

The self-adjusting clutch - SAC

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Introduction

The high performance engines preferred today need clutches capable of higher transmission torques, which in turn requires increased pedal effort. While there are ways to limit the increase in pedal effort (for instance, by improving the release system), there is a higher demand for clutches with reduced release loads.

This presentation will begin by analyzing whether any load at all is necessary for releasing a clutch. The answer may be surprising.

The presentation continues with a description of actual clutch designs which are the result of theoretical considerations, know-how that promises to produce considerably lower pedal efforts.

Counterbalancing for load reduction

In engineering, counterbalances are frequently introduced in order to reduce operating load. Examples are cable-car systems with two cars moving in opposite direction, counter-equilibrium weights on many machines, etc.; the simplest and easiest mechanism to understand is the scale or the seesaw.

Figure 1 shows a seesaw. In the left section only one seat is occupied, therefore a large load is necessary to move the seesaw.

When a second, equal weight is placed on the seesaw, the resulting equilibrium remains the same for any chosen position (see right section). This is called a neutral equilibrium.

The two weights exert opposing forces (or moments, to be precise) on the seesaw. In other words, the two forces cancel each other out and the sum of the external forces applied to the seesaw is zero. This does not mean that the internal forces are also zero, because each weight obviously exerts a load on the beam of the seesaw. However, if friction is ignored, only an insignificant load is now necessary to set the seesaw in motion. If the two loads are not equal, exactly the difference between their weight is required to start movement.

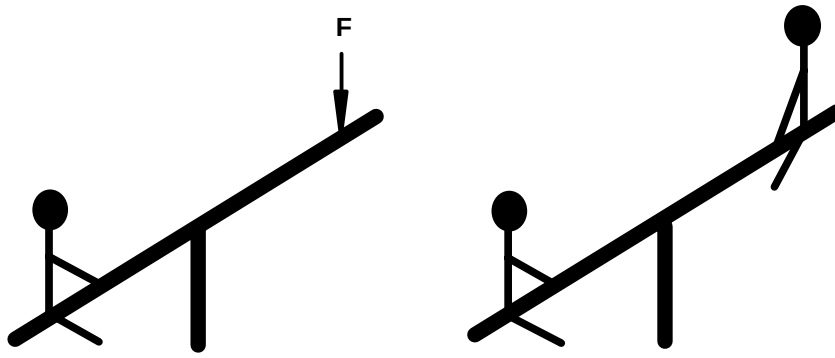


Figure 1: The counterbalance principle

This observation may actually be trivial, but it is decisive for our question, and it will be used in the following chapter to achieve considerable reduction in clutch release load.

Balancing loads in a clutch

Loads counteract each other in a clutch as well and can be used to establish equilibrium.

In the conventional clutch basically two loads are exerted on the pressure plate (Figure 2). The diaphragm spring tries to move the pressure plate in the direction of the engine, and the cushion between the facings forces the plate in the direction of the transmission. Without any external force, equilibrium is established when both loads are equal. This is the case when the clutch is fully engaged, as becomes clear when a diagram of the cushion deflection and diaphragm spring characteristic curves is made.

When both loads are equal, equilibrium is established precisely at the intersection of the two curves (point 1) as shown on the lower right side of Figure 2.

The equilibrium is removed when the clutch is activated. At point 2 the counteracting force of the cushion deflection is reduced considerably. In a conventional clutch a load equilibrium only exists at one condition (point 1), contrary to the seesaw, which remains in equilibrium in every possible position.

When the external force (the release load) is removed, the clutch always returns to the condition at point 1, a stable equilibrium, which only exists in this one position. This is why a relatively high load must be applied for complete disengagement of the clutch, a load that represents the difference between the two characteristic curves shown as shaded in Figure 2.

To be exact, it is this load that would need to be applied directly to the pressure plate in order to activate the clutch. Actually, the clutch is disengaged via the diaphragm spring fingers. These fingers act as levers, reducing the load, but increasing the travel accordingly.

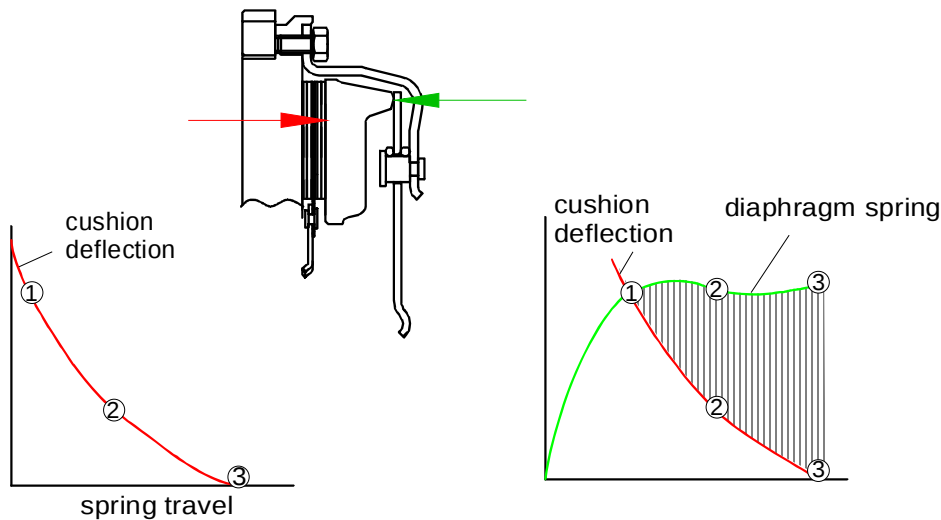


Figure 2: Loads in a conventional clutch

Figure 2 clearly shows how the characteristic curve of the diaphragm spring has to be changed in order to establish equilibrium at more than one position. Its characteristic curve must be adjusted to match the cushion deflection curve (Figure 3).

The cushion deflection and diaphragm spring curves now match over a considerable range. An equilibrium similar to the one on the seesaw is established. The counteracting loads are equal in all positions from 1 to 3, from fully engaged to disengaged. Within the marked area we therefore have neutral equilibrium. Outside of this area, of course, equilibrium no longer exists because the curves diverge from each other.

If the diaphragm spring characteristic curve in Figure 3 seems unusual, it is because we normally assume that the spring load does increase with additional travel; this applies for most springs used in technical applications. There are exceptions however, such as the diaphragm spring, which produce regressive curves over certain ranges.

Depending on the relation between thickness and formed height in free position, the diaphragm spring can deliver various characteristic curves (Figure 4). In extreme cases a snap spring can be designed that will snap over center at a certain spring travel. The only way to return it to its original

position is to apply a counter-force. At any rate, it is easy to produce a curve like the one shown in Figure 3.

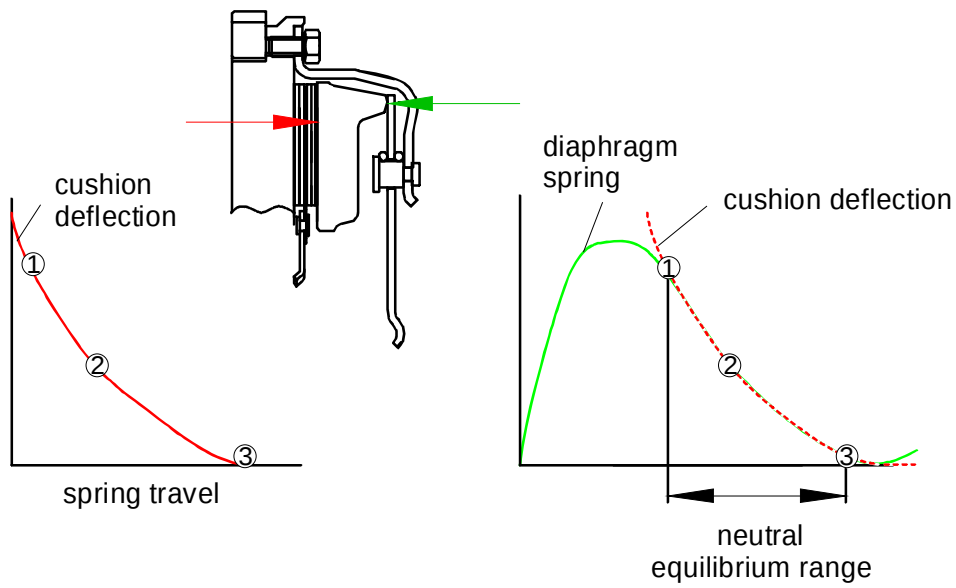


Figure 3: Load equilibrium

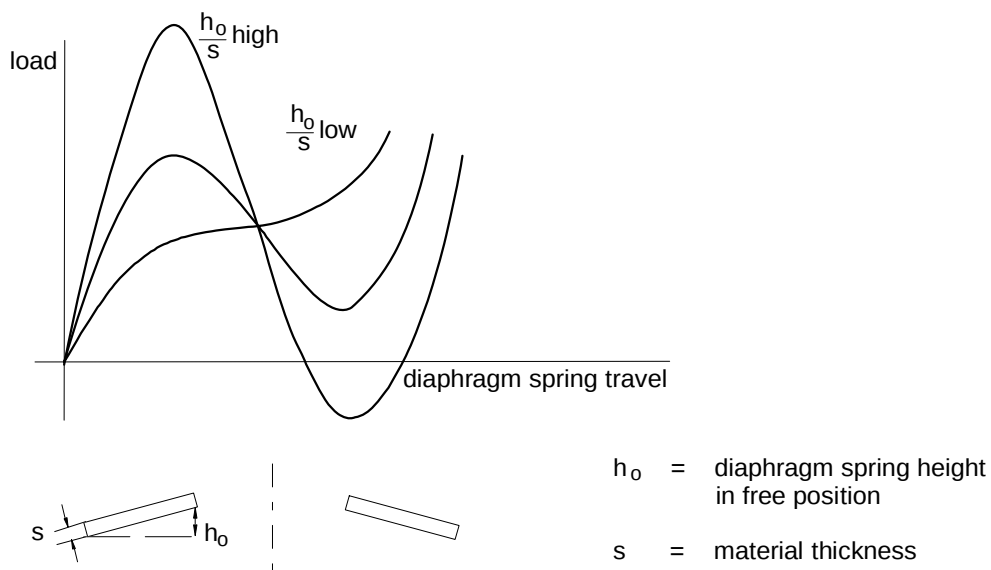


Figure 4: Diaphragm spring characteristic curve with reference to h_0/s

Figure 5 again shows the characteristic curves. In the range of neutral equilibrium (points 1 and 3), the clutch can be shifted without applying virtually any load. However, a clamp load (the diaphragm spring load) does exist between the friction surfaces at position 1. This load is required for torque transfer. Only at position 3 does this internal load decrease to zero.

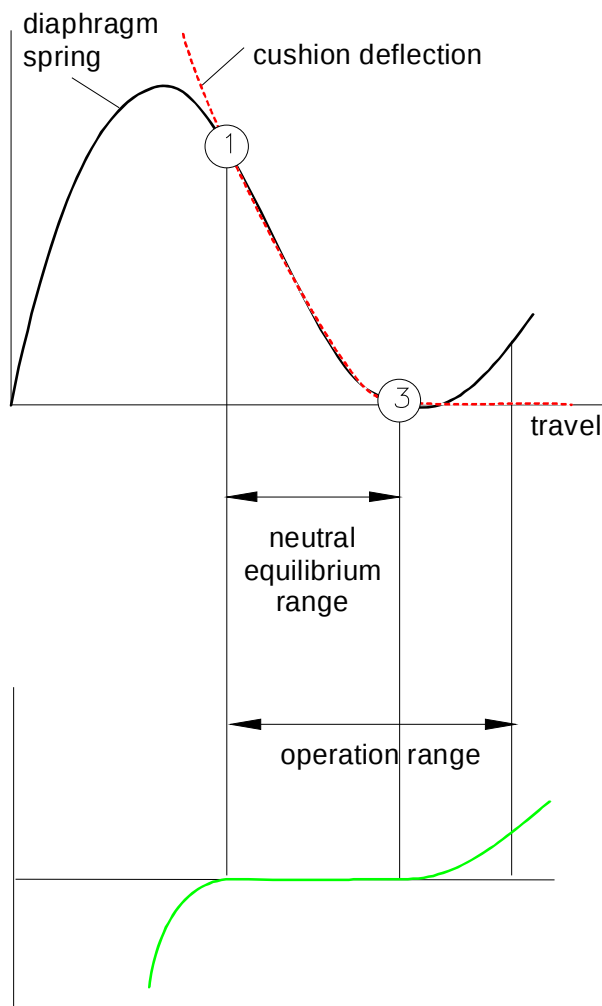


Figure 5: Equilibrium between diaphragm spring and cushion

The release bearing must travel somewhat further for complete clutch disengagement. One says, that the clutch requires clearance in order to prevent drag torque with slightly uneven clutch discs. Figure 5 shows that the diaphragm spring curve leaves the area of neutral equilibrium at this point because it rises steeply, while the cushion deflection has reached

zero and runs along the X-axis. One sees that force is required to provide clearance in the clutch.

Establishing an exact equilibrium would require absolutely matching curves for the cushion deflection and the diaphragm spring. Manufacturing tolerances make this impossible. Figure 6 shows the tolerance deviations that occur during production.

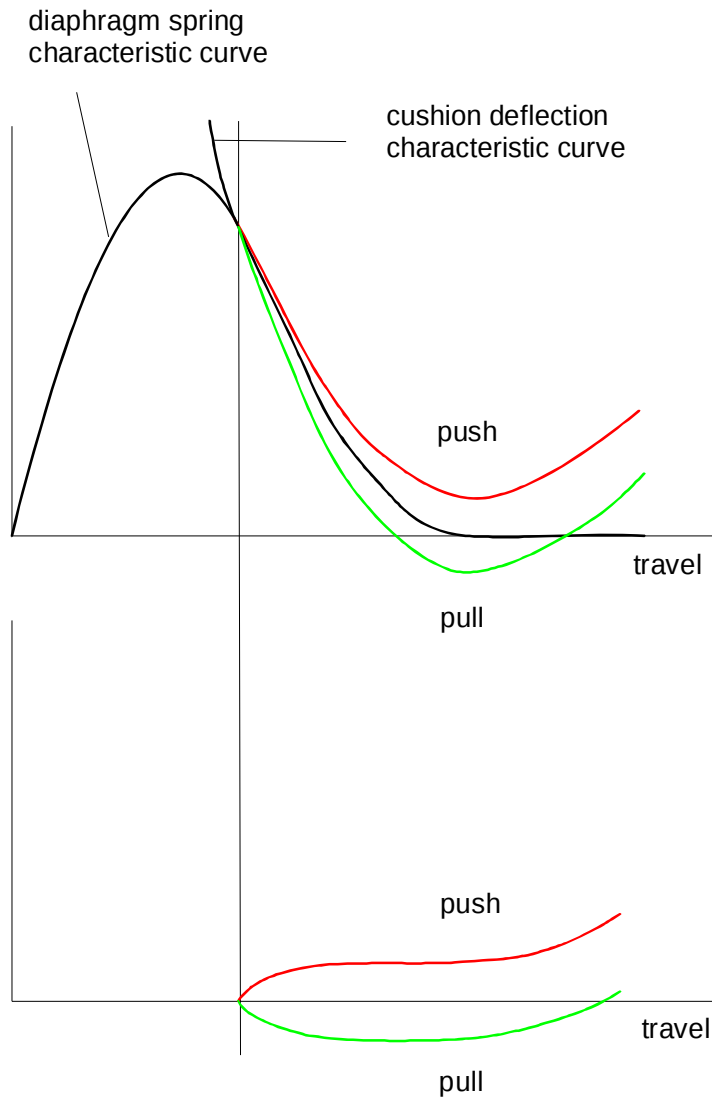


Figure 6: Disturbance of the equilibrium due to deviation from the ideal diaphragm spring characteristic curve

It is assumed that, depending on the tolerances, the diaphragm spring minimum may deviate somewhat from a zero load. If the diaphragm spring curve lies somewhat above the minimum range of the cushion curve, a load equal to the diaphragm spring load minus the cushion load is needed to operate or disengage the clutch.

But if the diaphragm spring minimum lies below the zero-line, the following surprising characteristic can be observed:

The difference between diaphragm spring and cushion loads becomes negative. The clutch then responds quite differently from the conventional design. If no external load is applied, the clutch is totally disengaged and can only be reengaged with a negative, or counteracting force. To put it simply: in order to close the clutch at this point, the diaphragm spring finger must be pulled.

This totally different, tolerance-related response is unacceptable for foot-operated clutches.

A sufficient distance from the ideal equilibrium must be maintained in order to prevent this effect.

Figure 7 shows an estimate of how large the deviation from the ideal equilibrium must be. A load tolerance for the diaphragm spring and the cushion deflection are assumed. The curves on the right show the effect of the tolerances on the release load.

The upper section of Figure 7 shows a conventional clutch, which yields a high release load with little relative variation.

A compromise between release load magnitude and variation is the partial equilibrium (center graph) at which the diaphragm spring minimum is reduced to only a third of the maximum diaphragm spring load.

Full equilibrium (lower graph) shows the effect discussed in Figure 6.

Evaluation of Figure 7 leads to the following conclusion:

The more exact the diaphragm spring and cushion deflection characteristic can be produced, the lower and more uniform the release load can be established.

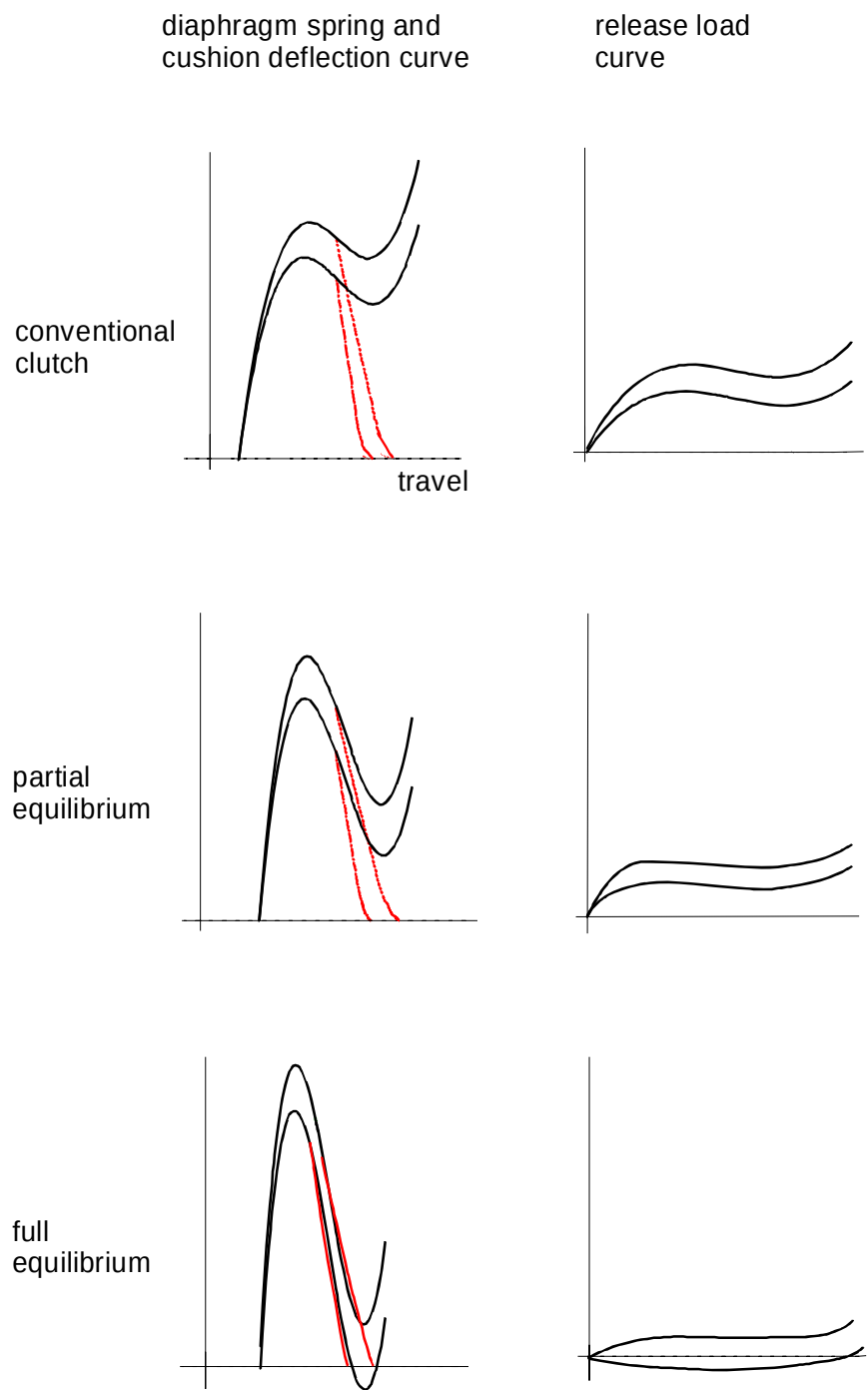


Figure 7: Influence of diaphragm spring tolerance on release load curve

Shifting equilibrium during wear

Until now it has been assumed that the intersection point of the two characteristic curves does not change. This would require absolutely wear-resistant facings because each thickness change in the clutch disc leads to a shift in the cushion deflection curve relative to the diaphragm spring curve (Figure 8).

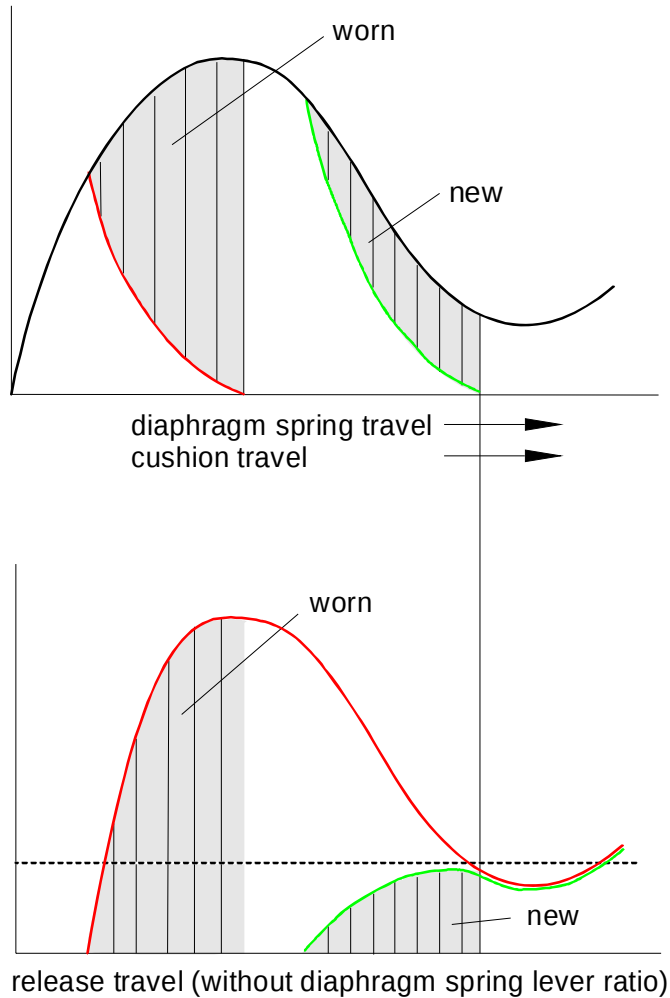


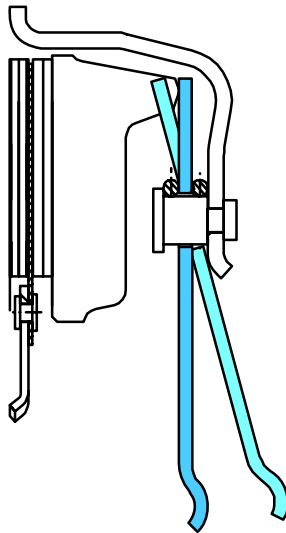
Figure 8: Influence of a shift in the operating point

The characteristic curve shown here by the red line develops after the facings have worn. Because the difference between the diaphragm spring and the cushion is now large, the release load increases accordingly. The advantage of the low release load would only exist in new condition.

To prevent this, the intersection of the two characteristic curves must return to the original location by adjusting for wear. There must be a compensation for the reduction in facing thickness.

Before proceeding, the amount of facing wear, if any, must be determined.

Basically several options are possible (see Figure 9):



Facing wear changes the following

- release load
- disc thickness
- diaphragm spring position

Suitable sensors:

- load sensor
- travel sensor

Figure 9: Changes during wear

Considerable increase in release load due to the thinning of the facings is the first indication of wear. This condition requires a load sensor, which can be included in the design of push-type clutches in a simple and elegant manner.

Facing wear also causes changes in disc thickness and therefore in the position of the pressure plate relative to the flywheel in engaged condition; a travel sensor can indicate this change in location.

LuK has also developed the last option, which would be advantageous in a pull-type clutch. Because the release loads of the SAC-clutch, as described in this paper, can be considerably reduced in a push-type application, the pull-type clutch offers no real advantage.

LuK has compared all the options and decided on the simplest sensor design, which is the load sensor. The following discussion is limited to designs that feature the load sensor and push-type clutch.

Load sensor

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

Figure 10 shows a schematic representation of these factors. As opposed to the conventional clutch, the (main) diaphragm spring is supported by a so-called sensor diaphragm spring instead of being riveted to the cover.

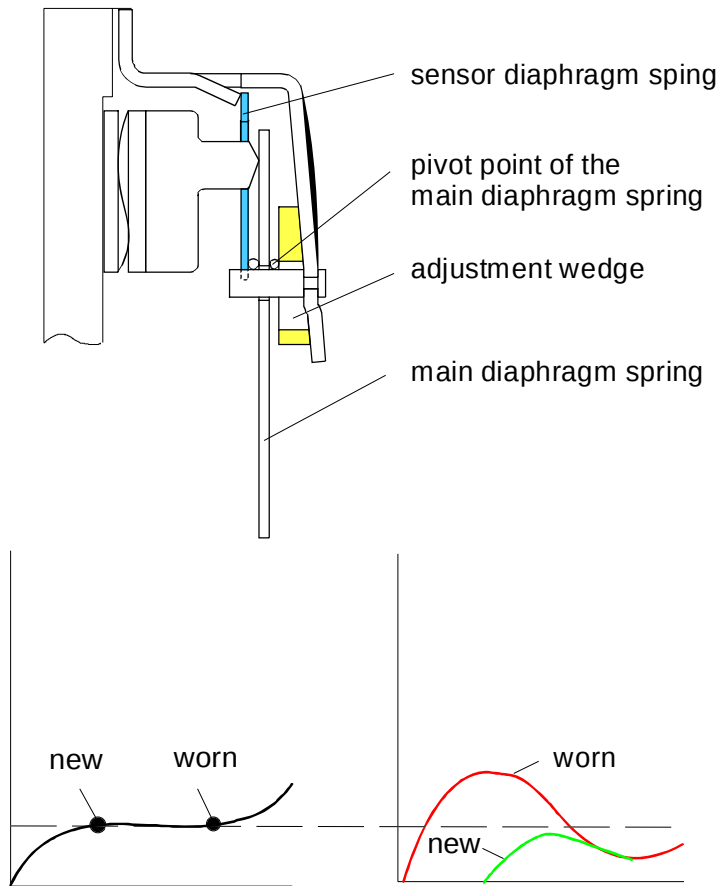


Figure 10: Principle of the self-adjusting clutch

In contrast to the strongly regressive main diaphragm spring, the sensor diaphragm spring provides a sufficiently wide range of almost constant load.

The horizontal range of the sensor diaphragm spring is designed to be slightly higher than the targeted release load. The pivot point of the main diaphragm spring remains stationary as long as the release load is smaller

than the clamp load of the sensor spring. When facing wear increases the release load (Figure 8), the opposing load of the sensor spring is overcome and the pivot point moves towards the flywheel to a position where the release load again falls below the sensor load. Graphically, this means that the intersection point between the two curves has returned to its original location. When the sensor spring deflects, a gap develops between pivot point and cover, which can be compensated for by introducing a wedge-shaped component.

Design of a wear-adjusting clutch with a load sensor

The load sensor with the thickness adjustment wedge can be realized in a simple and elegant manner. Figure 11 shows such a design. In comparison to the conventional clutch, the only additional parts required by this design are a sensor diaphragm spring (red) and a ramp ring (yellow).

The sensor diaphragm spring is suspended in the cover. Its inside fingers position the main diaphragm spring. Because of centrifugal forces, the wedges that provide the actual adjustment are positioned circularly instead of radially. A plastic ring with twelve ramps moves on opposing ramps in the cover. The plastic ring (adjustment or ramp ring) is circularly preloaded with three small coil springs which force the ring to fill the gap between the diaphragm spring and the cover when the sensor spring moves.

Figure 12 shows the release load curves for a conventional clutch in new and worn facing condition. In contrast, compare the lower release load of the SAC, which has a characteristic curve that virtually never changes over its service life.

An additional advantage is the higher wear capacity, which no longer depends on the length of the diaphragm spring curve (as in conventional clutches), but rather on the ramp height, which can easily be increased to 4 mm for small and up to 10 mm for very large clutches. This represents a decisive step towards the development of clutches with high durability.

Until now only the interaction of the characteristic curves for the main diaphragm spring, the cushion deflection and the sensor diaphragm spring has been mentioned. Actually, other resilience or deflection values come into play, which are assigned to the primary characteristic curves during the design process. One refers to operating or effective characteristic curves, meaning that the influence of other elastic elements in the system has been considered (Figure 13).

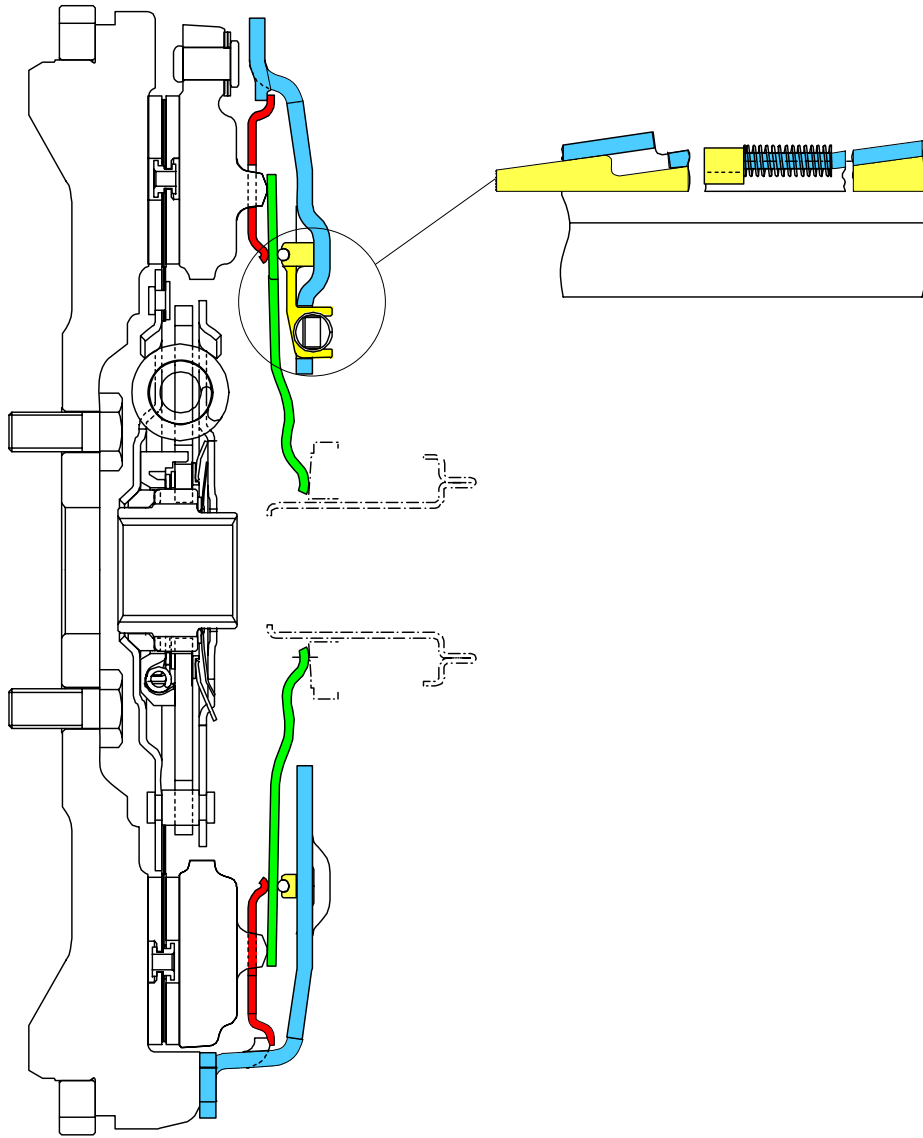


Figure 11: Self-adjusting clutch (SAC)

For example, the effective characteristic curve of the diaphragm spring includes the leaf spring characteristic, which transfers the torque from the pressure plate to the cover and has a lift function. If the sensor diaphragm spring does not position the main diaphragm spring exactly in the support position, the sensor spring curve must also be considered.

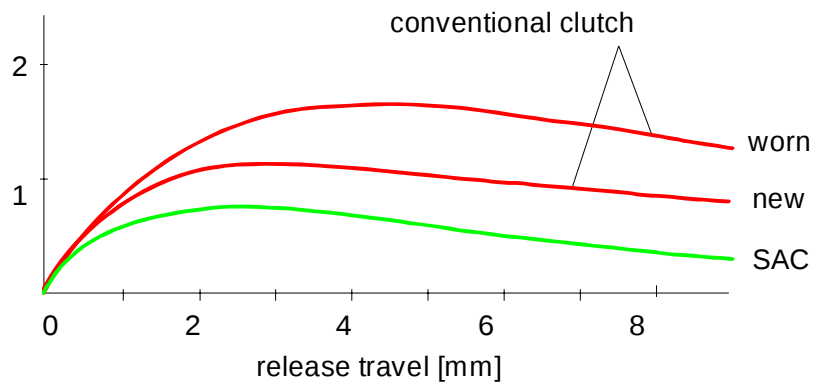


Figure 12: Comparison of SAC release loads with those of a conventional clutch

influence from	effective characteristic curves of		
	diaphragm spring	cushion deflection	sensor spring
diaphragm spring	++		
cushion deflection		++	+
sensor spring	+		++
leaf spring	+		
cover deflection		+	
adjustment spring			+

Figure 13: Effective characteristic curves

The cover deflection characteristic must be added to the effective characteristic curve of the cushion deflection because its effect is similar.

Also, a limited axial deflection of the cover is rather useful in the SAC because it is constant over the life span of the clutch.

The effective characteristic curve of the sensor diaphragm spring must include the effect of the leaf springs and the adjustment springs.

Once all these effective characteristic curves are accounted for, the clutch equilibria can be modeled.

Tolerance considerations

A complete description of tolerance requirements is impossible within this framework, but an attempt will be made to highlight at least those components and their tolerances that must be evaluated differently in the SAC than in conventional clutch design.

Figure 14 shows features that must be exactly controlled and others which may be treated more loosely:

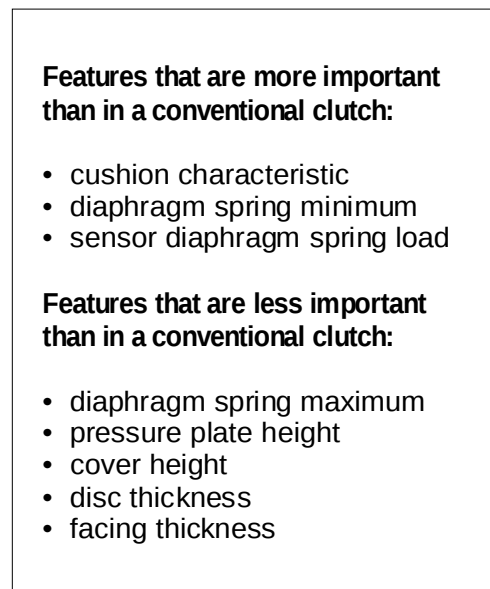


Figure 14: Features for wear-adjusting clutches

In the SAC-design it is very important that the cushion deflection remain constant over the service life of the clutch. In addition, the height of the diaphragm spring minimum is crucial, while the diaphragm spring maximum does not have any significance and may be more liberally tolerated. The sensor diaphragm spring load is also important.

In contrast, thicknesses of pressure plate, disc or facings are now virtually unimportant, as long as the tolerances are tight enough to avoid using up the adjustment capabilities.

The SAC requires that the designer reevaluate all the tolerances.

Cushion deflection requirements

Because cushion deflection plays an essential part in maintaining equilibrium in the SAC-clutch, it cannot be permitted to change significantly over the life span of the clutch. As Figure 15 demonstrates, this is not guaranteed with current cushion deflections. The cushion deflection seems to have stabilized after some run-in time.

After wear, the original segment deflection characteristic is usually restored when the facings are removed. What has happened is not that the segments have set, but have imbedded themselves in the facing because they do not press the facing evenly against the companion friction surface. The segments support the facing in certain locations and naturally the most wear occurs there. Adjacent areas remain thicker and become solid earlier when compressed, which results in a shorter cushion deflection.

Because the SAC-design does not tolerate such changes, the segment has to be modified for a more even unit pressure on the facing friction surface (Figure 16).

This is accomplished by increasing the number of waves (triple waving) with shorter peak (brown)-valley (green) distances, or with a "feathered" segment design featuring directly alternating waves. This provides sufficiently even unit pressure over the life span of the segment, even under heavy use.

In extreme cases, the release load has to be reduced to unusually low levels which requires an absolutely stable segment characteristic curve. This problem has been successfully solved by bonding the facings to steel carriers.

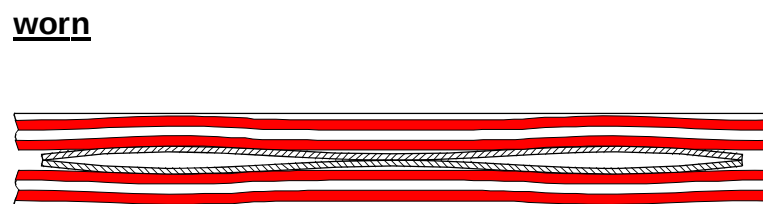
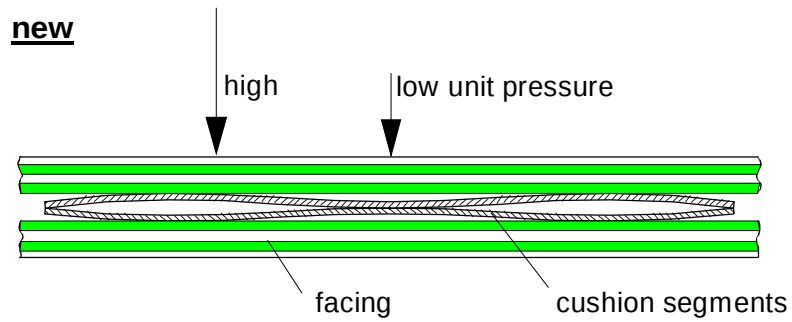
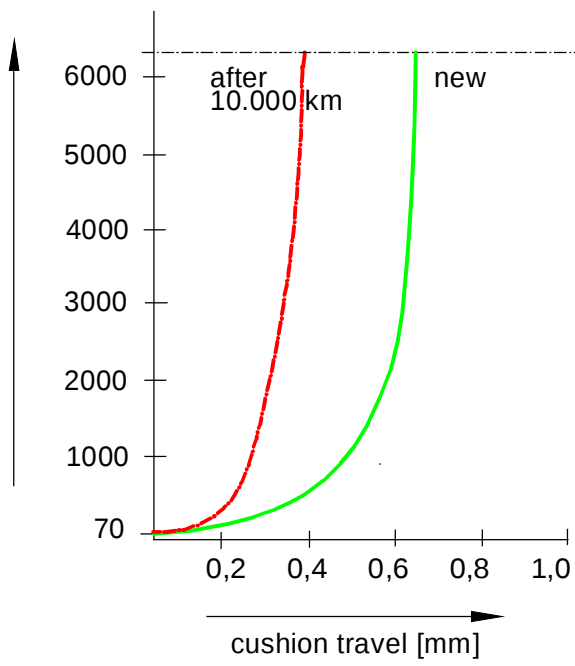
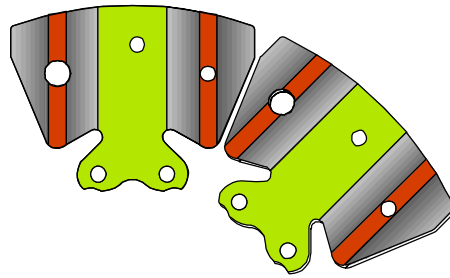
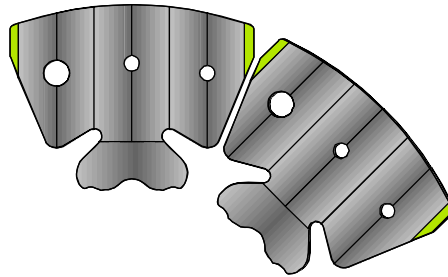


Figure 15: Changes in cushion deflection over the life span of the clutch

**conventional
segment**



triple waving



**feathered
segment**

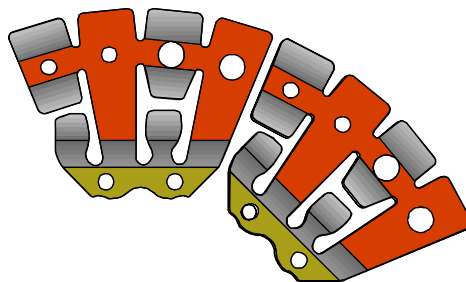


Figure 16: Segment variations

Release system requirements

In the SAC-clutch the adjustment ring readjusts itself each time the release load exceeds the sensor load. Normally this only happens when facing wear has occurred and the mechanism is actually supposed to provide a correction.

If the clutch is overstroked , which means the permissible release travel has been exceeded by 30 %, the release load (see Figure 8) increases towards the end of the release travel and causes an unintended adjustment even

without wear on the facing. This in turn causes the cushion deflection characteristic curve (Figure 8) to move too far to the right and the intersection point of the two curves, representing the clutch clamp load, to be set too low. Then the clutch can slip.

In some hydraulic release systems, quick pumping of the clutch pedal can actually provoke considerable overstroking. If this cannot be prevented reliably, a mechanical stop (as shown in Figure 17) must be included in the cover during the forming process.

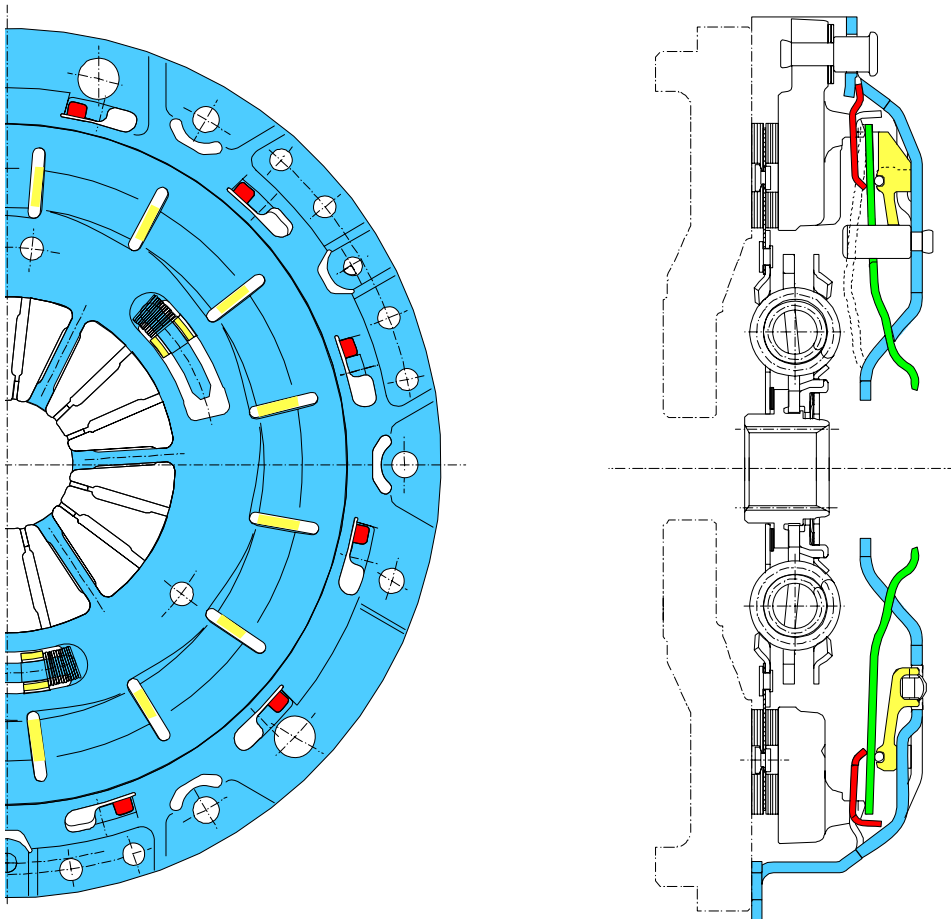


Figure 17: SAC-production model with mechanical stop

Release bearing overtravel is limited at the point where the diaphragm fingers meet the stop tab. Of course the stop must allow for a certain amount of diaphragm spring movement in direction of the flywheel due to the facing wear. Diaphragm spring assembly in the cover is still easy

because in free state the inside diameter of the diaphragm spring fingers is larger than the diameter of the stop tab.

In addition, the SAC-clutch requires a release system with low friction for reliable adjustment of smaller release loads.

Current developments in durability testing

At the time of this presentation, the first SAC-clutch with a diameter of 240 mm has been produced in quantity. Both customer and LuK vehicles have logged more than 2 million test kilometers with this design. Many vehicles have reached more than 100 000 kilometers. In some cases the disc had to be replaced because of facing wear.

Besides smaller problems, for which solutions were found immediately, only one more serious problem, involving overtravel in the release system, was encountered. The solution of this problem, the mechanical stop, was described above.

Summary

The self-adjusting clutch offers two main advantages:

- low release loads which stay constant over the life span of the clutch
- increased wear capacity and therefore longer life

This results in other possible secondary advantages, for instance:

- elimination of servo systems (in commercial vehicles)
- simpler release systems
- short pedal travel with low pedal effort
- new options to reduce the clutch diameter
- constant pedal load over the full range of engines
- shorter release-bearing travel during the life span of the clutch