Clutch and release system –

Enjoyable clutch actuation!

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Introduction

At the 1998 symposium, the phenomenon of increasing engine torque, primarily with diesel engines, as a result of new technologies (high-pressure injection, supercharging) was discussed in connection with the modulation ability of increased systems [1]. The associated problem of clutch pedal load or pedal travel appeared at that time to have been solved for the long term by the self-adjusting clutch (SAC). A look at the maximum torque curve of a 2.0 liter diesel engine, however, shows that engine torques have risen by about 40 % since 1998. This is due to the continued development of charging and injection technology (see Figure 1).

As a result of this truly “high-performance” work from of our colleagues in engine development, even new and intelligent technologies such as reduced release-load self-adjusting clutches have been pushed to their limits within a relatively short period of time.

The SAC, in combination with an over-center spring on the pedal, sets the standard with regard to good operating load levels for clutch systems currently used in production. The pedal load that can presently be achieved is shown in Figure 2 as a function of engine torque. This analysis is based on a high number of measured vehicles. The range of variation is due, among other things, to different operating travels, together with correspondingly varying ratios in the release system. The target range for the maximum pedal load is 90 N to 110 N. As a result, the SAC, in combination with an over-center spring, can cover applications up to approximately 300 Nm without compromise.

In order to provide the driver with a comfortable pedal load, even with high engine torques, it is no longer possible to restrict ourselves to innovations within the clutch alone. New approaches must be found by considering the entire clutch/actuation system.

Modulating and operating work – state of the art

In terms of ergonomics and comfort, any amount of work required from the driver is contrary to driver desire. Unlike automatic

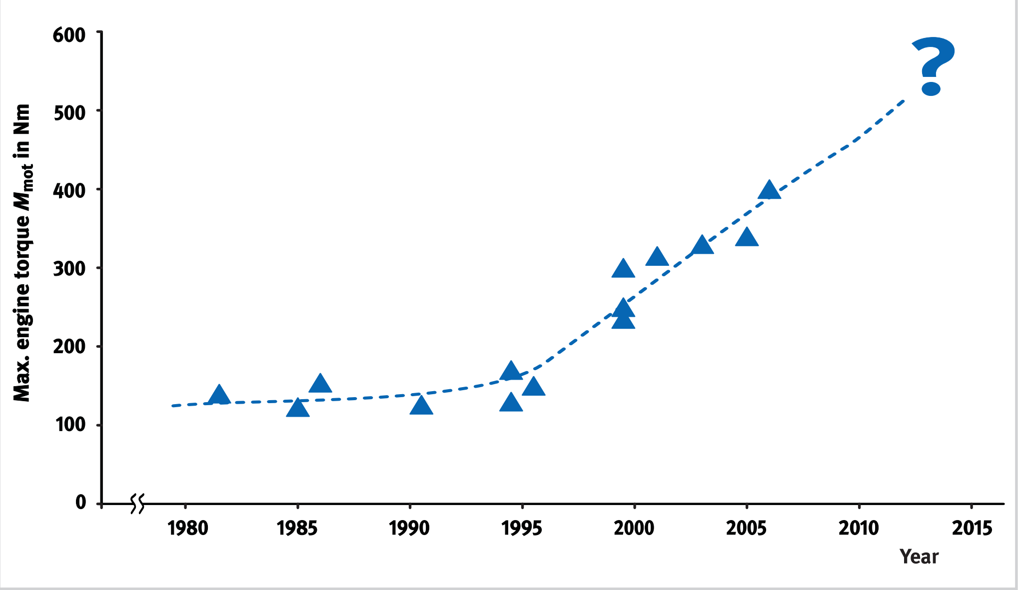

Figure 1 Development of maximum engine torque over time, 2.0 liter diesel

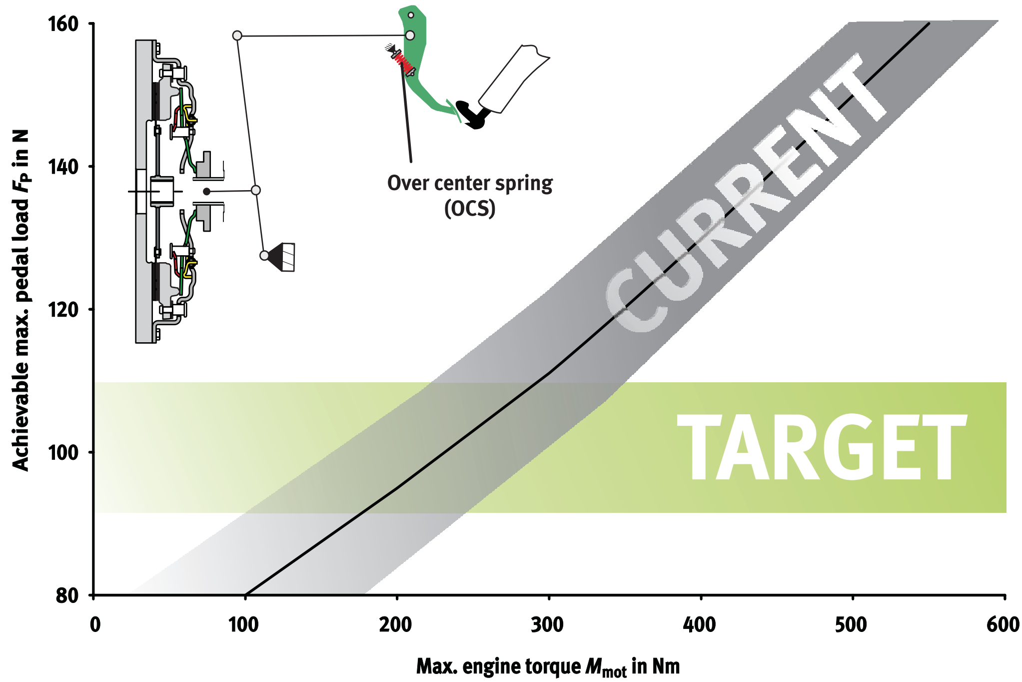

Figure 2 Achievable maximum pedal load in today's systems as a function of engine torque

transmissions, this work cannot be arbitrarily avoided or reduced in a manual transmission, since the driver must be able to modulate the flow of torque between the engine and transmission, and this modulation is supported, to a certain degree, by the work.

The work on the pedal required from the driver to disengage the clutch is currently, depending upon application, between six and twelve joules (mean value without friction). For example, an application with 300 Nm requires eight joules (Figure 3).

In the first approximation, there is empirically a proportional relationship between the operating work and the maximum pedal load. In order to identify the optimization potential of the operating work, its composition must first be analyzed. The following consideration assumes an infinitely stiff clutch cover and a frictionless system.

The work required to compress the cushion spring corresponds to the area below the cushion spring characteristic curve and will be considered as the 100 % reference value (Figure 4). At an engine torque of 300 Nm,

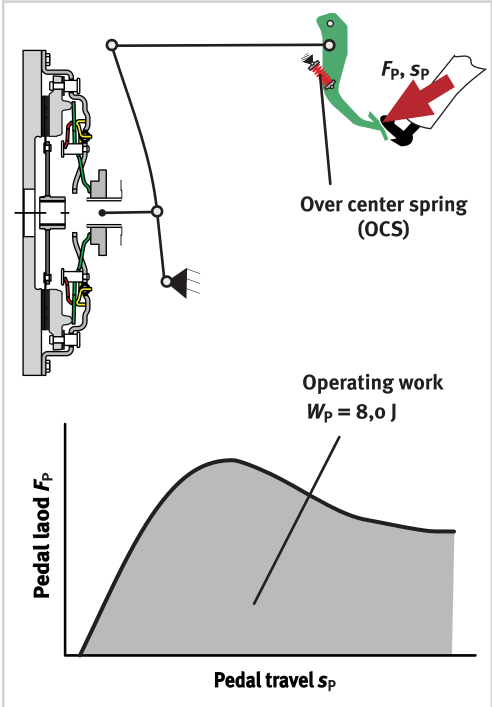

Figure 3 Operating work on the clutch pedal, based on the 300 Nm example application

this work corresponds to about 1,0 J. This work is required so that the driver can modulate the torque transmitted by the clutch. Therefore the clutch disc must not be infinitely stiff, otherwise only a digital torque transfer would be possible.

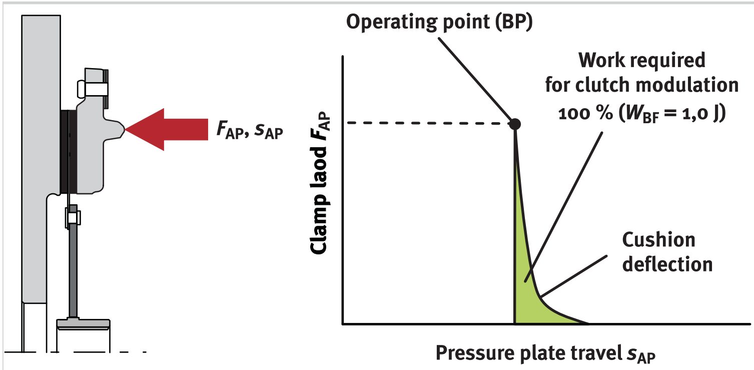

Figure 4 Torque transfer and modulating work in a normally open clutch

So that the clutch cannot be engaged, but rather disengaged by the driver, the cushion spring work, as is known, is done by an energy accumulator in the form of a space-saving diaphragm spring. The work required from the driver to release the clutch disc is, therefore, the resultant of the diaphragm and cushion springs. Work must also be done to provide clearance to the clutch disc.

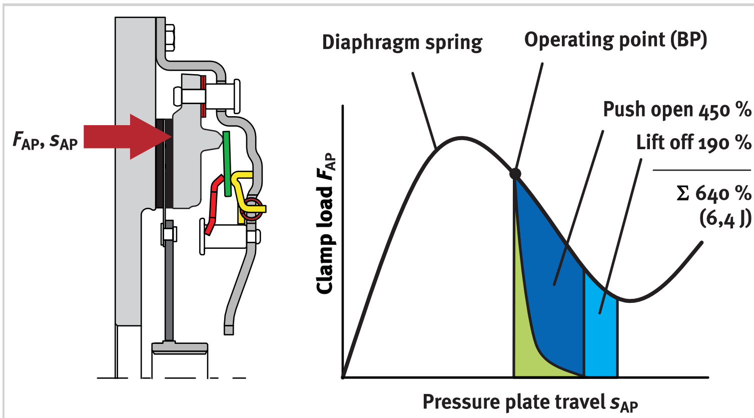

Figure 5 Work required to disengage the normally closed clutch (SAC)

In a conventional clutch, the diaphragm spring curve is made flat to avoid too great an increase in work over its service life. That means an operating work on the pressure plate in the order of 800 % for the new condition, rising to about 1000 % as wear increases.

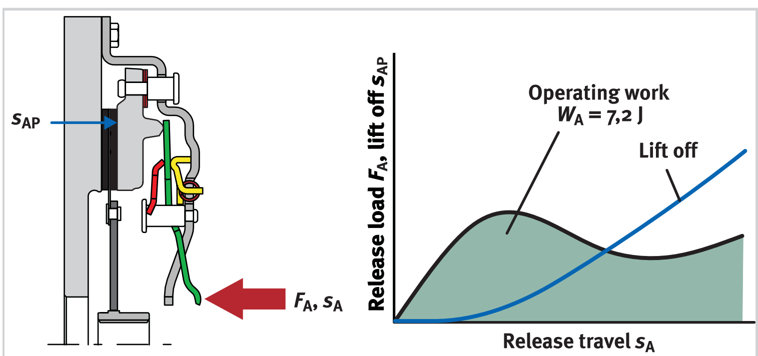

Figure 6 Load and travel on the diaphragm spring fingers

Without the operating point displacement due to facing wear, the diaphragm spring characteristic can be approximated to the curve of the cushion spring characteristic, which reduces the disengagement work required at the pressure plate. This principle was put into practice with the SAC, introduced by LuK in 1994. As a result of the loadcontrolled adjusting mechanism, the operating point remains approximately constant, which can significantly reduce the work required (as compared to a conventional clutch in the new condition) to about 640 % (Figure 5) with no significant increase over the service life. This notably increases the advantage of an SAC over a conventional clutch.

Irrespective of whether a conventional clutch or an SAC is used, levers are required to operate the diaphragm spring. In today's diaphragm spring clutches, these are realized in the form of diaphragm spring fingers (Figure 6). This design is favorable with regard to cost and installation space, but unfavorable with regard to stiffness and the associated travel losses during disengagement. This raises the work required to disengage an SAC to approximately 720 %. An application with 300 Nm thus requires approximately 7,2 joules to disengage the clutch.

There are additional stiffness losses in the release system. The work required from the driver to disengage an SAC finally increases to approximately 800 %, with reference to the energy stored by the cushion spring and required to ensure the torque transfer (Figure 7).

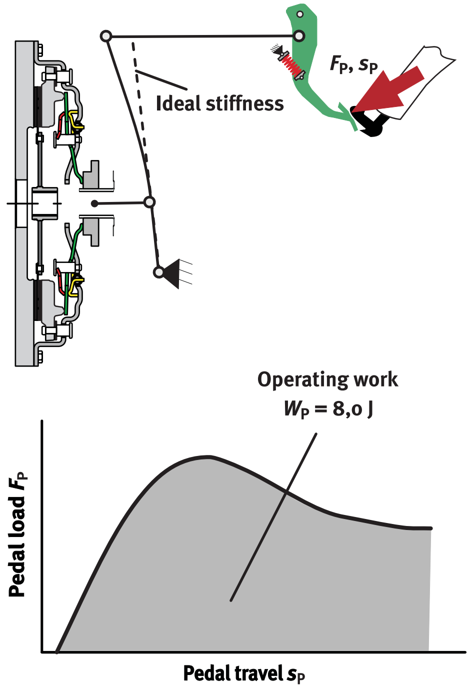

Figure 7 Loads and travels on the clutch pedal

As already shown in Figure 3, this corresponds to 8,0 joules for an application with 300 Nm.

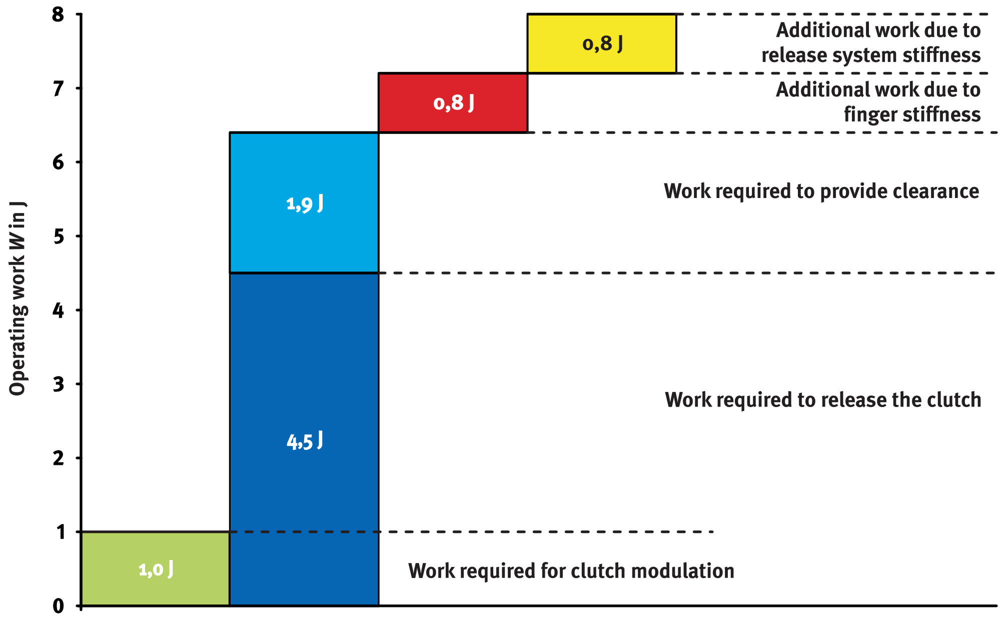

Figure 8 Composition of the operating work in an SAC clutch system

The composition of the work required to disengage the SAC as described above is a typical example of the state of the art. These systems have certainly reached a high standard as a result of the developments in the past several years. Nevertheless, it can be presumed that there still remains significant potential to be exploited, considering the fact that stiffness losses still account for some 20 % of the total operating work (Figure 8).

These losses can be reduced by a thorough optimization of the individual components on the one hand and by a consideration of the system as a whole on the other.

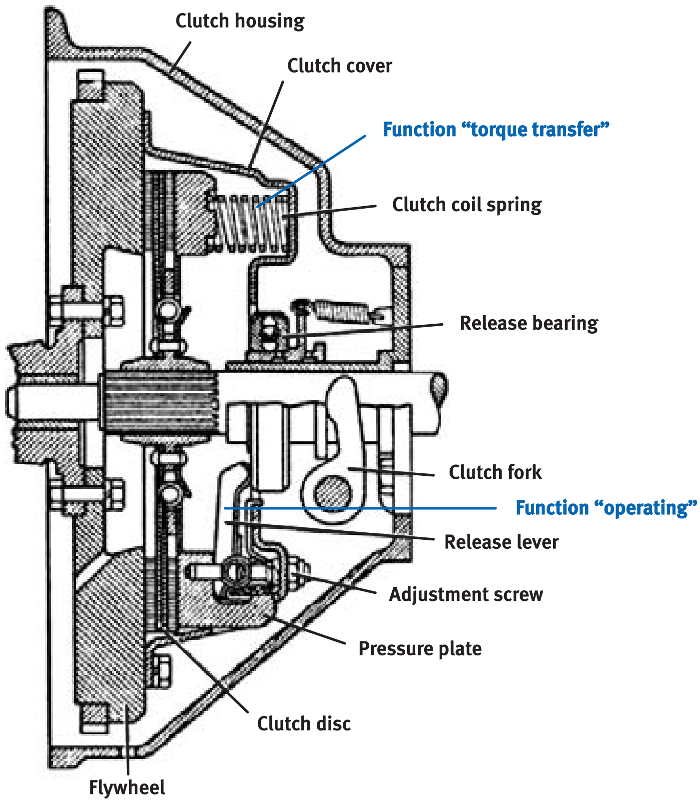

Figure 9 Coil spring clutch

Reducing losses through component optimization

As shown in Figure 8, approximately half of the losses can be traced to the stiffness of the diaphragm spring fingers, through which 10 % of the work required on the clutch pedal is caused. The next obvious step is to study this component on its own with respect to possible optimization potential.

The introduction of the diaphragm spring clutch in 1962 appeared to bring nothing but benefits. Springs could be designed with low space requirements with nearly constant load and high clamp loads.

In the diaphragm spring clutch, the diaphragm spring is used both to generate the clamp load and for operation. The thickness of the diaphragm spring also essentially defines the stiffness of the spring fingers and thus the stiffness of the operation.

That functionality was clearly separated in the coil spring clutch (Figure 9).

The conflicting aims of clamp load and operating stiffness can also be resolved with the diaphragm spring clutch. LuK has studied several possibilities in this regard.

Machined load ring

By “weakening” the load ring, it is possible to create diaphragm springs with high finger stiffness. This makes it possible to increase the thickness of

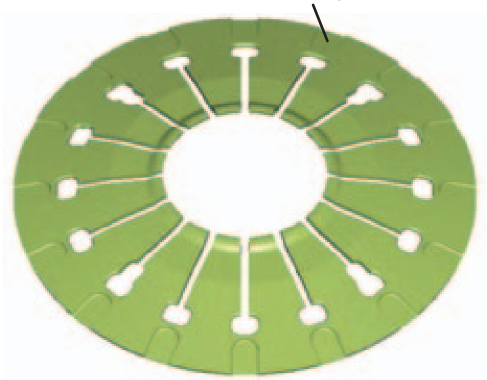

Figure 10 Diaphragm spring with machined load rings

the diaphragm spring by 50 % whilst maintaining a comparable clamp load characteristic (Figure 10).

Reinforced diaphragm spring

Reinforcing elements can be added to increase the bending stiffness of the diaphragm spring. Consequent application of this approach yields an 80 % increase in stiffness (Figure 11).

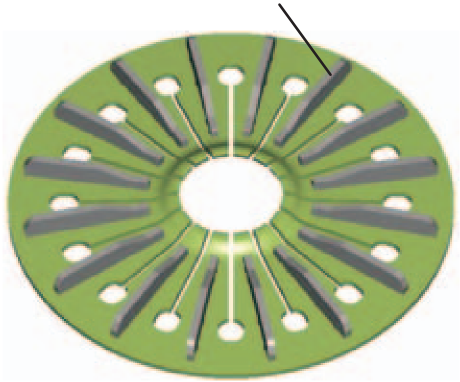

Figure 11 Diaphragm spring with additional elements

Diaphragm spring design

Here, the thickness of the diaphragm spring is optimized with respect to operating stiffness (Figure 12). FE optimization tools are used to define the geometry of the load ring to generate the desired clamp load curve. With this method,

Figure 12 Geometrically optimized diaphragm spring

the thickness of a diaphragm spring can easily be doubled whilst maintaining a comparable clamp load characteristic.

Reducing losses through system optimization

An extensive examination of the entire clutch and operation system can reveal weaknesses, which can then be eliminated through an improved design in future developments.

In addition to the traditional tuning cycle approach, LuK is increasingly using simulation-supported optimization methods. One method used is statistical test plan calculation. The procedure used here is described in detail in the article “Simulation Using the DMFW Example” [1].

Using this method, optimized results can be achieved through many small changes, and general trends can be deduced. One example of this is the

“loss reduction through stiffness reduction”

of a hydraulic release system. This is an apparent contradiction that requires closer consideration before it can be understood.

A common representation of the stiffness of the release system components is the connection between volume expansion and pressure. Figure 13 is a schematic representation of the

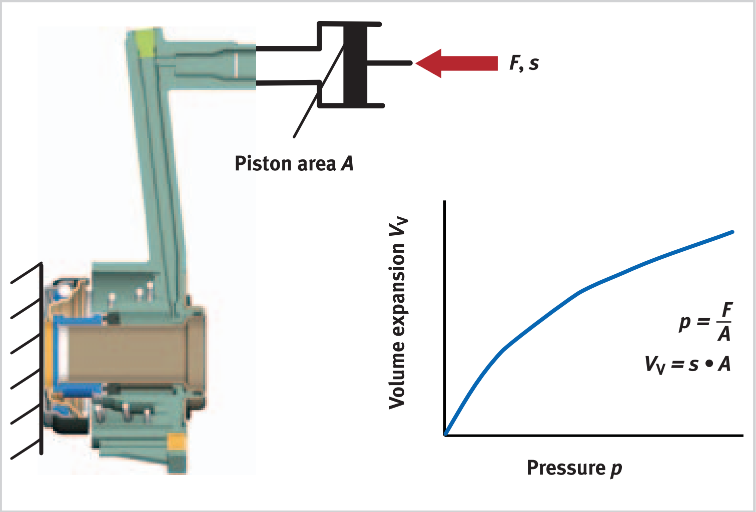

Figure 13 Measurement of the volume expansion of release system components

test setup for determining this characteristic and the typical method of expression for this result, using the example of a concentric slave cylinder (CSC). This procedure is certainly correct for describing the individual components, but is not entirely suitable for the complete system. This is because only the relation between the travel loss on the pedal and the release load of the clutch is important here.

The volume expansion of the master and slave cylinder is significantly influenced, among other things, by the dynamic seal.

In the depressurized state there is free distance between the seal and cylinder wall. This can be calculated from the area marked in red in Figure 14 multiplied by the seal length. When the cylinder is subjected to pressure, this free volume is first taken up by the seal. Thus the oil displaced represents a loss. As the seal itself can be considered virtually incompressible, a proportional relationship between the volume expansion and the seal length can be presumed.

It follows, therefore, that an increase in the piston area will lead to an increase in the volume expansion. Thus the stiffness of the master and slave cylinders is first decreased in the usual representation and the volume expansion raised (Figure 15). It would be a mistake to conclude that this is also accompanied by increased loss for the system as a whole, as is shown below.

The initial parameter for the overall system is not the pressure, but the release load of the clutch. When the surface area is increased, the system pressure drops as a result. The advantage remains unclear as long as we only consider the volume expansion. If, on the other hand, we do not consider the volume expansion as a function of the system pressure, but the travel loss as a function of the release load, we find a significant advantage (Figure 16).

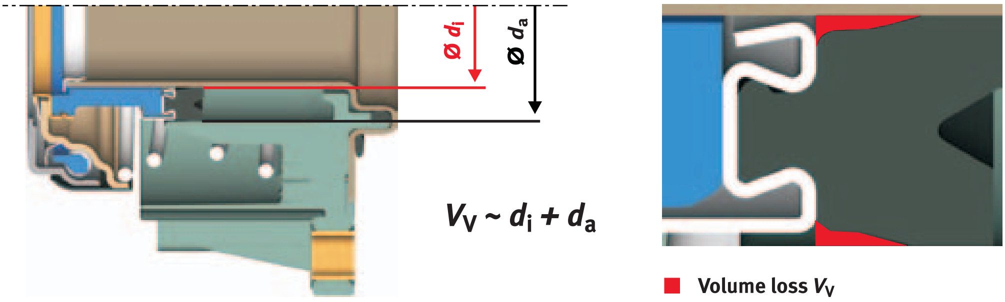

Figure 14 Volume expansion of a concentric slave cylinder (CSC) as an example

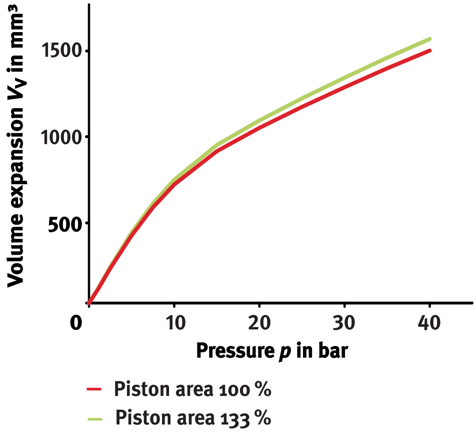

Figure 15 Increased volume expansion with increased piston area

This finding has already been verified using prototypes. Based on a system that is usual for today (master cylinder area = 285 mm², slave cylinder area = 775 mm²) the cylinder areas were increased by 30 % as an example (master cylinder area = 380 mm², slave cylinder area = 1025 mm²). As a result of this step, the travel loss on the clutch pedal was reduced by 30 % from 25 mm to 17 mm with a release load of 2000 N (Figure 16). A remarkable improvement based on an overall consideration of clutch and operation.

This highlights a decade-old error – one made even by the experts.

The usual surface areas of the master and slave cylinders today arose largely from the standardized size of the brake cylinder. Those, however, must now be called into question and possibly redefined in light of this new finding.

Measures for reducing operating load

The first and also obvious approach was described in the previous section – optimizing the existing system. From what is known today, this potential can be used to reduce the operating loads by about 10 % to 15 %, so that engine torques up to 350 Nm can be covered with acceptable pedal travels and loads. Over and above, further measures are necessary in order to be able to attain operating loads lower than 110 N. There are a number of options for doing this, which are described in the following sections:

- Work redistribution
- Energy accumulators
- Multi-plate clutch
- External energy (active support)
- Clutch-by-wire

Work redistribution

In work redistribution, the total work remains constant and is merely distributed more favorably over the pedal travel. The principle is simple. The work should be increased where the pedal load is currently low, so that it can be reduced where the pedal load is currently high (Figure 17).

Reduced-load clutches have in principle a strong "drop-off" in the release load characteristic because of the approximation of the diaphragm spring to the cushion spring curve. A

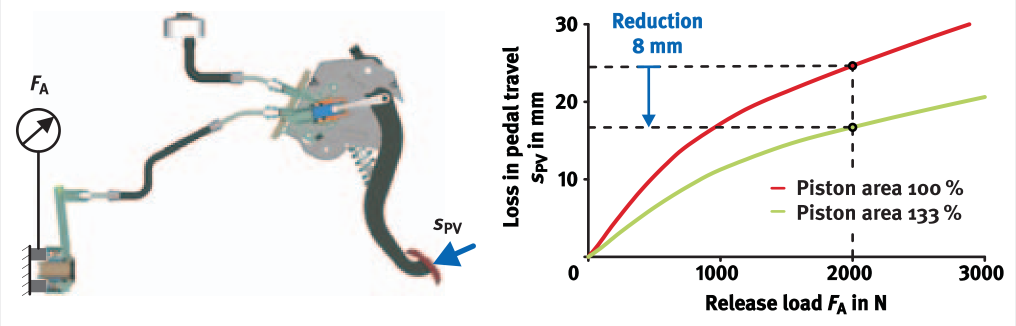

Figure 16 Relationship between pedal travel loss and release load

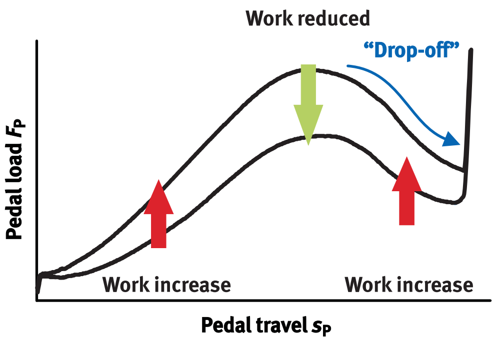

Figure 17 Work redistribution in clutch operation

dual effect can thus be achieved by this redistribution: the realization of a harmonic load characteristic and the reduction of the maximum pedal load.

The technical solution is a variable ratio for clutch operation. At low pedal loads the ratio is reduced and at high pedal loads it is raised. A clutch system with clutch, hydraulics and pedal provides three possibilities for variation. Installation space and tolerance sensitivity argue against a design implementation inside the clutch. At LuK, therefore, the "variable hydraulic ratio" and "variable pedal ratio" concepts are being pursued.

Variable hydraulic ratio

The hydraulic ratio is calculated from the slave to master cylinder area ratio. This means that the variability can be achieved by changing one of the two areas as a function of the piston stroke. Because of the tolerance and wear situation of the clutch system, implementation in the slave cylinder is not feasible. This is because the slave cylinder piston position does correspond to any specific clutch position. For this reason, LuK is developing a master cylinder with a variable piston surface (Figure 18). A design with a moving primary seal and variable cylinder diameter is preferable, since the greatest pressure therefore occurs at the smallest seal gap. The risk of gap extrusion is therefore minimized.

The advantages of the variable master cylinder include a relatively simple design with no additional components and neutral installation space requirements. Since the variability is achieved through the variable sealing gap, the design is limited. The spread (difference between the largest and smallest ratio) currently being tested is 14 %. Since the total work must remain constant, there is potential for a load reduction of about 7 %. Combined with the optimization measures described above, as well as a modified over-center spring, this solution can cover engine torques up to 400 Nm with pedal loads less than 110 N.

Variable pedal ratio

For engine torques greater than 400 Nm, a variable pedal ratio (VPR) is a very promising approach. The principle is the same as that in the previous section, but more work can be redistributed with a mechanical design with a possible spread of up to 60 %. The potential for load reduction is thus approximately 30 %, because of which, applications up to 500 Nm can be covered.

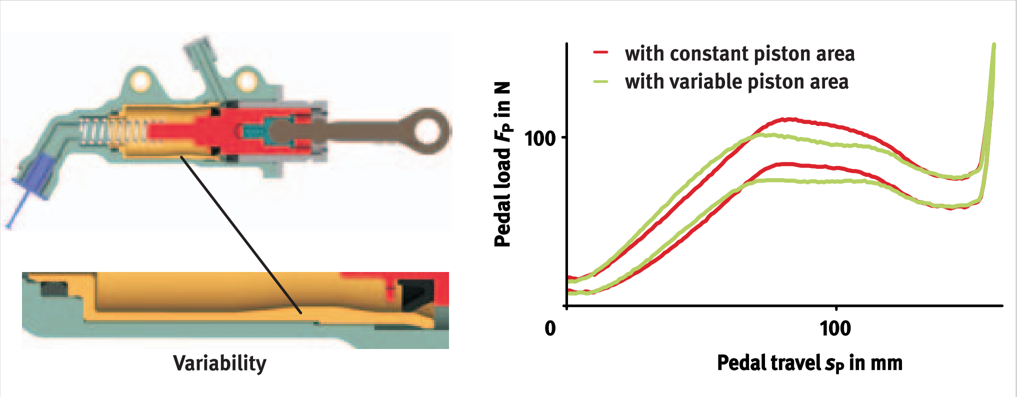

Figure 18 Master cylinder with variable cross section

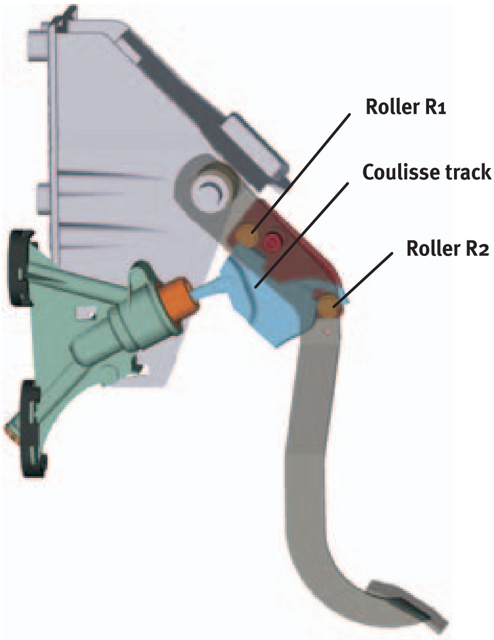

Figure 19 Clutch pedal with variable ratio

Figure 19 shows a production drawing for this solution. The system consists of two rollers mounted on the pedal and one piston rod permanently connected to a coulisse track. When the clutch pedal is operated, the rollers follow the coulisse track, with each of the two rollers able to support a load that is perpendicular to the track (Figure 20). The function lines f_{R1} and f_{R2} of these two loads and the function line f_K of the piston load intersect in the load center of the system. The parallel distance $l_H(s_P)$ from function line f_K to the pedal rotational axis determines the lever arm and thus the ratio of the clutch pedal as a function of pedal travel s_P.

Through the design of the coulisse, any ratio curve is possible as a function of pedal travel. There are, however, design limitations in the form of the piston cross forces, surface pressure and system stiffness.

Figure 21 shows a measurement of a functional example. The original maximum pedal load of 200 N and drop-off of 100 N could be reduced by this system to 160 N and 40 N respectively.

A favorable side effect of the flatter pedal load curve is a likewise flatter clutch torque characteristic curve between the engagement point and approximately 100 Nm. This is because a greater ratio acts in this range like one of a comparable non-variable system. The result is improved clutch modulation capability in the lower torque range (traffic jam or maneuvering operations).

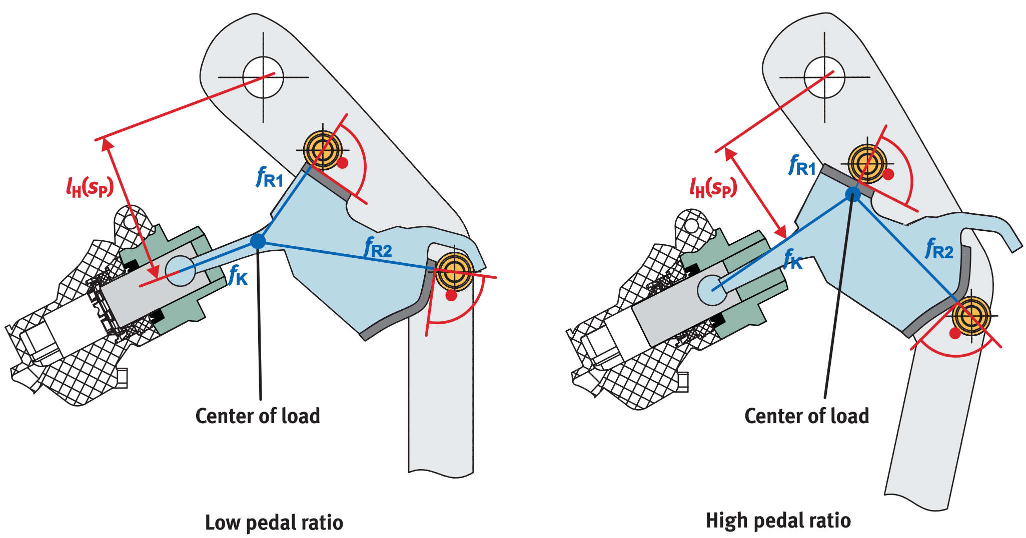

Figure 20 Relationship between coulisse track and pedal ratio

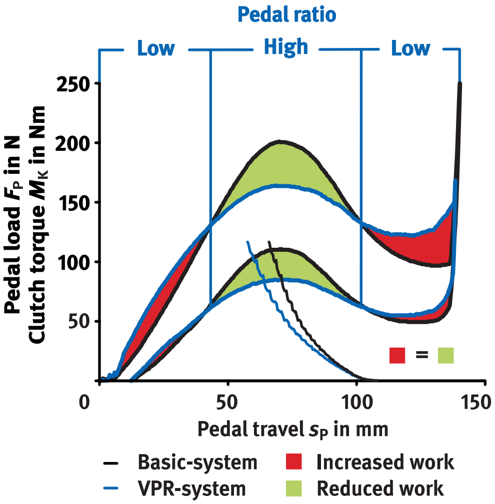

Figure 21 Pedal load measurement with and without variable pedal ratio

The complexity of the variable pedal ratio system is higher than that of the variable hydraulic ratio system. Additional components are unavoidable and the installation space requirements at the pedal interface increased. In order to keep these requirements to a minimum, the version developed by LuK is largely integrated in the master cylinder. In the design shown in Figure 19, only a second fastening pin is required on the pedal side.

Energy accumulator

Energy accumulators are charged at low pedal loads and release their energy in areas of high pedal loads. In today's systems, this principle is already frequently used on the pedal in the form of an over-center spring. The advantage of over-center springs is that they are already charged and pre-stressed before the system is actuated. This not only redistributes the operating work required from the driver, but also reduces it. Over-center springs always have a symmetrical characteristic curve with respect to the cross-over point, which means that limitations are already placed on the pedal load correction. Two-stage characteristic curves, especially in combination with the SAC characteristics, are a further development (Figure 22). It has enabled even critical applications to be configured comfortably.

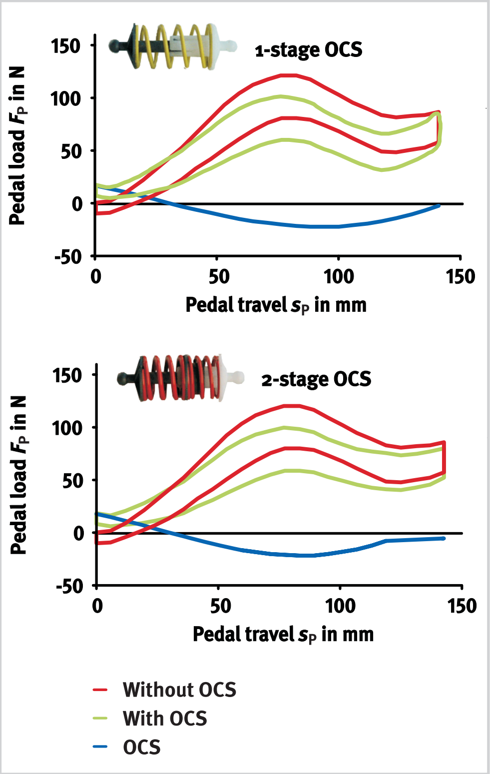

Figure 22 Pedal load curves with one- and two-stage over-center spring

These systems reach their limits with increased engine torque. As mentioned at the start, a pedal load of less than 110 N can hardly be realized with an engine torque of 350 Nm and above.

LuK is therefore studying alternatives. Again here, the clutch system offers three possible installation positions: the clutch, the hydraulic system and the pedal assembly. Integration into the hydraulic system would only be possible with an extensive reconfiguration. LuK is therefore pursuing the following solutions: “Clutch with servo spring” and “Pedal assembly with leaf spring, roller and coulisse.”

Clutch with servo spring

The integration of an independent servo spring in the clutch provides a potential pedal load reduction of up to 20 N. However, all the tolerances of the diaphragm spring finger height and clutch wear lead to significant variation in the level of load and load characteristic.

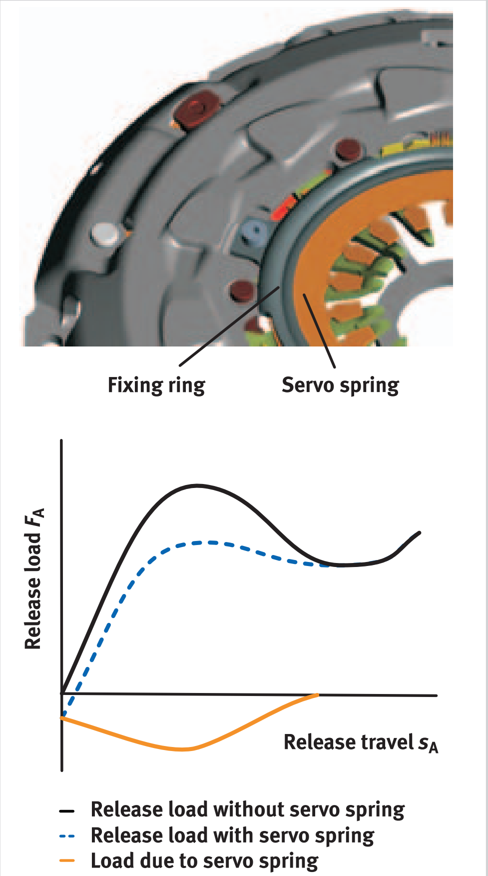

Figure 23 Design and operating principle of the servo spring

Pedal assembly with leaf spring, roller and coulisse

In this system, a leaf spring, one end of which has a fixed connection to the pedal box, serves as an energy accumulator (Figure 24). The free end has a roller bearing that presses on a coulisse fixed to the pedal. The required torque about the pedal rotation point is therefore generated.

This system offers a number of advantages compared to the conventional over-center spring:

- Any assistance characteristic is possible
- Preload independent of assistance load
- Push-pull-push possible

With the over-center springs used today, only one reversal of load direction is possible. Before the cross over point, the over-center spring generates a positive load to return the pedal and after the cross over point a negative load to reduce the pedal load. In the spring system presented here, the load direction can in principle be reversed any number of times (push-pull-push function, Figure 24). So in case of insufficient return load with the pedal in the down position, the load can be increased accordingly.

Measurements with functional samples produced good results with a clear gain in perform-

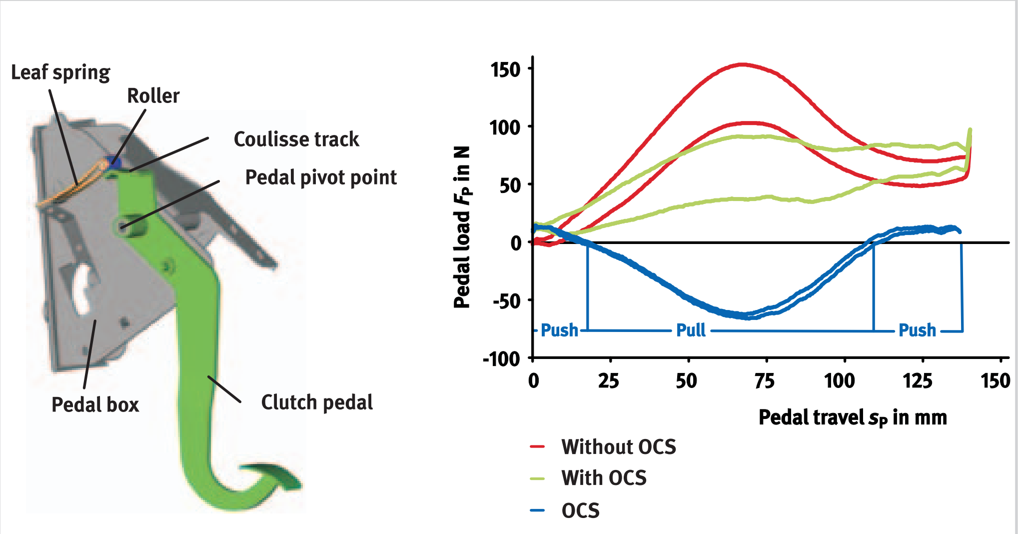

Figure 24 Energy accumulator with leaf spring, roller and coulisse

ance compared to the over-center springs used today. In the example shown in Figure 24, the maximum pedal load is reduced by 65 N whilst achieving the desired return load of 15 N. It is worth noting that the operation is virtually frictionless (low hysteresis increase).

This concept can be used to cover applications up to 500 Nm with a desired pedal load of 110 N. Design limits are set by the permissible stresses in the leaf spring and the surface pressure between the roller bearing and coulisse track.

Multi-plate clutch

By multiplying the number of friction surfaces, the multi-plate clutch is able to reduce the operating load. In a twin-plate clutch, for instance, the operating load can be lowered by 40 % with the same moment of transmission. Some of the benefit is lost in ensuring the separation of the two clutch plates.

In addition to the load benefits, the multi-plate clutch also provides improved thermal capacity and the potential to reduce the diameter of the clutch. About 20 mm more axial installation space is required.

LuK now has several twin-plate clutches in standard production for engine torques above 500 Nm. These clutches all take advantage of the proven SAC technology.

External energy to reduce pedal loads

Passive systems are limited based on their efficiency. LuK currently believes that a pedal load of 110 N can be achieved with engine torques up to 500Nm using the current passenger car sector pedal travels (120 ... 160 mm) and clutch dimensions. For applications above 500 Nm for which a twin-plate clutch cannot be used, there is the option of actively supporting the clutch pedal in a manner similar to power-assisted steering. In order to make the expense of an active system pay off, LuK currently envisages the following application areas:

- Required pedal loads below 110 N for applications between 400 Nm and 500 Nm
- Applications above 500 Nm
- Shortening pedal travel/redefining ergonomics
- Applications outside the automotive sector (e.g., commercial vehicles, tractors)
- Retrofits
- Optional equipment

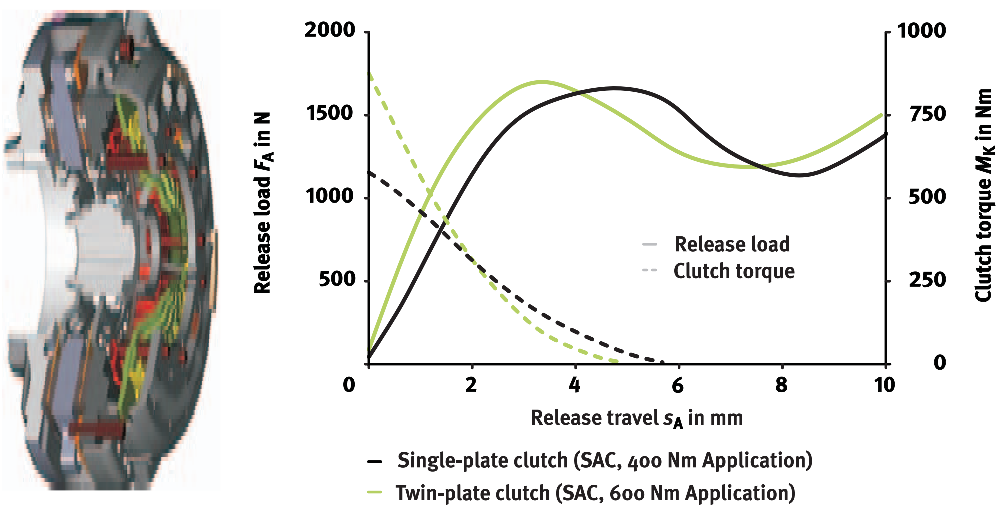

Figure 25 Design and characteristic curves of a twin-plate SAC

LuK has studied a number of possible versions and configurations for active support with the following development goals:

- Self-contained, easily adaptable unit
- Low interface requirements (add-on)
- Function independent of other components (e.g., combustion engine)
- Differentiation from clutch-by-wire
- Maintaining a direct connection between clutch and pedal

These requirements are met in the electro-hydraulic CSA (Clutch Servo Assistance) system developed by LuK. This is a pump unit driven by an electric motor positioned directly between the slave and master cylinder (Figure 26).

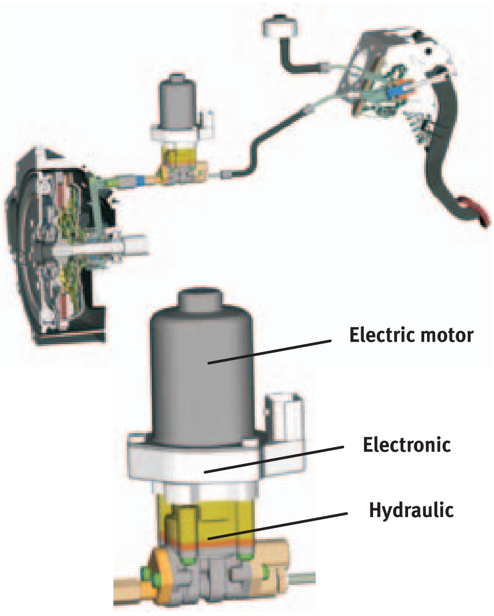

Figure 26 Clutch servo assistance (CSA) – arrangement in the overall system

This unit consists of an electric motor, electronics and hydraulic system. The electronics are for monitoring only and protect the system from overload (temperature, current). They are not required for the actual function of the unit.

The hydraulic system (Figure 27) consists of an internal gear pump, a regulating valve and a safety valve, which ensures clutch operation under all conditions.

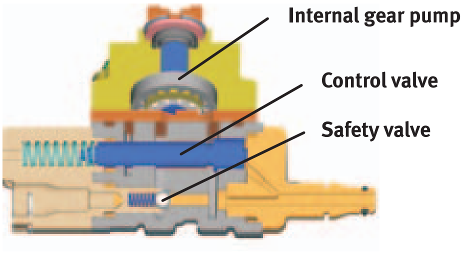

Figure 27 Hydraulic system of the CSA

The system has five possible operating modes:

Sleep	Electronics and electric motor not powered
Stand-by	Electronics powered, electric motor not powered
Pump turning	Regulating valve open
Pump turning	Regulating valve at operating point
Emergency	Safety valve operation

The system is in sleep mode when, for example, the ignition key is out. In this state, the entire system is inactive and without power. The system therefore has no energy requirements and no functionality.

When there is a possible intention to operate, the system is in stand-by mode. This state may be defined by the presence of the ignition key in the ignition. The electronics are now active, but there is still no power to the electric motor.

If the clutch pedal is operated from stand-by mode, the pump is driven by the electric motor. Up to a freely definable pressure threshold p_S in the release system, there is no support, since the regulating valve is pre-loaded by a spring. The control edge is still completely open and the pump cannot build up pressure.

When the pressure threshold p_S is exceeded, pressure-proportional support is generated and the release system is divided into pressure ranges: high pressure (p_{slave}) and low pressure

(p_{master}) (Figures 28 and 29). The proportionality is expressed as the reduction factor k and is determined by the area ratio $A2{:}A1$ of the regulating valve. If, for example, the goal is to bisect the pressure level above the start-up threshold p_S in the master cylinder, then the result is a regulating valve surface ratio of $A2{:}A1$ = 1:2.

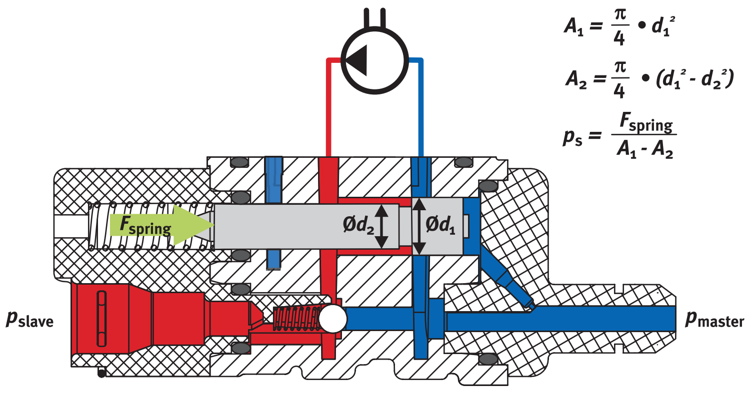

Figure 28 Pump turning, regulating valve at operating point (support)

In summary, the following equation characterizes the layout of the regulating valve:

$$\frac{p_{master} - p_S}{p_{slave} - p_S} = \frac{A_2}{A_1} = k$$

Through the pressure threshold p_S and reduction factor k parameters, the pedal load curve can be accurately corrected to two target values. Figure 29 shows an example of this. The target was to reduce the original 250 N pedal load to 120 N with a subsequent drop in pedal load (drop-off) of 20 N. The target could be achieved with a switch-on threshold of 30 N (corresponds to a pressure threshold p_S of 5 bar) and a reduction factor of 0,4.

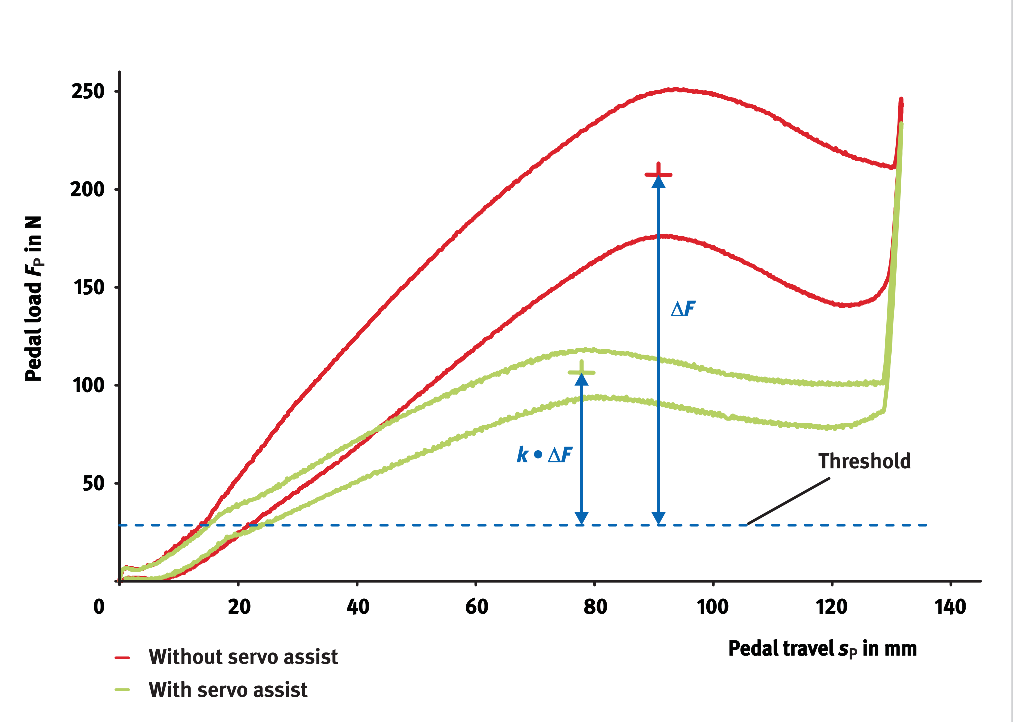

Figure 29 Measurement of the pedal load with and without CSA

A rational alternative to the CSA would be, for example, a pump driven directly via the accessories. The benefit of this would be the elimination of the electric motor and electronics, making it possible to, amongst other things, significantly lower the system costs. The disadvantages include higher interface demands on other systems and the lack of functionality when the engine is not running.

Clutch-by-wire

In modern engines, the gas pedal is no longer mechanically linked to the throttle valve or the injection system. Instead there is just a sensor on the gas pedal that forwards the driver's commands to an actuator via a control device (electric gas pedal). This creates the possibility to adapt the engine characteristic as required to the driving situation. This is now a basic prerequisite for making optimal use of the potential of today's engines and ensuring a comfortable drive.

A clutch-by-wire system promises comparable potential for the clutch (CBW, Figure 30). This is the entry-level version of the family of automated shift transmissions and represents the most complex version for clutch operation for the manual shift transmission. The CBW system eliminates the release system as a fixed connection between the clutch and clutch pedal and uses a pedal travel sensor, a controller and an actuator to operate the clutch (Figure 30). The pedal operating load

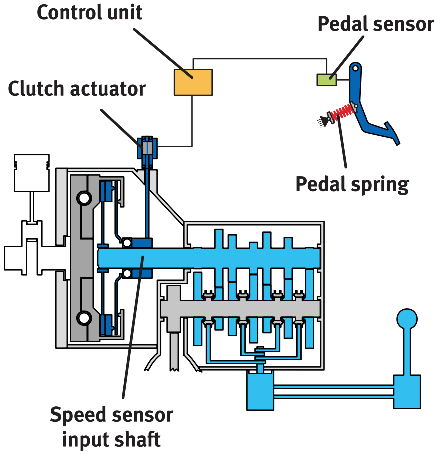

Figure 30 Clutch-by-wire (CBW)

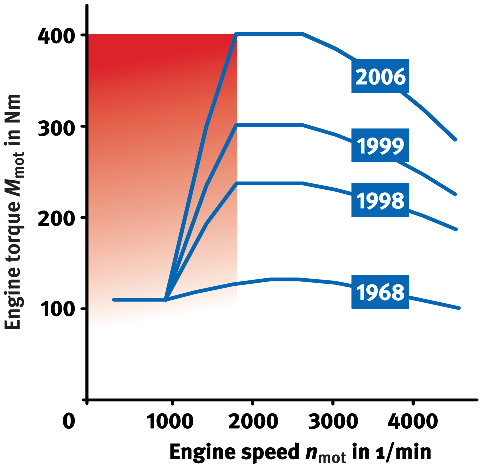

Figure 31 Development of the full load characteristic of a 2.0 liter diesel engine

can be chosen freely using a pedal spring or comparable mechanism. This makes an optimal layout possible regardless of the engine torque.

In order to justify the high expenditure on hardware and software, a CBW system offers numerous possibilities for improving driving comfort. These include, in particular, slip control for vibration isolation between the engine and transmission, which is dealt with in the article “Software for Automated Transmissions” [1].

In a similar manor to how the electric gas pedal adapts the engine characteristics, CBW can be used to adapt the torque characteristic of the clutch to the specific driving situation. An important example in this context is a launch process, which is influenced by the combination of engine torque and clutch torque. Figure 31 shows how the full-load curve of a 2.0 liter diesel engine has changed over the years. In the idle speed range, the engines operate both then and now in induction mode, with the maximum engine torque practically unchanged at low speeds. By contrast, the maximum engine torques, and therefore also the clutch torques, have tripled over the years (see also Figure 1). As a result, a weak engine is combined with a strong clutch at low engine speeds (red area in Figure 31).

This circumstance can be rectified with a CBW. Using information such as the engine and transmission speeds, the clutch torque as a function of the pedal travel can be varied to best suit each and every driving condition.

LuK has already installed CBW prototypes in several different vehicles and tested them. The launch performance has improved remarkably compared to conventional clutch systems as a result of the function just described.

Summary

Despite the large number of technical solutions presented here, the application areas of the different concepts can be clearly defined.

The controlling parameters are the maximum engine torque and the desired maximum pedal load as shown in Figure 32. This should be viewed as a recommendation, which can vary depending on the philosophy of the vehicle manufacturer.

Smaller steps can be achieved through the separate optimization of individual components. The correct technical solutions for the torque increases of the coming years can only be found with the correct dimensioning of the complete system.

The complexity of the solutions for high torque applications may appear high at first. However, if the intension is to offer manual transmissions in this class, there is no alternative but to start with the total system development as soon as possible.

In order to support this, LuK has put the prerequisites in place with a complete switch from a component supplier to a system supplier. This requires close cooperation between the clutch and chassis departments at the automobile manufacturers. Together with the automobile manufacturers, LuK will continue to present innovative solutions, which offer the end consumer optimal operating comfort despite significantly increased engine torques.

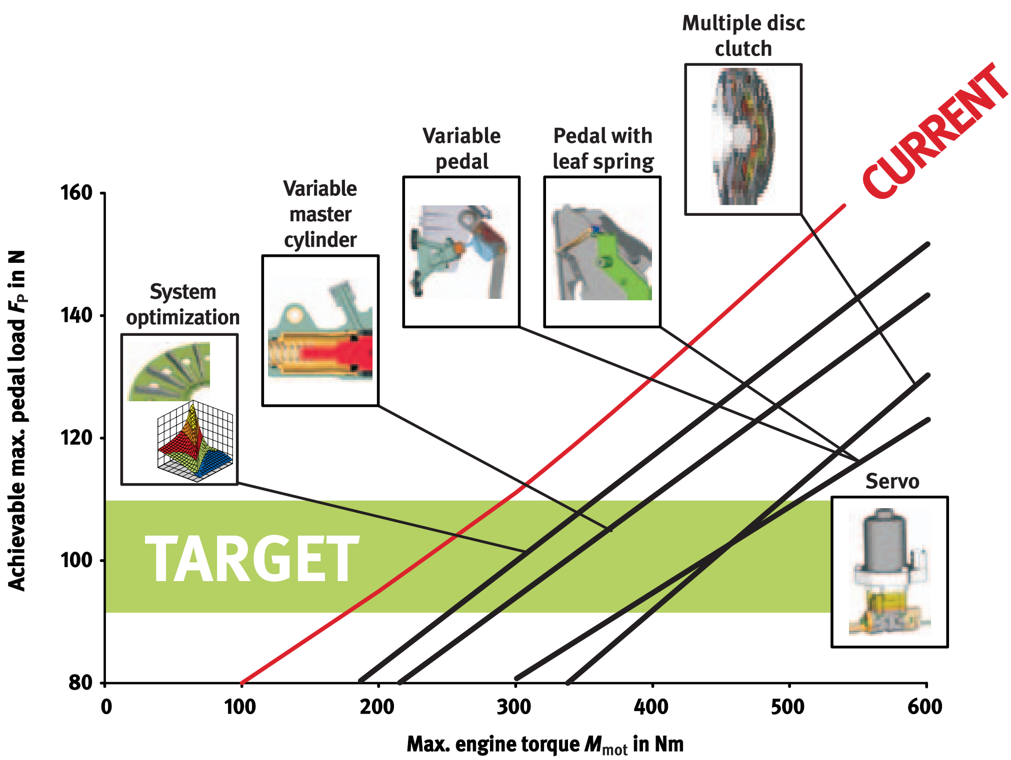

Figure 32 Options and potential for reducing operating load

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