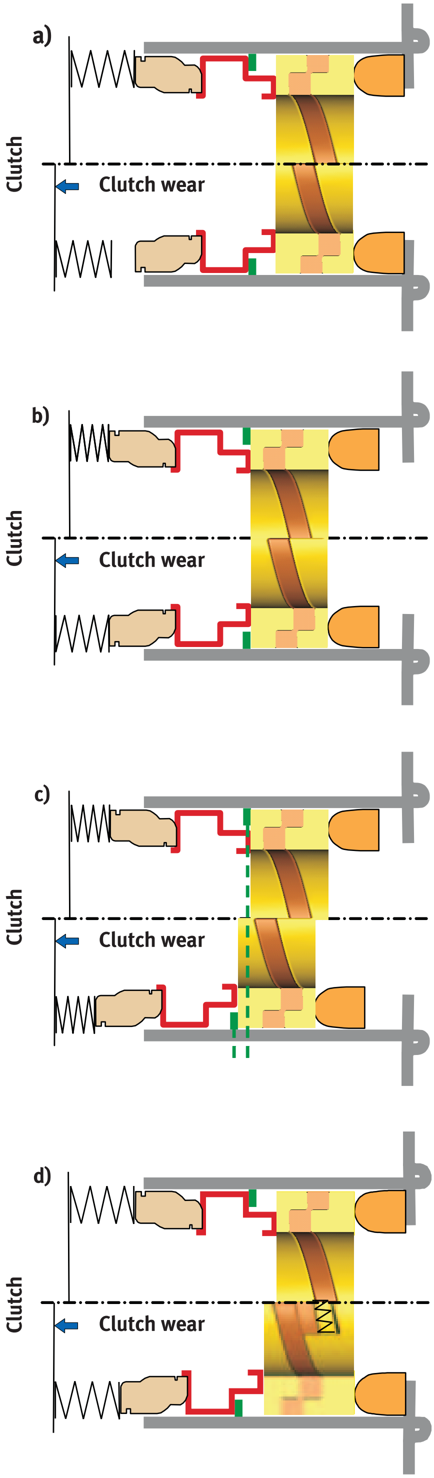


Figure 24 Wear adjustment in the clutch engagement system – function

plete system, the question arises as to how far wear adjustment in the clutch engagement system offers advantages over wear adjustment in the clutch. Cost and space are also important boundary conditions in this exercise.

LuK has developed a concept for compensating tolerances and wear adjustment in the engagement system of a double clutch. The schematic design of this adjustment system is shown in Figure 23.

Between the actuator and the engagement bearing there is a wear sensor, a sleeve defining the stroke to be detected and a ramp system. The function of the adjustment system is shown in Figure 24. The upper half of each of the four diagrams shows the initial system without wear of the clutch, the lower half shows the system with wear.

a) Clutch in position without actuation

b) The clutch is engaged to the last position detected, corresponding in the original condition to the maximum torque capacity (spring compressed in the upper half). Due to additional wear (lower half), the clutch is not yet engaged to its maximum torque capacity.

c) In order to transmit the necessary torque even under wear, the clutch must be engaged further. The wear sensor is displaced (lower half).

d) The wear sensor is self-locking backwards. The sleeve contacts the sensor and thus blocks the return motion of the engagement bearing during clutch release. The ramp mechanism is rotated by preloaded springs, spread and fills the gap occurring as a result between the lever and sleeve (lower half).

Figure 25 shows a design with two self-adjusting clutch engagement systems for a double clutch actuation system.

This system is of particular interest where a very compact double clutch can be designed for small torques (small engines) and low wear reserves. Under these conditions, omitting wear adjusters in the clutch simplifies the clutch itself. The additional engagement travel required on the engagement bearings due to wear, which is compensated in the engagement system, can be kept low.

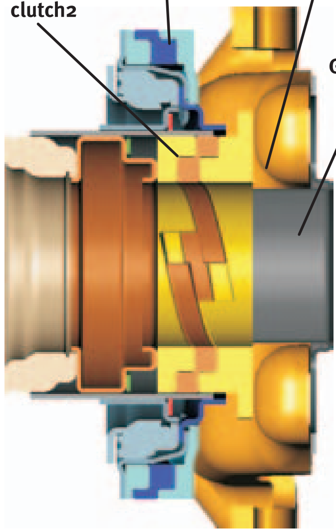


Figure 25 Double clutch engagement system with wear adjusters

Summary and outlook

The double clutch transmission utilising dry clutches currently represents the automatic transmission concept with the greatest potential in terms of cost and efficiency. Actuators driven by electric motors are used to automate the clutch and gearshift.

The electromechanical actuator design presented for the double clutch as a lever actuator offers excellent characteristics in terms of controllability and dynamics. This clutch actuator also requires a minimum of additional energy. Since it is partially integrated in the clutch housing, the actuator occupies only a small proportion of the complete transmission package. This clutch actuator system is supplemented by the Active Interlock shift actuator, also with excellent performance capacity and minimal energy demand. The single motor transmission actuator design offers further potential for reduced cost and design envelope.

Designing the actuator system as an electro-hydraulic power pack is an alternative option with comparable performance capacity. It allows implementation of the optimisation issues described such as a mechanical pressure accumulator, rotary slide valve and simplified valve components. It must be decided for each individual case which actuator system represents the better solution for the specific application of a double clutch transmission.

There may be further potential for simplification of a dry double clutch system in displacing the wear adjuster into the clutch engagement system. Due to the travel and forces involved, such a solution is only conceivable for small torque levels up to approx. 150 Nm.

In wet double clutch transmissions as in classical automatic transmissions, direct drive control hydraulics are the state of the art. The high amount of hydraulic losses with these systems, however, balances out to a large extent the efficiency advantages of the basic transmission. If the wet double clutch transmission is fitted with disc springs and engagement bearings, however, it is possible to use the electromechanical or electro-hydraulic actuators described here in combination with a low pressure coolant oil pump. This gives a system with considerably reduced additional energy requirements.

Furthermore, the electrically driven actuator systems described represent the ideal basis for expansion to give a hybrid in both dry and wet double clutch transmissions. The actions required in the transmission for hybrid functions such as stop/start and coasting with regeneration can be carried out using the integrated drive and independently of the internal combustion engine.

Actuation by electric motors is thus a central factor in determining the success of the double clutch transmission and the expansion of this transmission into a hybrid system. It represents a key component for the dry double clutch transmission and offers enormous potential for reducing consumption in the wet double clutch transmission. Since it can be applied on a modular basis to wet and dry clutches, it is possible to achieve families of transmissions from the same basic transmission and clutch technology matched to the specific application.

Literature

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