Foreword

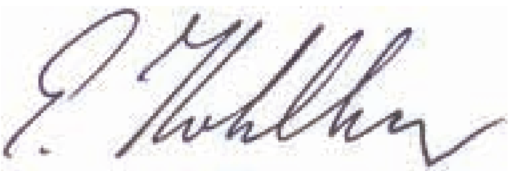
20 years ago, in 1978, the LuK Symposium was launched.

At that time LuK resolved not only to offer our customers delivery performance, quality and creativity, but to develop our work within a theoretical and conceptual framework. In partnership with the motor vehicle manufacturers, we decided to establish a long term basis for recognition and discussion of trends in vehicle drive train technology.

The original concept has proven successful, which is illustrated by the large number of participants, both from Germany and abroad, attending this year.

This document, detailing the lectures from the 6th LuK Symposium furthers our vision of technical developments in this field.

We look forward to an open discussion with our valued customers.



Ernst H. Kohlhage

Contents

The Self-Adjusting Clutch SAC of the 2nd Generation

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Introduction

The self-adjusting clutch (SAC) has proven itself in almost 1 million vehicles. in particular vehicles with large engines, the actuation of the clutch is achieved far more comfortably with the SAC. In addition, the goal of having a clutch which lasts the entire service life of the vehicle was also achieved with the SAC.

Despite the additional expense of the SAC, the total costs for the clutch system (clutch + release system) were actually reduced in several cases, for e.g., by:

- Elimination of a hydraulic booster
- Reducing the clutch size
- Standardizing clutch types and actuation system
- Eliminating one over-center spring

Figures 1 and 2 illustrate an overview of the current SAC applications and the expected development up to the year 2000.

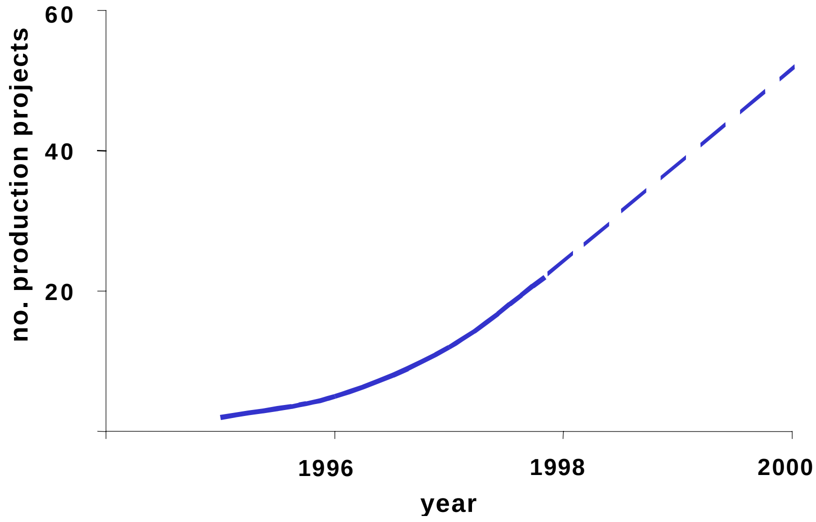

Figure 1: Trend in the number of SAC applications

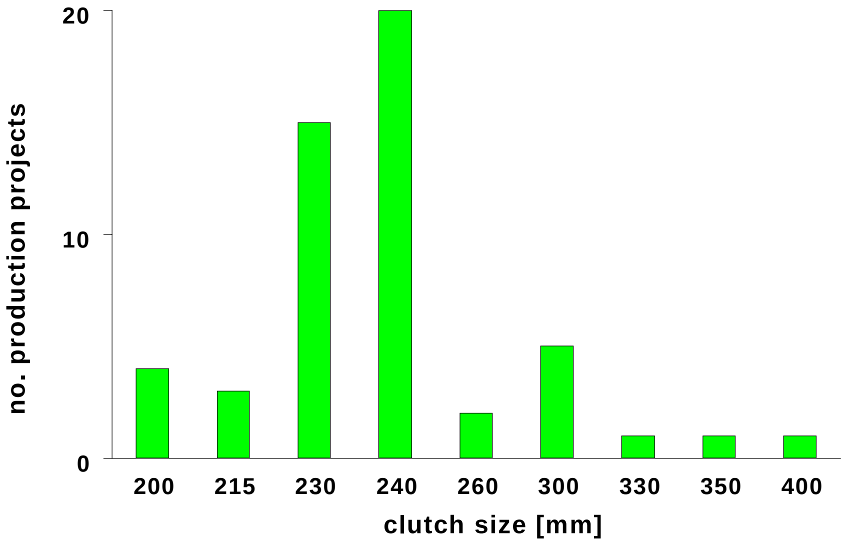

Figure 2: Distribution of SAC projects

The following summary can be made from the experience gained from the production projects:

- The SAC has a high functionality and functional reliability.
- The system can be manufactured without problems in mass production, despite the high demands of detail parts such as springs, adjuster rings, clutch covers and cushion segments, as well as having a more complex function.
- The introduction of the SAC significantly reduces pedal effort in comparison to conventional clutch assemblies. However, the possibility of drastically reducing pedal effort has not yet been fully exploited with the production designs, due to the clutch modulation and tendency to judder.

Ideas have been considered in previous years on how the SAC can be further improved, with regard to pedal effort and torque increase characteristic curve. In addition, there is extensive development potential to reduce the material and manufacturing costs with comparable functionality.

The various solutions are described in greater detail below.

Description of the SAC (Self-Adjusting Clutch) Function

With conventional clutches, the actuation force increases with increased facing wear. With the SAC, the facing wear is compensated for means of a wear adjusting system, so that there is no change in the actuation force.

The SAC differs from the conventional clutch by adjusting the position of the diaphragm spring during wear (Figure 3). The adjustment occurs such that the angle position of the diaphragm spring, and hence the actuation and clamp load, remain constant regardless of wear (primarily facing wear). In order to realize this wear compensation, the main diaphragm spring is not permanently riveted to the clutch cover or mounted with keyhole tabs, as with conventional clutch assemblies, but is only retained axially against the cover by a defined force (sensor force). A ramp ring, which extends into the ramp of the cover, is located between the diaphragm spring and the clutch cover, and is rotated by the coil springs.

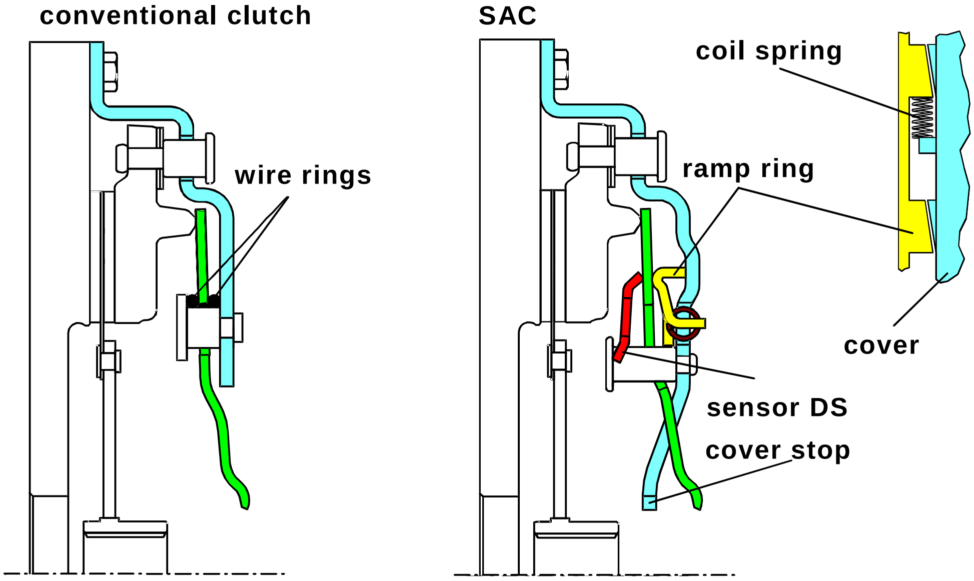

Figure 3: Comparison of a conventional clutch to a SAC

The sensor force is sized such that it can normally resist the actuation force. When the actuation force increases because of wear to the facings and the sensor force is no longer sufficient as a counter force on the main diaphragm spring, the main diaphragm spring moves axially away from the cover contact position, towards the engine. The resulting play is compensated by the preloaded ramp mechanism mounted between the diaphragm spring and the clutch cover. The adjustment procedure lasts until the actuation force has dropped to the sensor force, i.e., to the desired level, and the original diaphragm spring angle position is again achieved.

Figure 4a and 4b schematically illustrates the procedure of wear adjustment by the forces acting on the diaphragm spring.

Figure 4a: Conventional clutch new and worn

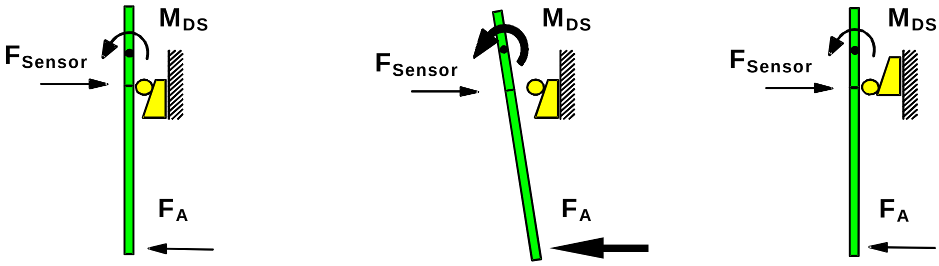

Figure 4b: SAC new and worn, before and after wear adjustment

In Figure 4a, the diaphragm spring of a conventional clutch is permanently seated symbolically at the rotation point. Due to its shape, the diaphragm spring supplies a torque, which is overcome via the actuation force on the diaphragm spring fingers during the actuation (rotation) of the diaphragm spring. The angle position of the diaphragm spring changes during wear, which causes an increase in the diaphragm spring torque and the actuation force due to the characteristic curve typical of a diaphragm spring.

With the SAC, the diaphragm spring - in contrast to the conventional clutch - is not permanently seated, but is only supported axially via the sensor force (Figure 4b). In the new condition, there is a force equilibrium between the sensor force and the actuation force. During wear, the actuation force increases and presses the diaphragm spring to the left, against the sensor force, so that the spring preload is relieved on the ramp on the right side of the diaphragm spring and it can then readjust. At the end of the adjustment procedure, the diaphragm spring again assumes it's initial angle position and there is a force equilibrium between the sensor force and the actuation force.

SAC Actuation Force Characteristic Curve

With the SAC as it is currently used in a wide variety of vehicles in production, the pedal effort was reduced significantly and hence the clutch comfort was increased in comparison to the conventional clutch.

The actuation force curve is, however, somewhat unfavorable with the SAC because there is a greater system-related difference between the maximum and minimum actuation force. Hence it is necessary to adapt the release system to the modified actuation force characteristic curve.

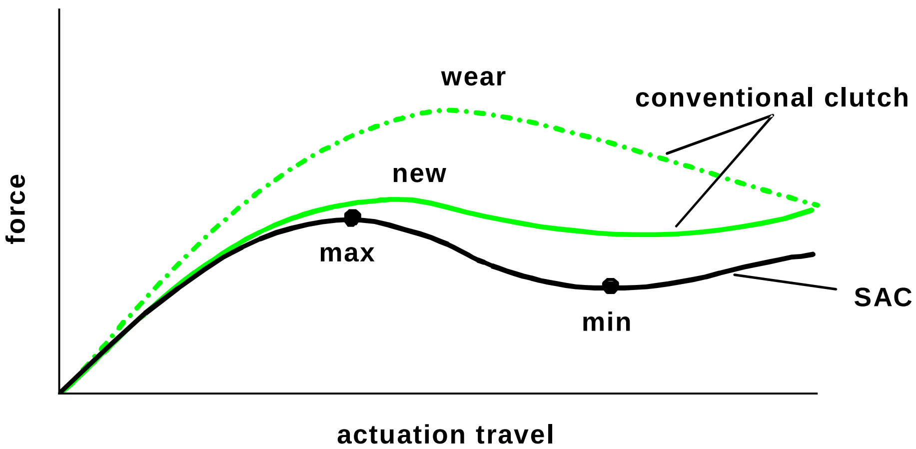

Figure 5: Comparison of the actuation force characteristic curve of a conventional clutch to a SAC

The reasons for this specific actuation force curve can be explained by the force equilibrium on the clutch diaphragm spring. If a free-body diagram of the diaphragm spring is made (Figure 6), it can be seen that the force of the cushion deflection (F_C), the leaf springs (F_L) and the sensor diaphragm spring (F_{SDS}) on the one side acts to counter the actuation force (F_A) on the other side. The sum of the facing, leaf and sensor spring forces can be designated as the total sensor force (F_S), which limits the amount of actuation force. If the actuation force becomes greater than the total sensor force during clutch release, which occurs during wear, the diaphragm spring is pressed away from the clutch cover (ramp ring) and the ramp mechanism can readjust.

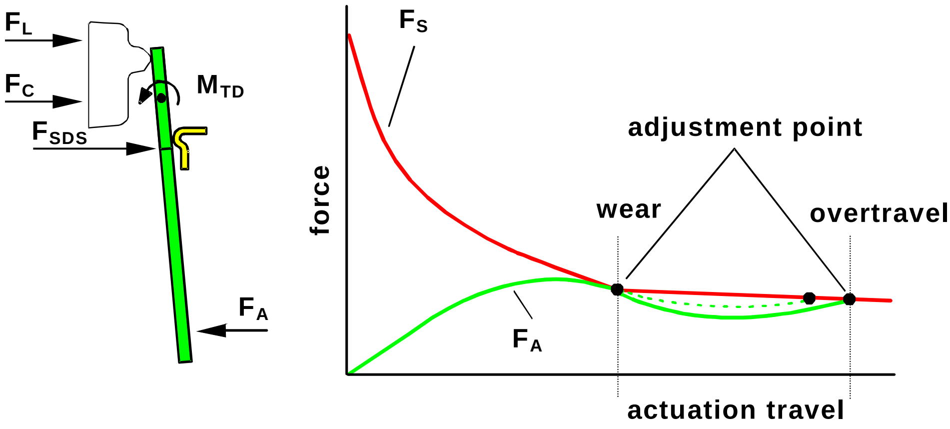

Figure 6: Forces on the main diaphragm spring in the SAC

In Figure 6, it is evident that the actuation force can exceed the total sensor force basically at two points over the actuation travel. The first point is the wear adjustment point. This point is in the range where the cushion deflection force is almost zero. The adjustment for wear occurs at this point. During continued release stroke, there is a second point (the overtravel adjustment point), at which undesired adjustment occurs. At maximum actuation stroke, there must be sufficient reserve to the overtravel adjustment point. This can only be ensured if the actuation force minimum is significantly lower than the total sensor force and / or overtravel can be avoided by a stop integrated into the cover.

From the current perspective, there are two technical options that can be realized with the SAC: there can be greater flexibility in the actuation force characteristic curve and the actuation forces can be further reduced.

SAC with Compensation Spring

A relatively simple way of generating an actuation force characteristic curve on the clutch, that is as flat as possible, is by adding a spring with a linear spring characteristic curve which is riveted onto the clutch cover. The compensation spring increases the minimum actuation force - related to the actuation force curve - and thus leads to a flatter total characteristic curve.

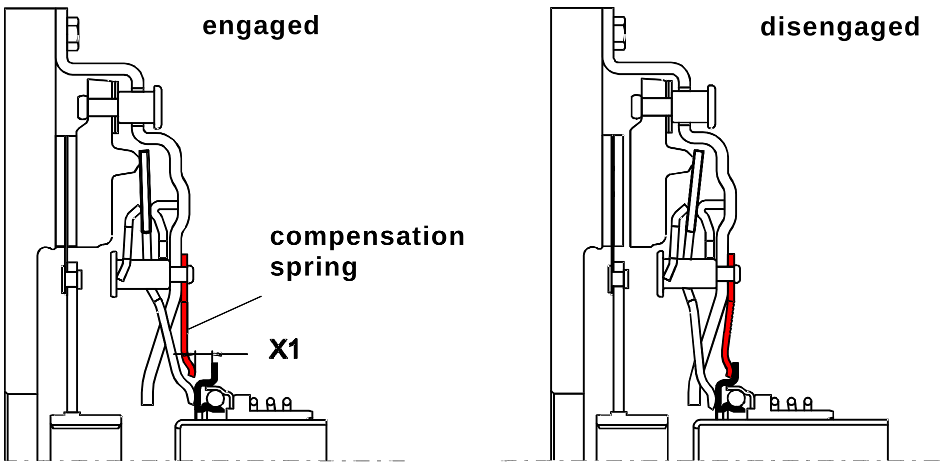

Figure 7: SAC with compensation spring

The compensation spring, as shown in Figure 7, directly affects the release bearing and only influences the actuation force characteristic curve and not the inner forces of the clutch. Hence, the adjustment mechanism and the adjustment function of the SAC are not influenced. Due to the migration of the diaphragm spring finger toward the engine during wear, the maximum actuation force remains almost constant only up to approximately 1.5 mm facing wear. If there is more wear, e.g., at 2.5 mm, the actuation force increases slightly by approximately 10%.

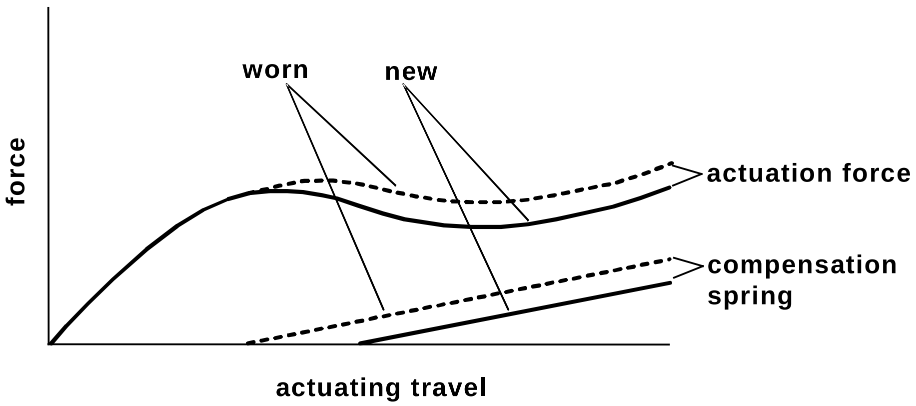

Figure 8: Actuation force characteristic curves with compensation spring

Without a compensation spring, the same function can also be achieved via diaphragm spring fingers that are seated deeper. When actuating the clutch, the deeper seated fingers contact the cover stop after reaching the maximum actuation force and are elastically stressed during further actuation.

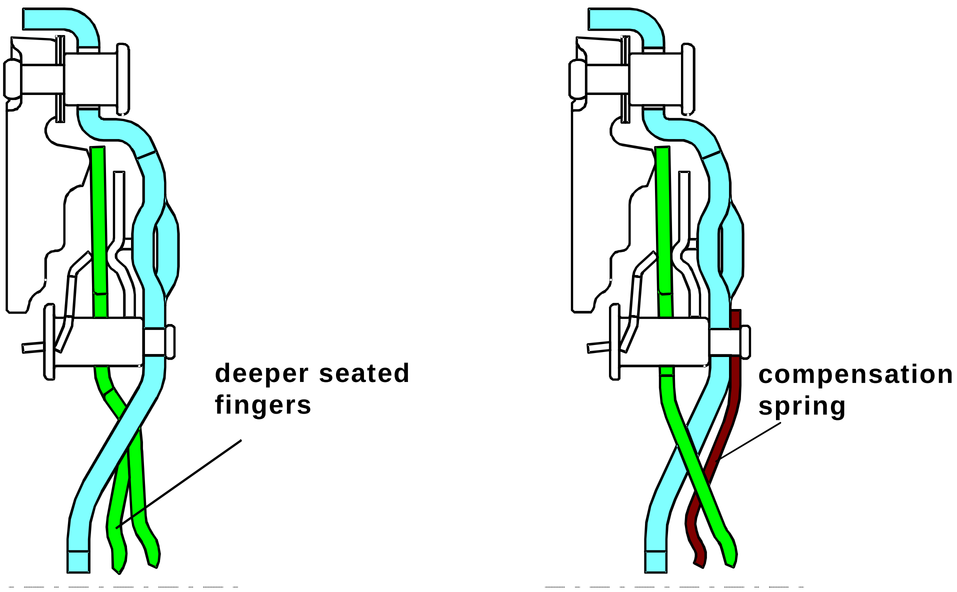

Figure 9: Comparison of the SAC with deeper seated fingers and SAC with compensation spring

SAC with Sensor Force Increase over Release Travel

Adding a compensation spring produces a flat actuation force curve by increasing the minimum actuation force. It would be better, however, to lower the maximum in order to achieve lower actuation forces in general.

This can be achieved if the total sensor force increases over the release travel after the adjustment point, by changing the tuning of existing spring elements, such as the main diaphragm spring, the sensor diaphragm spring and the leaf springs.

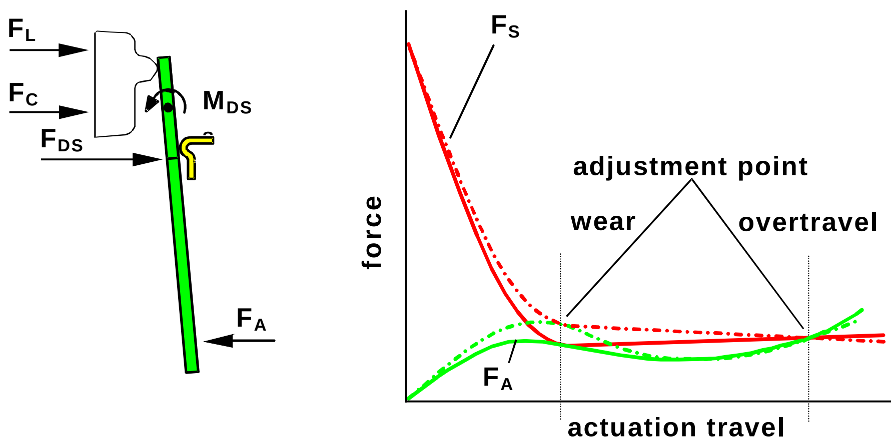

Figure 10: Actuation force characteristic curve for decreasing and increasing total sensor force

The effect is illustrated in Figure 10. Based on the force equilibrium on the diaphragm spring, the maximum actuation force is reduced while the overtravel safety and the minimum actuation force remain the same, if the total sensor force increases between the wear and overtravel adjustment points. The total sensor force between the wear and overtravel adjustment points decreases slightly in the current SACs, because the leaf springs relax with increasing actuation travel or lift.

It is possible to achieve an increasing sensor force in this range if leaf springs with degressive characteristic curves are used, instead of the leaf springs having linear characteristic curves.

Corrugated leaf springs maintain a degressive characteristic curve if they are fixed tightly to the clutch cover on both ends and if they are deflected in the center by the pressure plate. This causes a snap effect that is dependent on the corrugation.

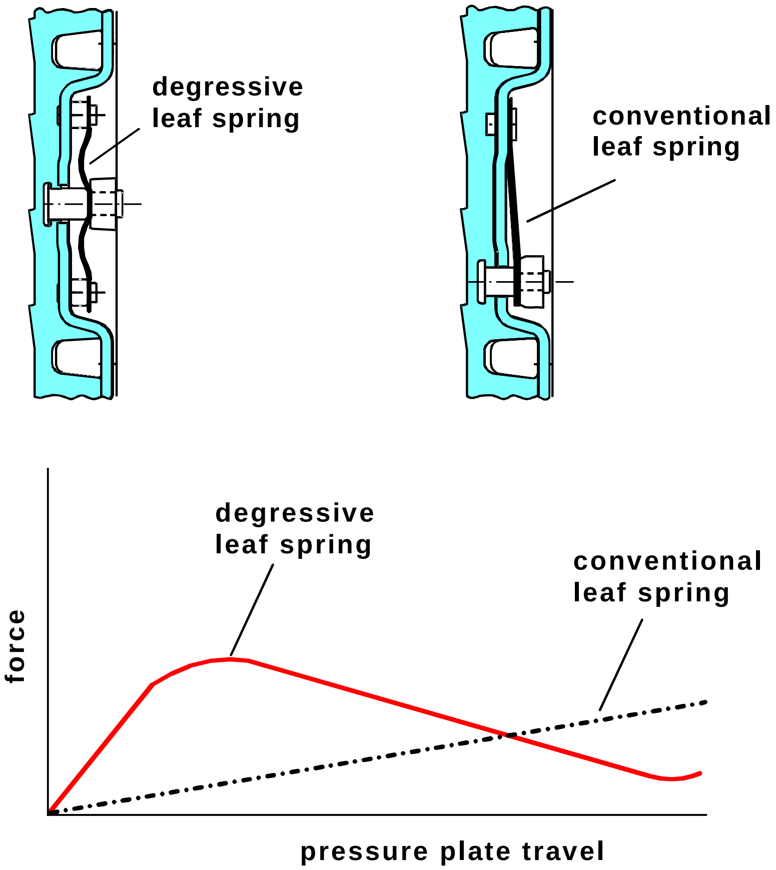

Figure 11: Leaf spring configuration with linear and degressive characteristic curve

The sensor force increase along the release travel is enhanced further, if the sensor diaphragm spring with increasing force characteristic extends radially further in at the diaphragm spring fingers instead of at the rotation point of the main diaphragm spring (Figure 12).

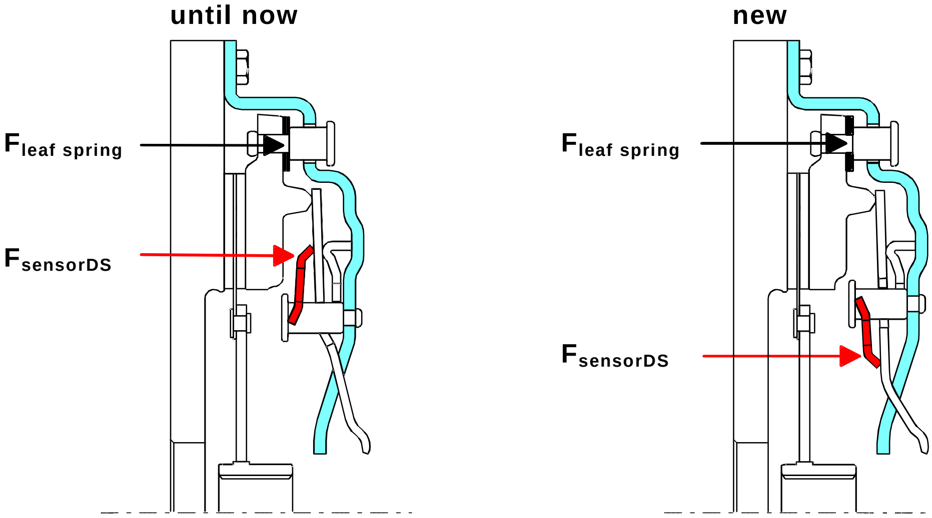

Figure 12: Sensor force increase via the sensor diaphragm spring

In comparison to the current SAC, an additional decrease in the actuation forces of 20-30% is possible. The drop in the actuation force characteristic curve (drop off) can be reduced by 50% by using leaf springs with a degressive characteristic curve and is thus comparable to conventional clutches.

The solutions presented thus far allow for a further reduction in the actuation forces, which is advantageous, for example, for high-torque engines or electric motor clutch actuation. There are a large number of applications (small vehicles), however, for which there is no need to reduce the actuation force when the clutch is new. Yet the SAC has the following advantages that are desirable even for these applications:

- constant actuation force over the service life
- higher wear reserve
- less axial installation in the diaphragm spring finger area

without significant cost increases in comparison to the conventional clutch.

For this reason, reducing costs in the further development of the SAC is a priority.

Reducing Costs

To implement wear adjustment in a clutch, a wear sensor and an adjustment mechanism are necessary. Hence, the SAC requires the following additional components or component modifications in comparison to a standard conventional clutch:

- Sensor diaphragm spring
- Adjuster ring
- Ramps in the clutch cover
- Coil springs to rotate the adjuster ring

In addition, assembly of the SAC is more costly in comparison to a conventional clutch.

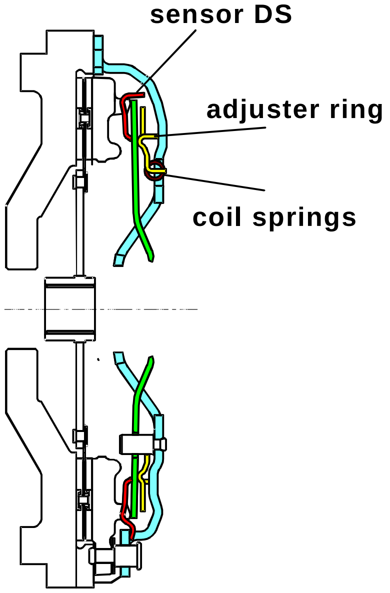

Figure 13: Additional components for the SAC

Efforts were made to reduce the number of additional parts and to combine multiple functions in the single components.

Two methods are described below:

Sensor Force From Leaf Springs

The total sensor force in the wear and overtravel adjustment points is determined essentially from the force of the sensor diaphragm spring and leaf springs in the SAC. Both spring forces are tuned such that the sum of both spring forces remains constant along the wear travel (e.g. 2.5 mm). Since the leaf spring force increases during facing wear, due to increased preload, the sensor diaphragm spring force must drop as the wear travel (preload path) increases. Therefore, the sensor diaphragm spring must have a degressive characteristic curve. The sensor diaphragm spring force can act on any point of the main diaphragm spring for the wear adjustment function, so the sensor diaphragm spring can also be replaced by leaf springs with a degressive spring characteristic (see Figure 14).

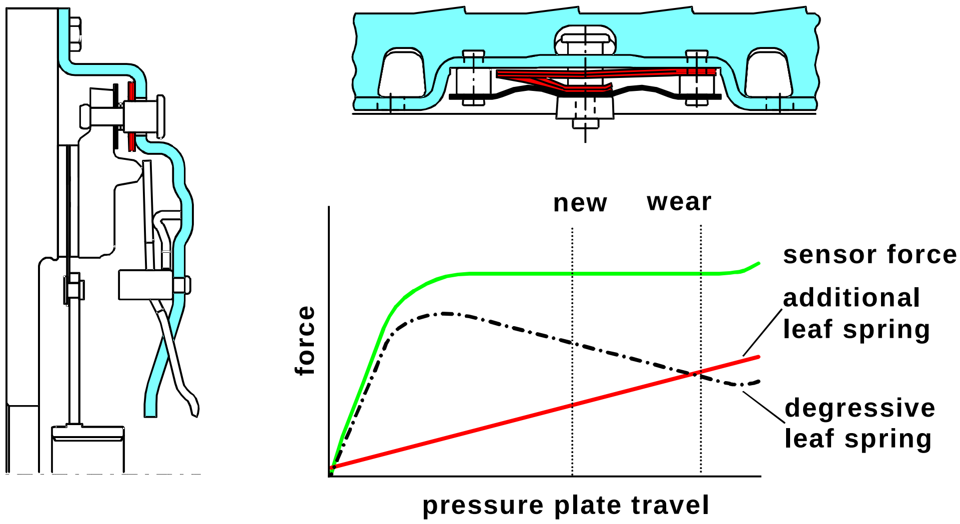

Figure 14: Sensor force from leaf springs

Since the sensor force in this design acts on the pressure plate, the diaphragm spring force is reduced. This must be maintained in the diaphragm spring design. In addition to the lower cost, this self-adjusting clutch is advantageous in that it requires less axial installation space because the sensor diaphragm spring is eliminated.

Sensor Force from the Main Diaphragm Spring

The necessary sensor diaphragm spring force can also be applied by individual, appropriately formed, diaphragm spring fingers. Thus the main diaphragm spring is supported against the cover with a defined force (sensor force) (Figure 15).

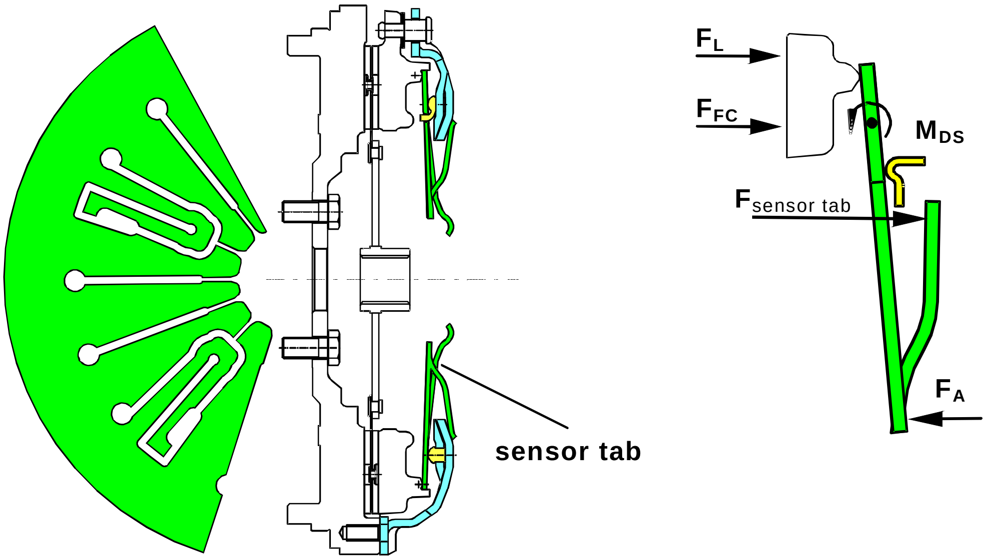

Figure 15: SAC with sensor force from the main diaphragm spring

During clutch wear, the increased actuation force again pushes the diaphragm spring away from the cover so that the wear can be compensated for between the diaphragm spring and the cover.

So that preload of the sensor tabs remains constant, the contact on the cover must be configured as a ramp and the diaphragm spring must rotate in relation to the cover during wear. The torque for the main diaphragm spring rotation results from the radial movement of the sensor tabs on the tangentially-sloped cover ramp, when the clutch is actuated.

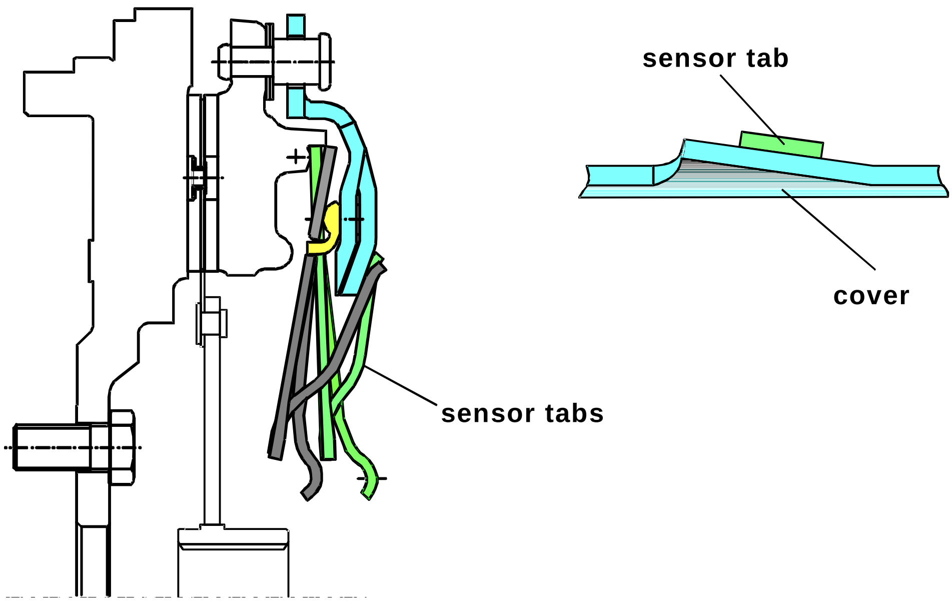

Figure 16 Sliding movement of the sensor tabs on the cover ramps

Since with this principle, the diaphragm spring rotates relative to the cover during wear, the coil springs, which generate a propulsive force on the adjuster ring in the SAC, can be eliminated. The adjuster ring must therefore be connected to the diaphragm spring so that the ring cannot rotate relative to the diaphragm spring.

In comparison to a conventional clutch, only a ramp ring is added in addition to the complicated diaphragm spring for the SAC II. However, the diaphragm spring retainer bolts and the pivot rings are eliminated.

Speed fluctuations in the drive train and particularly the irregularity of the engine, create varying torques on the diaphragm spring, which can lead to undesired adjustment in the SAC II. Hence, the SAC II is primarily unsuitable for any clutches in which the mass moment of inertia of the diaphragm spring must be kept at a low value.

Summary

The wear-adjusting clutch, SAC, has been implemented in the meantime in numerous high-torque engines with clutch sizes ∅ 200 - ∅ 300 mm. In addition to the description of the basic function, this presentation has served primarily to illustrate further development potential.

The development goals are to lower further the actuation force, i.e., optimize the actuation force curve, and to reduce the costs.

Options and advantages for the various developments are:

SAC with compensation spring

- More favorable actuation force characteristic

SAC with increasing sensor force

- Lower actuation force
- More favorable actuation force characteristic

SAC with sensor force from leaf springs

- Lower costs

SAC with sensor force from main diaphragm spring SAC II

- Lower costs

The SAC still has great potential for further development and with its additional advantages will be utilised in the future, in the lower vehicle class.

Literature

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[2] US patent 5,450,943

[3] SAE Presentation 1992 “LuK Self-Adjusting Clutch”

[4] LuK Colloquium 1994

[5] Albers, A.: Self-adjusting Clutch (SAC) and Dual Mass Flywheel (DMFW) to Improve Drive Train Comfort, VDI Reports 1175, p. 153 – 168

[6] Patent application DE 43 22 677

[7] Patent application DE 43 06 505

[8] Patent application DE 44 18 026

[9] de Briel, SAE 960981: Self-Adjusting Technology in a Clutch

[10] Patent application DE 42 39 289

[11] VDI Volume No. 1323, March 1997: Selbsteinstellende Kupplungen für Kraftfahrzeuge

Chatter - Causes and Solutions

Prof. Dr.-Ing. **Albert Albers**

Dipl.-Ing. **Daniel Herbst**

Definition of Chatter

Vibrations that arise during the slip phase of the clutch in the drive train of a motor vehicle and are generated in the clutch area should by definition be included under chatter. This definition is consciously kept general; it makes no statement on the causes of the vibrations. It is also used by other authors [1].

With an **engaged clutch**, the drive train can also vibrate in a frequency range similar to a true chatter. This “pseudo-chatter” can be caused by extreme lagging, defective engine mounts or a clunk in engagement and is often mistaken for true chatter.

Causes and Manifestations of Chatter

Chatter is caused when a periodic torque change is generated in a slipping clutch, whose natural frequency range is similar to that of the drive train dynamically separated from the clutch. The first natural frequency of passenger car drive trains is between 8 and 12 Hz under these conditions and thus with an engine speed of approximately 480 to 720 rpm (with a 1st order of excitation).

The drive wheels convert the rotating vibrations of the drive train to a longitudinal vibration of the vehicle. The chatter is expressed as a vibration in the longitudinal direction of the vehicle and is transferred via the operating elements and driver’s seat. The driver senses unpleasant vibrations (see Figure 1), which can also be connected to noises.

In the resonance range, even the smallest excitation amplitudes are enough to cause strong vibrations in the drive train. Hence, for example, certain drive trains with a maximum transferable clutch torque of 500 Nm can excite vibration amplitudes of 1 Nm, or 0.2 % (!), and generate clearly detectable chatter.

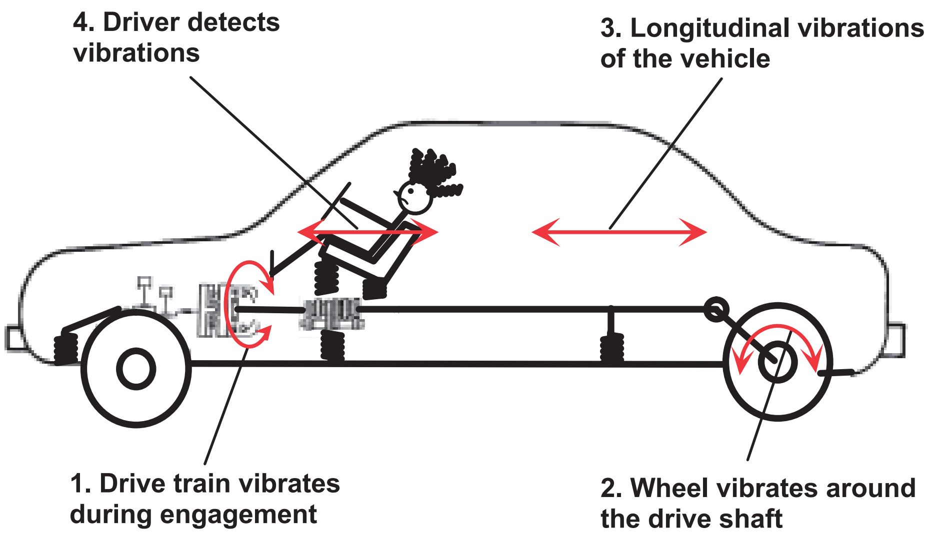

Figure 1: Causes and manifestations of chatter

Physical Causes of the Types of Chatter Vibrations

There are two different types of chatter vibrations that can occur:

- **self-induced** chatters (friction vibrations)
- **pressure-induced** chatters

The **self-induced chatter** is caused by a friction coefficient change with regard to the slip speed. Figure 2 shows a pseudo-model of this: a body is pressed on the belt by its own weight. Friction arises between the body and the belt. When the belt is set in motion, it takes the body with it because of the static friction, and deflects the spring. Above a certain spring deflection, the body remains still because the spring load corresponds to the static friction. There is a relative motion between the body and the running belt. If the dynamic friction coefficient of the contact becomes less than the static friction, the friction load suddenly decreases and the spring draws the body back over the belt until there is adhesion once more and the body is drawn forward once again. The process begins again from the beginning - the body vibrates.

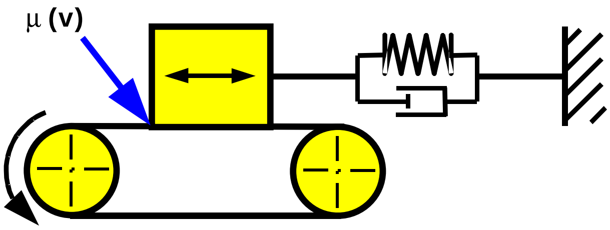

Figure 2: Pseudo-model for self-induced vibration

A vibration can thus only occur when the dynamic friction coefficient is lower than the static friction coefficient or the dynamic friction coefficient drops with increasing slip speed, because otherwise a stationary balance develops. If the dynamic friction coefficient decreases with increasing slip speed, the friction contact also acts as a stimulant, since the friction load - which counteracts the spring load - decreases when the slip begins and the body is accelerated more strongly via the spring load.

The characteristic size in this case represents the friction coefficient gradient. It is defined as the increase of the friction coefficient over the slip speed:

$$\mu' = \frac{d\mu}{d\Delta v}$$

There are three possibilities (see Figure 3):

1. The **friction coefficient decreases** with increasing slip speed: Energy is supplied to the system during connection, i.e., it is excited. This case was discussed.

2. The **friction coefficient is independent** of the slip speed: The friction contact behaves neutrally, the body immediately adopts a stationary state of balance.

3. The **friction coefficient increases** with increasing slip speed: The friction contact dampens because during back swinging, the slip speed and thus the friction load increases, which brakes the body. Energy is thus drawn from the system (at $\mu_H > \mu_G$). The body adopts a stationary state of balance here as well.

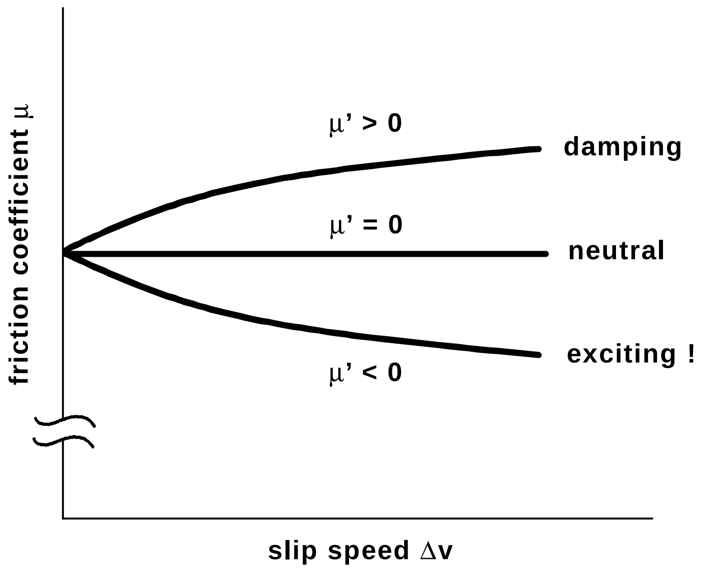

Figure 3: Principle friction coefficient curves

Pressure-induced chatter is the result of an outside impulse source with periodic excitation. The belt model can be useful here as well to understand the excitation mechanism (see Figure 4). A periodically changing normal force affects the body shown in the diagram. The current spring load also changes due to the changing friction load between body and belt and thus the equilibrium of the body on the belt. The body vibrates on the belt with the excitation frequency. If this frequency is the same as the natural frequency of the body-spring system, it results in resonance magnification and thus in large body vibration amplitudes. Naturally the pressure-induced chatter can also occur with neutral friction coefficient behavior, because it is excited by outside force modulation. The damping effect of the friction coefficient increasing with the slip speed naturally occurs again, because it counteracts an increase of the vibration amplitudes near the resonance.

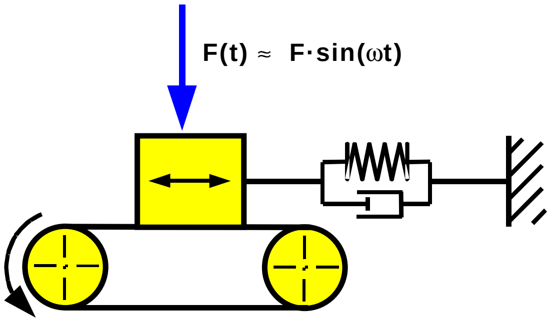

Figure 4: Pseudo-model for forced vibration

Chatter in the Vehicle

Measurement and Evaluation of Chatter

Occurrences of chatter in vehicles can be recorded through measurement and subjective evaluation.

With objective measurement, an acceleration sensor near the driver (e.g. on the seat rail) records the longitudinal vibrations of the vehicle. At the same time, the speeds of the transmission input and engine are measured. Figure 5 illustrates measurements of the longitudinal acceleration of the vehicle (upper diagram) and of the engine and transmission speed (lower diagram). The advantage of this process is that the measurement is independent of driver sensation.

However, a subjective rating of the chatter by an experienced driver using an evaluation system is indispensable. The driver can rate the vehicle from 1 to 10, for example, whereby a 10 is an absolutely chatter-free vehicle. This subjective evaluation has the advantage that it reflects the driver's sensation of the vibration and noise. Only this subjective sensation is relevant to the customer. Because of the overall increasing demands for comfort and the great improvements in the area of vehicle noise and vibration behavior (NVH) in recent years, the chatter assessment becomes more and more critical. The evaluation scale has changed. Naturally the limited selectivity and the dependence on individual evaluators must be taken into consideration with subjective evaluation. Statements must therefore be supported with basic statistical research. Correlation of acceleration measurements and subjective evaluation is possible by approximation. The **vehicle-independent**, objectively comparable measurement of chatter vibrations with actual correlation to the subjective sensation, however, has not been fully solved thus far.

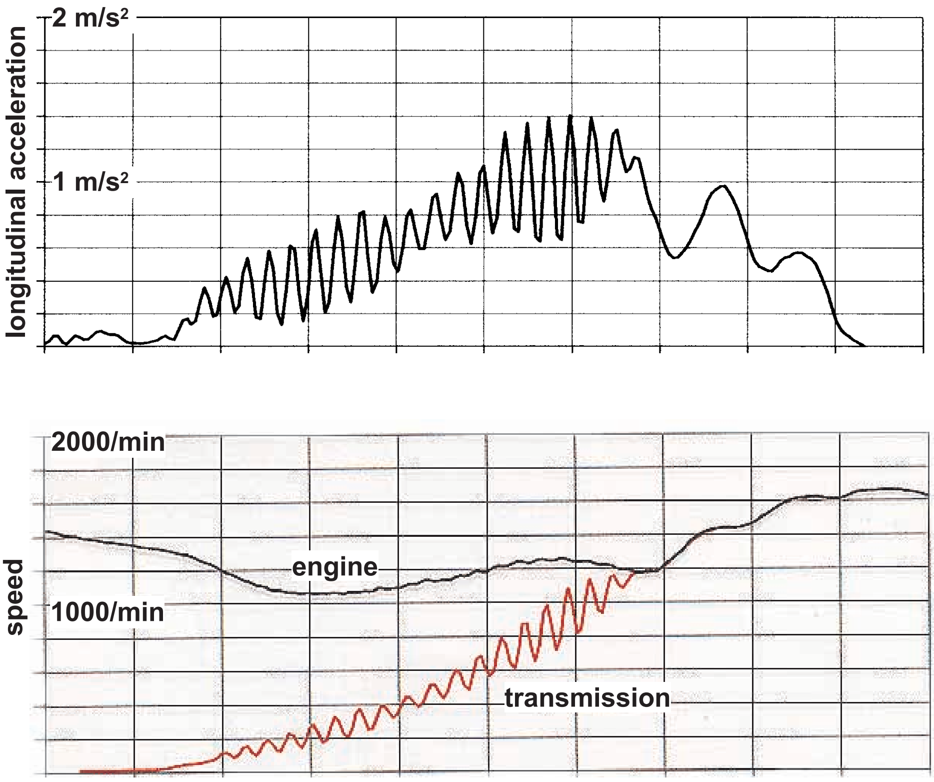

Figure 5: Forward vehicle acceleration (above), engine and transmission speed (below)

Developing Models

A vehicle's drive train can be represented by a torsional vibration chain consisting of rotating mass and couplings (springs, dampers and friction contacts). A pseudo-model, suitable for simulation, can be generated from six rotating masses (see Figure 6).

The important influencing variables are:

- engine (axial crankshaft vibrations)
- clutch with actuation,damping in the drive train
- overall transmission elements between the drive train and vehicle (tires, wheel suspension, etc.)
- vehicle layout (as inert mass)
- transfer from the vehicle to the driver (seat, etc.)

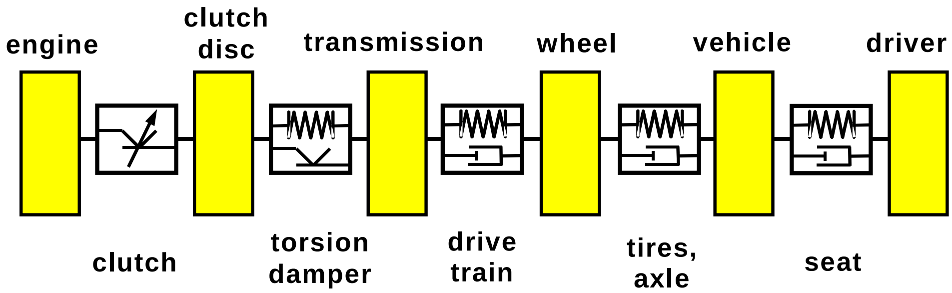

Figure 6: Six-mass model

The simulations represented in the following paragraphs were carried out with the program “TORS” [3], which was developed by LuK. In this program, rotating masses are connected with coupling elements, like springs, dampers and Coulomb friction. The simulation models reflect the chatter-triggering excitation mechanisms in the clutch area, and reproduce the suspension and damping qualities, as well as the natural frequencies of the drive train. This allows detailed parameter variations of the clutch and drive train. Individual parameters can be purposefully modulated without additional disruptive influences. The good correspondence between simulation and measurement data is shown in Figure 9 and Figure 10.

Self-Induced Chatter (Facing Coefficient Gradient Chatter)

As explained above, self-induced vibrations occur when the friction coefficient decreases while engaging during the slip phase with increasing slip speed in the friction contact. The friction coefficient is thus negative.

The friction coefficient gradients of today's facings lie between $\mu' = 0$ s/m and $\mu' = -0.015$ s/m. Figure 7 illustrates real friction coefficient curves of clutch facings. It becomes clear that the friction coefficient gradients that are discussed and relevant here are very low and may in no way be evaluated with the excessive increases often used in principle representations (see Figure 3).

With some newly developed facings, positive gradients have already been achieved. In practice, however, the "chatter-sensitive" facings with certain operating conditions also have a decreasing friction coefficient and thus have the potential for excitation. On the other hand, there is no damping-free drive train. For this reason, there is always a certain remaining damping, so that a facing with only a slightly decreasing friction coefficient can lead to an overall chatter-free vehicle. For vehicles built at this time with drive train damping, a slightly negative friction coefficient gradient of $\mu' = -0.002$ s/m is not critical (see Figure 7). If the friction coefficient increases strongly in the relevant slip speed range, damping occurs that can even eradicate the pressure-induced chatter. For this reason, a strongly positive friction coefficient gradient is the goal.

But even in such cases, the relationship can suddenly reverse itself when oil, grease or water enter the friction contact. The effect of moisture can be explained as follows: at high slip speeds, more heat is generated in the friction contact. Steam bubbles form, which allow the friction surfaces to float. Thus the friction coefficient is reduced. Rust-protection coating with sodium nitrite that was used formerly, prevents sticking, but is hygroscopic. Particularly after long periods of inoperation in humid weather, strong chatter can sometimes occur, because the water of crystalization is suddenly freed. After a few drives, the water evaporates and the chatter disappears again. Because the sodium nitrite is only used on the surfaces, this effect no longer occurs on facings that have been used longer. It may be surmised that a similar effect could occur with oil or grease contamination as with water.

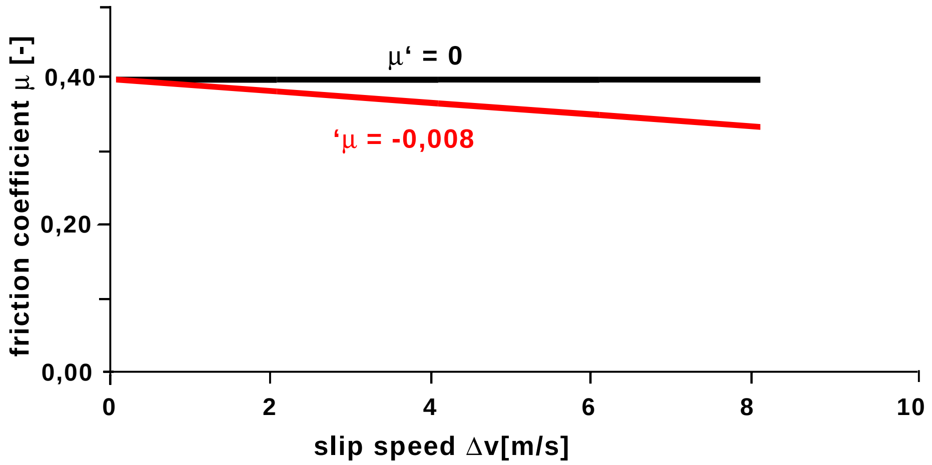

Figure 7: Friction coefficient curves

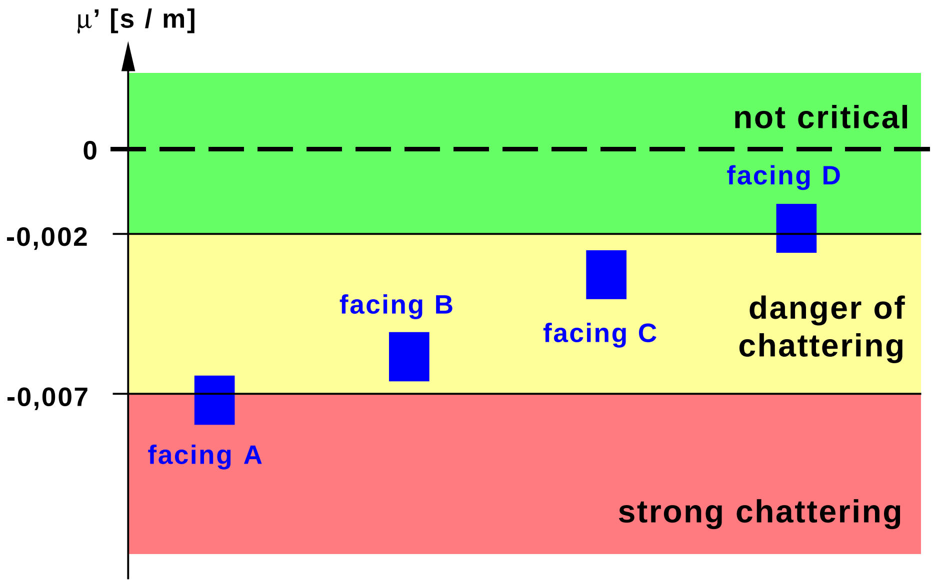

Figure 8: Friction coefficient gradients for different facings

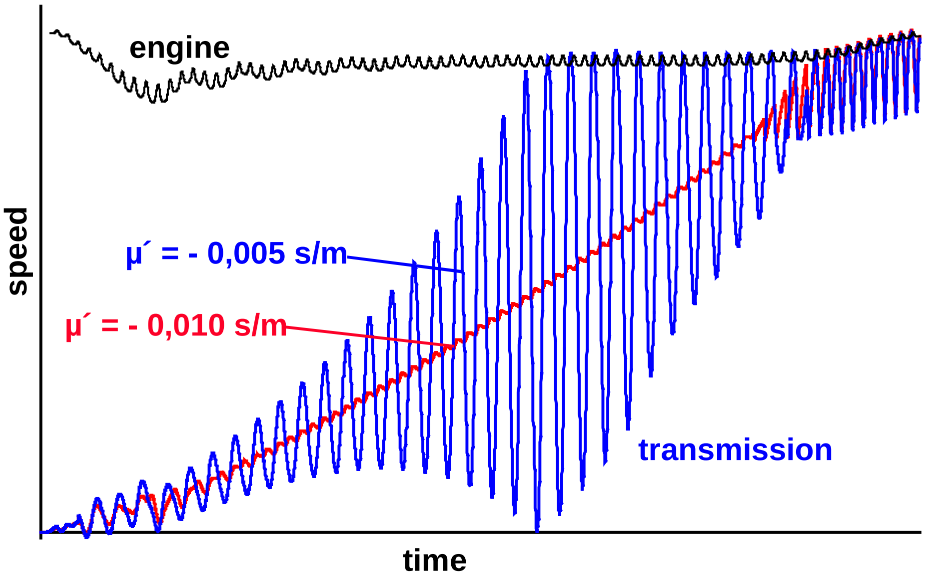

Figure 9: Facing chatter (simulation)

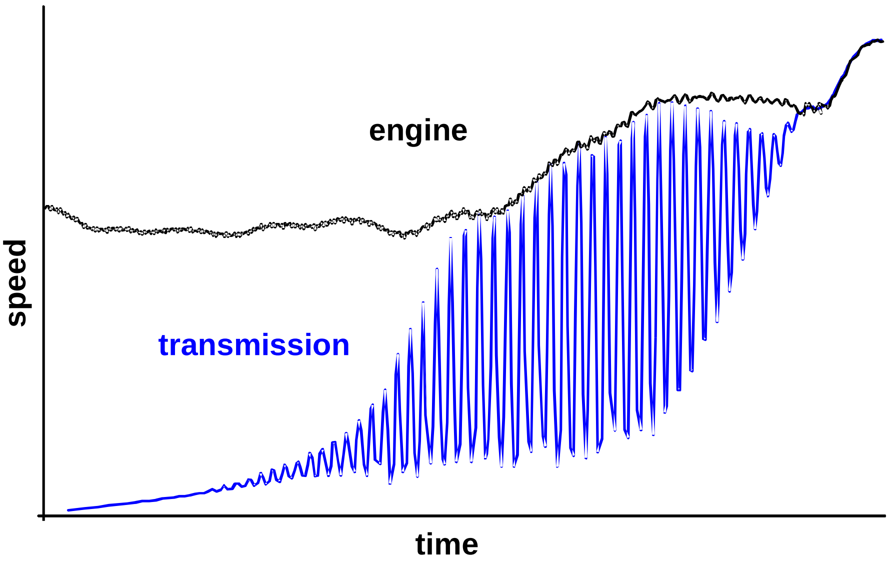

Figure 10: Facing chatter (measurement data)

Figure 9 shows the simulated torsional vibrations on the transmission input at friction coefficients of $\mu' = -0.010$ s/m and at $\mu' = -0.005$ s/m. With a more sharply decreasing friction coefficient, it is apparent that the amplitude surges further, until it is limited by the engine speed. At friction coefficient gradients of $\mu' = -0.005$ s/m, the drive train damping and facing excitation maintain near equilibrium. In comparison, the nearly identical measurement data shown in Figure 10 represent a chatter vibration induced by the friction coefficient.

Pressure-Induced Chatter

Variations in components and crankshaft axial vibrations lead to periodic clamp load fluctuations and thus to periodic torque fluctuations. The results are pressure-induced vibrations.

In order to generate a pressure-induced excitation, at least two deviations must exist.

This can be explained with a simple model (see Figure 11). A component deviation - here represented as a lifting of the pressure plate – which rotates with the drive speed n_1 and slips on the drive-side clutch disc with a relative speed (see “above”, “below” in Figure 11), still does not cause a clamp load change. If a second deviation is introduced - represented as angular displacement - the clamp load fluctuates during the slip phase according to the position of the pressure plate relative to the drive-side clutch disc. The output speed n_2 is thus irregular.

The **engine speed-dependent chatter** can be caused by axial crankshaft vibrations or the out of perpendicular of the diaphragm spring **and** skewed release of the clutch via the release system (see Figure 12). The frequency of the pressure-excitation is derived from the absolute engine speed.

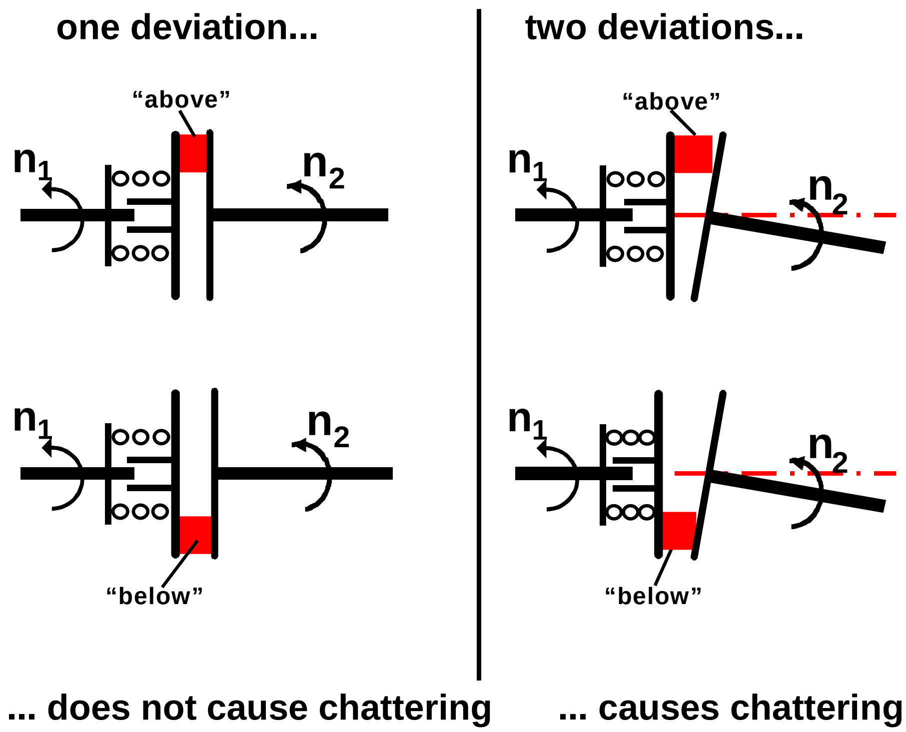

Figure 11: Model observation for pressure-induced chatter

For each combination of the different geometric disruptions

- **absolute (engine) speed-dependent**
- **speed differential dependent**
- **transmission speed-dependent**

chatter can be induced.

A release bearing travel of $\Delta s = 0.01$ mm at a maximum transferable clutch torque of $M_{max} = 500$ Nm leads to a torque change of approximately 1 Nm (see Figure 13). A deviation of the pressure plate travel has an even stronger effect (see Figure 14). The chatter excitation increases with the same geometric disturbance with the transferable torque. More powerful motorized vehicles also usually have more danger of chatter.

Engine speed-dependent chatter can occur during the entire drive-off (see Figure 15).

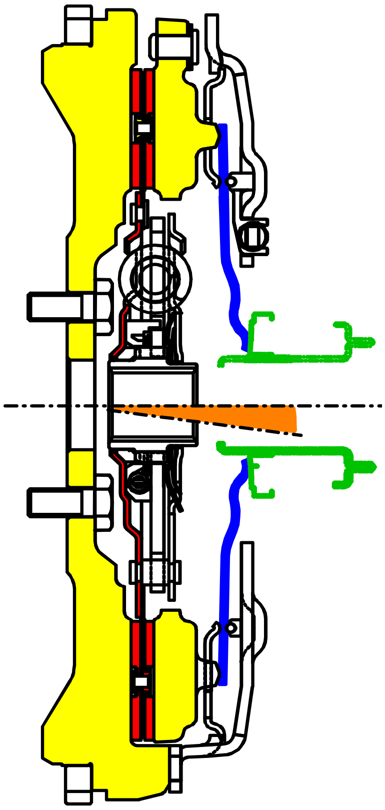

Figure 12: Geometric disturbances in the clutch system with actuation:

- Axial vibrations of the pressure plate (yellow)
- periodic finger movement of the diaphragm spring (blue)
- support on the release bearing (green)
- skewed lift-off of the pressure plate (yellow)
- skewed lift-off of the clutch disc (red)
- positive deviation of the drive shafts

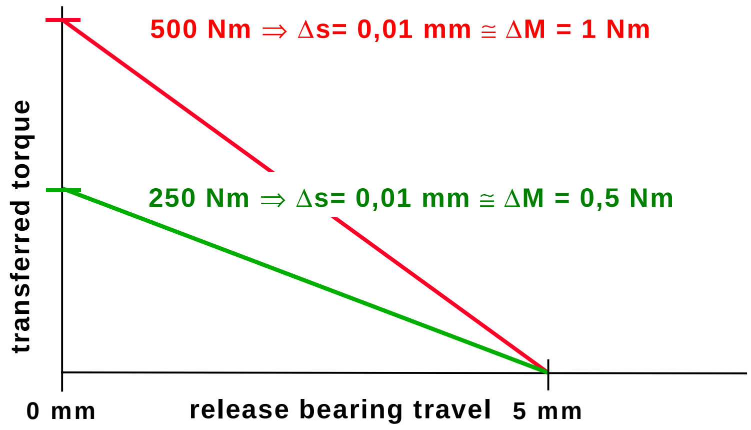

Figure 13: Clutch torque / release bearing travel dependence

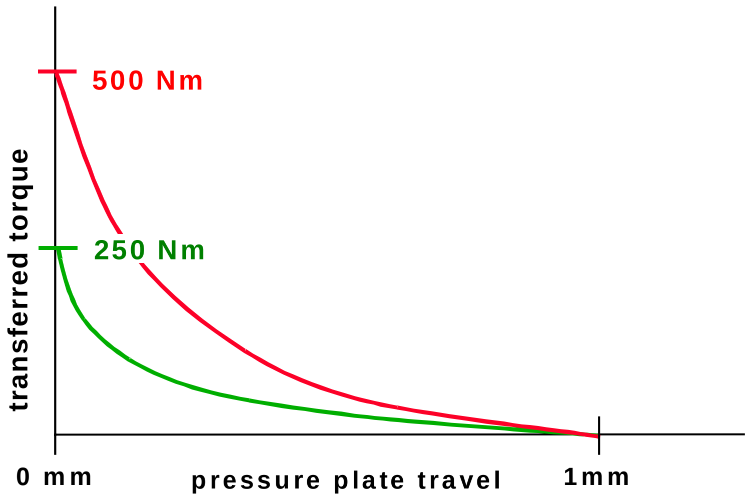

Figure 14: Clutch torque / pressure plate travel dependence

Chatter dependent on the speed differential is elicited by deviation in parallelism on the clutch pressure plate, deviations in the clutch disc and angular displacement between the crankshaft and transmission input shaft (see Figure 12). It only leads to chatter during engagement when the speed differential between the clutch disc and pressure plate is within the resonance range (see Figure 16).

Chatter dependent on transmission speed only occurs with deviations dependent on both engine and speed differential. It represents the most harmless of all three pressure-induced forms of chatter, because the resonance range is only completed with very low speed difference, shortly before the clutch closes (see Figure 17).

The deviations normally move within the (sometimes very narrow) determined tolerance ranges in a static distribution and have mutual influences.

The pressure-induced chatter is thus to be seen above all as a static problem, as two possible extreme cases should make clear:

- All components deviate only slightly from the ideal values. The effects of the deviations, however, happen to build up and generate a strong chatter.
- Some deviations are on the tolerance border. The effects, however, happen to increase, and no chatter is generated.

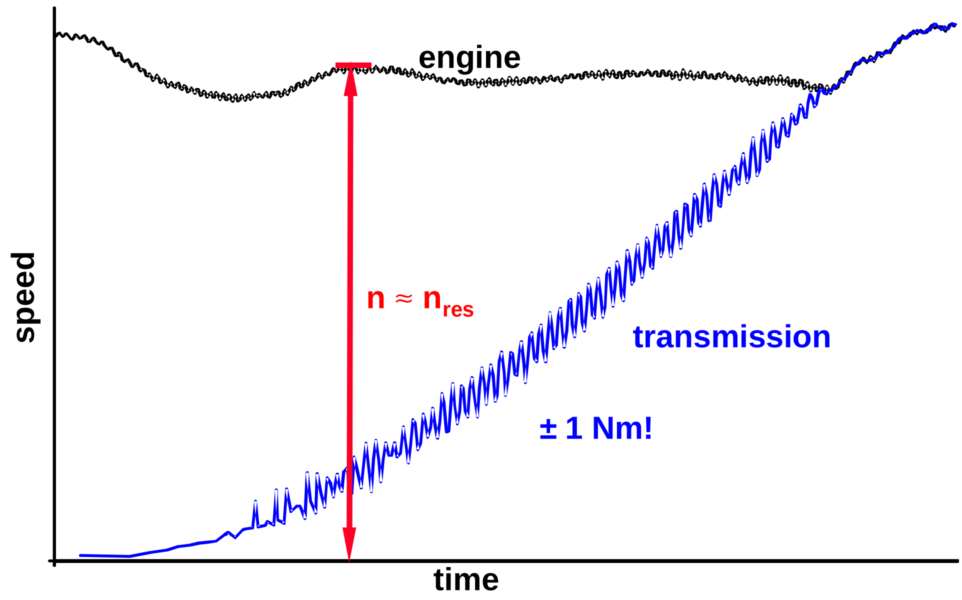

Figure 15: Chatter dependent on engine speed (measurement data)

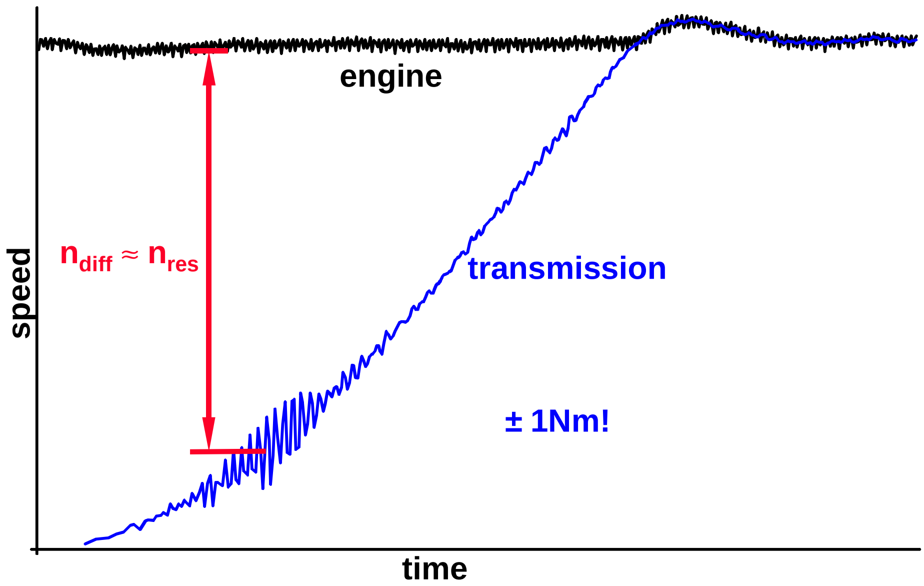

Figure 16: Chatter dependent on slip speed (measurement data)

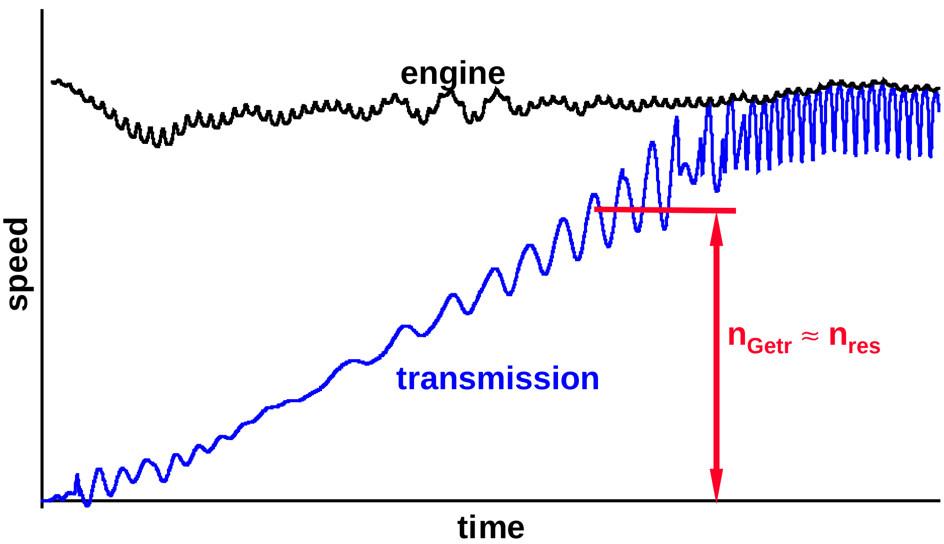

Figure 17: Chatter dependent on transmission speed (simulation)

Influence of the Drive Train Damping

High damping in the drive train reduces the chatter amplitudes with pressure-induced chatter. In the case of facing chatter, high drive train damping can eliminate chatter almost completely (see Figure 18), if the damping value outweighs the excitation from the facing. The drive train damping of recent vehicles is between 0.05 and 0.10 Nms. It is, however, based largely on friction (transmission, bearings, seals, etc.) in the overall drive train. Because of this, however, the friction losses are also higher. Since the general trend in vehicle design is toward increasing efficiency and lowering fuel consumption, the damping in the drive train decreases more and more and the chatter sensitivity increases. As an example, Figure 19 illustrates the chatter ratings of two higher-class vehicles of the same type, but of different model years, whose **system chatter excitation was the same**. The clear deterioration of the chatter rating and the measured longitudinal vibration with the same chatter excitation shows the increasing chatter sensitivity.

This connection should be considered in the specification for new vehicle models. In particular, the best solution must be found in early overall observations of the vehicle and its drive train, because a general optimization of the clutch does not lead to a technical and economically defensible result.

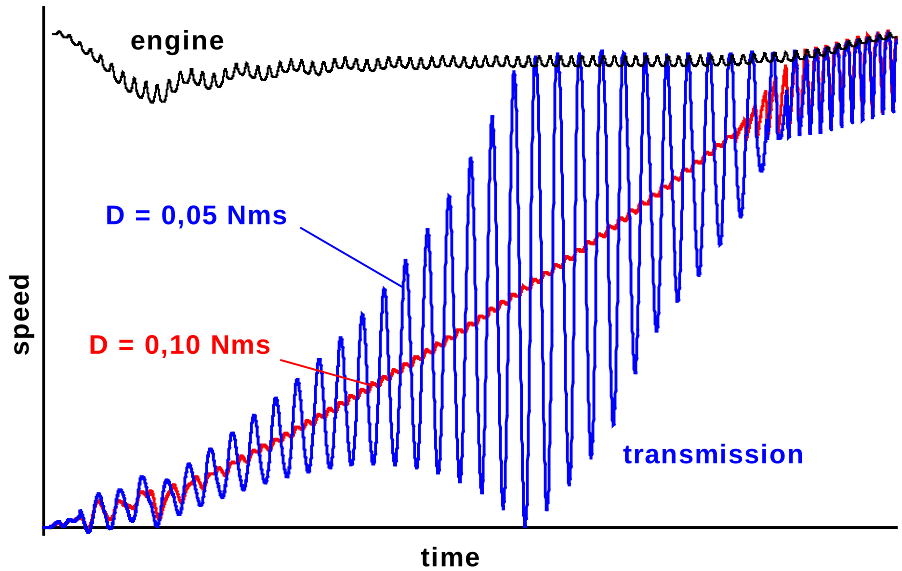

Figure 18: Consequences of different drive train damping based on the example of the facing chatter with $\mu' = -0.010$ s/m (simulation)

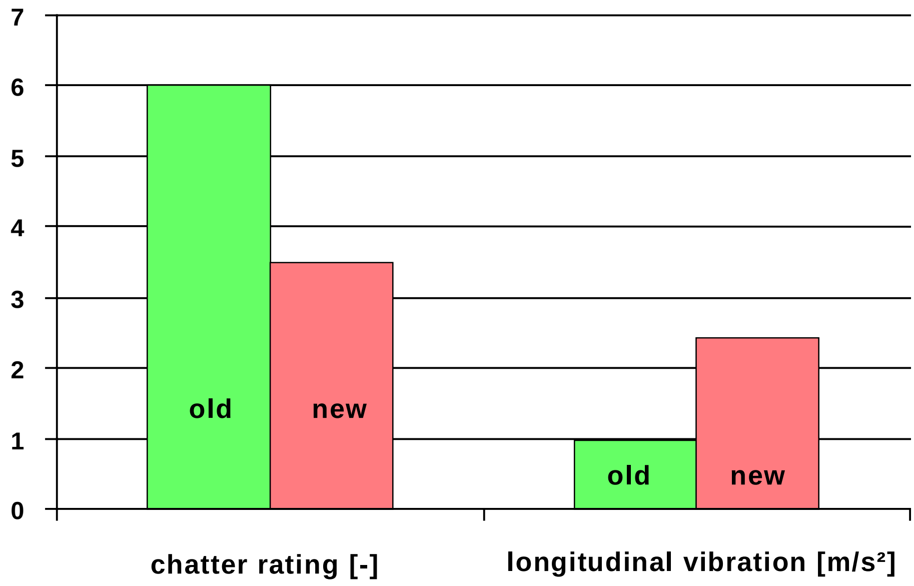

Figure 19: Comparison of old and new models of the same type

Transfer of Vibrations to the Body

The vibrations of the drive train are transferred to the vehicle body. Transfer elements are in order of the flow of force (see Figure 20):

- engine mounts
- transmission bearings
- drive shaft bearings
- tires
- axle suspensions

In several measurements, the transfer relationship between the drive train and the layout of different vehicles was determined. Apparently the transfer function is dependent on the vehicle weight. Otherwise, it is influenced by the elements named above, which are usually made of rubber compounds. The transfer function is thus non-linear.

Defective bearings (mainly engine mounts and drive shaft bearings) substantially increase the tendency toward chatter.

With tires (see Figure 21), contact with the roadway comes into play as another important factor (tire contact surface). The torsional and longitudinal vibration dynamics of the drive train has often not been considered enough thus far. There is still research potential here for the near future. Research projects at the Institute for Machine and Vehicle Design are being conducted about the problem.

Finally, it should be mentioned that even the vehicle seats have a large influence through higher or lower damping characteristics and thus influence the subjective sensation.

The complete cataloging of the chatters and their effects on the driver can only succeed through further improvement in the simulation models and through observation of the "human" transfer function.

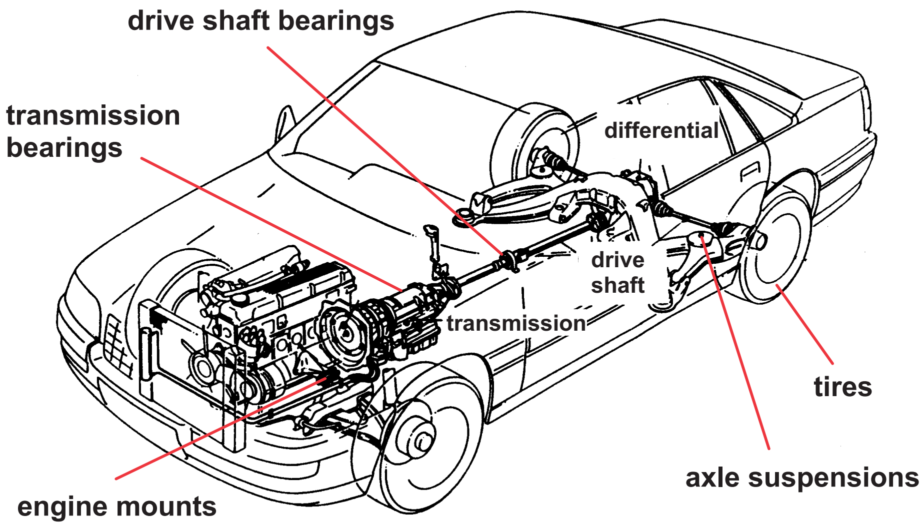

Figure 20: Transfer elements between drive train and vehicle body

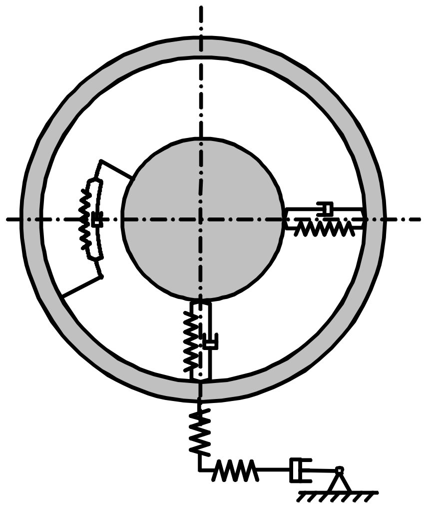

Figure 21: Transfers to the tires

Countermeasures and their Limits

Friction Coefficient

A friction facing with an increasing friction coefficient curve over the slip speed has damping characteristics. Mass-produced friction facings, however, exhibit no such behavior over the entire temperature range.

If the friction coefficient gradient in the relevant slip speed range exhibits a clearly increasing positive curve, chatter can be completely avoided. If such a friction facing is successfully developed, the pressure-induced vibrations could also be reduced. No more detectable chatters would occur.

The development of dry-running friction facings with limited positive friction coefficient gradients should therefore be advanced. For this to occur, an exact physical and chemical understanding of the friction pairing in the clutch is necessary.

Further Restriction of the Production Tolerances

With a further restriction of the production tolerances, only the vibrations caused by geometric component deviations decrease. This method makes the production process more expensive because multiple tolerances that influence the chatter must be considerably reduced in order to obtain the desired result. These measures can only lessen the chatter but not prevent it, as long as a facing with chatter-sensitive quality is used, because the pressure-induced chatter can only be reduced in this way.

With today's clutches, a pressure plate typically has a straightness value, measured at a particular circumference, of 0.1 mm. A reduction in this value means a considerable additional production expense, due to surface grinding, for example.

Softer Cushion Deflection

A softer characteristic curve of the cushion deflection in the clutch disc leads to a lower deviation of the clamping force with geometric deviations of the contact elements, and also of the transferred torque (see Figure 22). It is thus possible for given geometric deviations to reduce the chatter excitation generated in this way. The softer characteristic curve can only be realized in some areas with attention to the geometric relationships and the final increasing clamping force. The effects on the other system relationships must be considered.

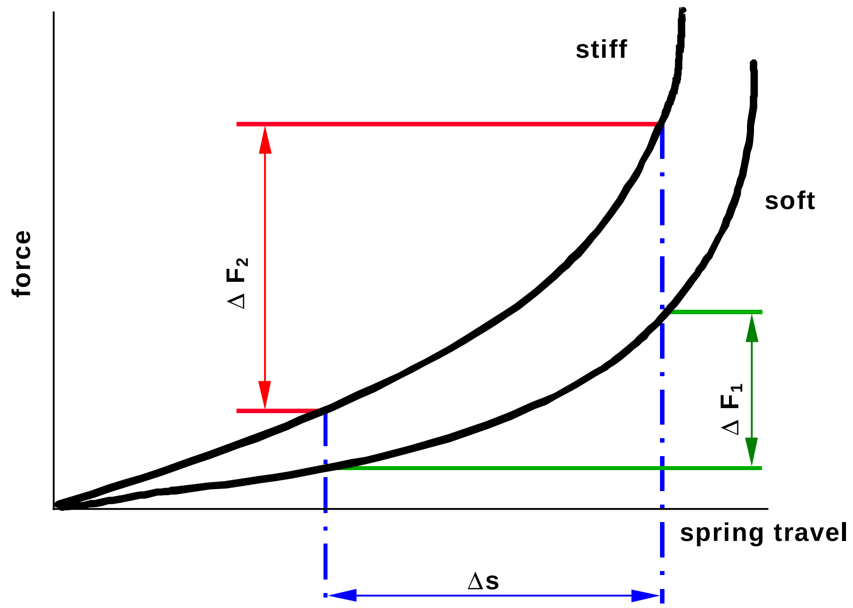

Figure 22: Friction facing curves

Damping in the Drive Train

High damping in the drive train can entirely eliminate the facing chatter and reduce the pressure-induced chatter. A damping increase, through higher viscosity transmission lubricants, for example, is not realized because it is not effective enough.

The chatter vibrations occur only during the slip phase of the clutch. Thus it is worth considering integrating a switchable vibration damper. An electronically controlled eddy current brake would be conceivable here. The development is nevertheless made difficult by the specification guidelines (installation, weight, cost).

Countermeasures on the Assembly Side

The pressure-induced chatter can be **effectively reduced** on the assembly side. Here, all measures are significant that generate components with the natural tensions and deformities within tolerances. The force-free screwing down of the pressure plate can - as many concrete applications can prove - improve the geometric disturbances in assembled clutch systems.

Summary

Chatter occurs only during the slip phase of the clutch and is divided into two different types:

- self-induced facing chatter induced by the friction coefficient
- pressure-induced chatter as a result of component deviations and axial vibrations

The significance of the facing chatter is decreasing because the facings are getting better. For this reason pressure-induced chatter occurs more often, because the drive train damping of modern vehicles is falling for efficiency reasons and thus ever-smaller deviations in the torque in the clutch area lead to chatter problems in the vehicle. In addition, engine performance is increasing on average, whereby the clutch must transfer more torque and thus the torque deviations increase. Finally, due to increasing comfort demands and the clear improvement in the vehicle’s general noise and vibration behavior achieved in recent years, sporadically occurring small chatter becomes more relevant to the customer.

The chatter vibrations are influenced not only by the clutch and the actuation system itself, but also by the engine, the drive train and the drive shaft, the axle suspension and the vehicle layout. All transfer elements influence the chatter sensitivity and must be considered.

A chatter-insensitive clutch without additional new components or assemblies is possible with today's technology if:

- the facing has an increasing friction coefficient gradient
- a soft cushion deflection is built in
- the production tolerances are logically restricted
- natural tensions are avoided in assembly

Understanding the causes, transfers and effects of chatter vibrations in relation to the entire vehicle system must still be further improved through research in order to avoid or protect against chatter in the development of new vehicles now.

Finally it must be determined that chatter **cannot** be prevented long-term through isolated measures on one part of the vehicle - such as for example the clutch - but only through observation and tuning of the entire vehicle system.

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Clutch and Operation as a System

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Dipl.-Ing. **René Shead**

Introduction

New technologies and increasing demands for comfort, require increased total system thinking, also in the area of clutches and clutch actuation. In addition, the automotive industry requires system suppliers in this area, who can optimize the functional chain in practical ways.

LuK has taken on the task of understanding the theory of the clutch operation, including the dynamics, and of improving it from the pedal to the transmission input.

This presentation will examine the individual load transferring components and how the driveaway performance is improved by tuned interaction and how the actuation force can be reduced.

General Goal

The clutch manufacturer is required to develop an optimal pedal solution with the clutch parameters (Figure 1).

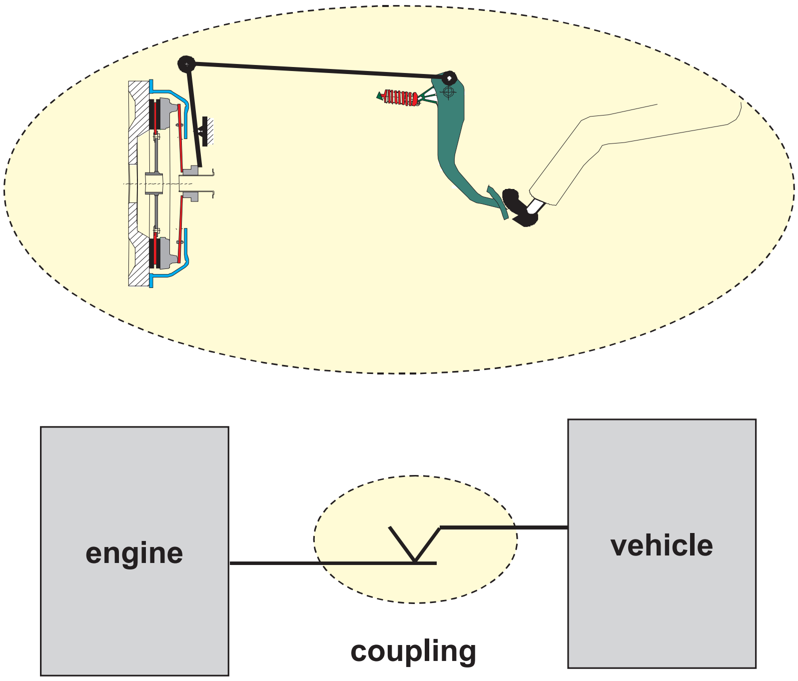

Figure 1: Clutch / operation set-up

For the driver, this should provide a vehicle with a clutch that is guaranteed to have

- flawless release behavior,
- favorable actuation pedal force and pedal travel characteristics,
- noise- and vibration-free actuation, and
- a good modulation behavior during driveaway and shifting.

The clutch manufacturer can only influence a part of the major parameters to meet the above demands.

Hence, it is apparent that optimal function is only achieved if the entire functional chain is considered.

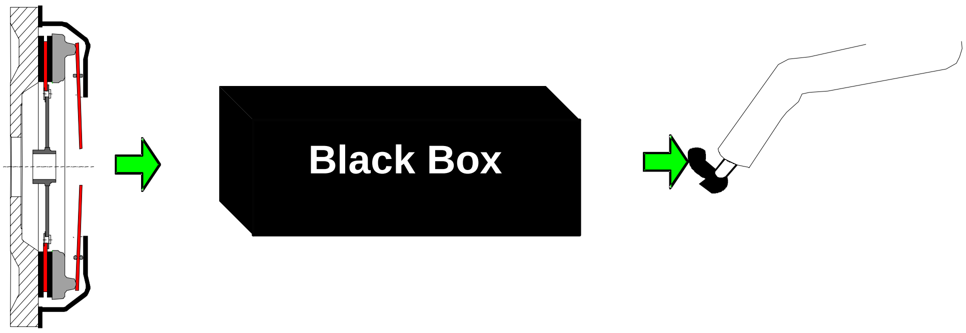

Figure 2 : Previous consideration of the release system by the clutch manufacturer

Only if all part functions of the clutch system are tuned to one another in a practical way and if the influences from the engine and chassis are considered, can a first class overall function of the clutch system be expected.

The overall view must be guaranteed by this system consideration, which was complicated thus far by the different requirements of the car manufacturers for the engine, transmission and chassis.

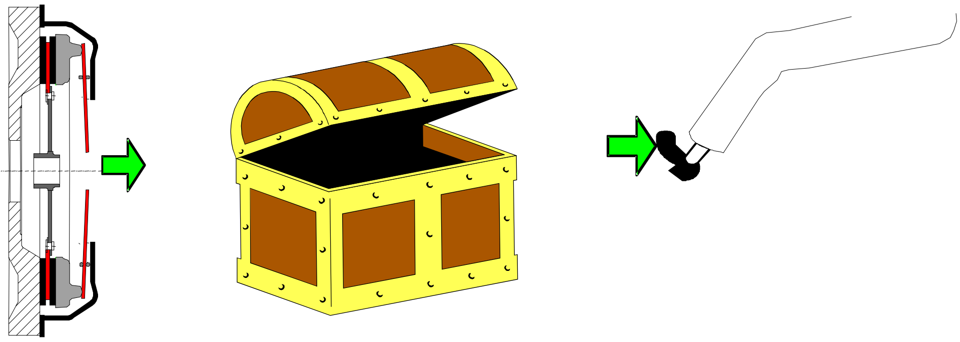

Figure 3 : Consideration of the release system by LuK

The previous black box "release system" (Figure 2) should be broken down, with the intent of better exploiting the potential of clutch and actuation and hence of optimizing the system overall.

Therefore, a few years ago, LuK established a team of five engineers who took on this task and determined a series of new effects. The most important parameters to influence the entire system will be summarized below.

Elasticity in the Clutch / Operation System

Figure 4 illustrates the travel transmission function "release travel over pedal travel" of an actuation system. The ideal curve as well as a measurement at room temperature (green line) and in warm operating conditions (red line) are shown; deviations from the ideal curve represent the travel losses of the release system. The increased elasticity of the release system as a function of the temperature leads to a significant shift in the clutch separation point towards the end of pedal travel.

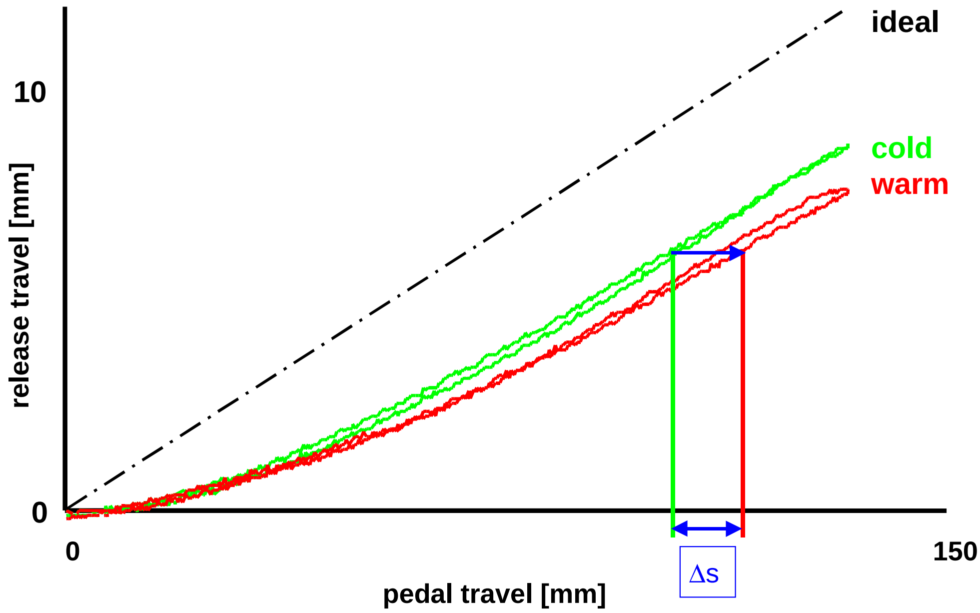

Figure 4: Measurement of the release travel when cold / warm

The transmission function of the entire release system can essentially be described by a single ratio and elasticity.

In order to determine the loss portions of the individual actuation components, as well as the dependence of these losses on temperature, the representation illustrated in Figure 5 was used for the entire system analysis.

The travel losses on the pedal (x axis) present for the defined release force on the clutch (y axis) are shown here. Hence, for maximum release force, the loss travel increases from 30% of the total pedal travel to 55% in warm conditions.

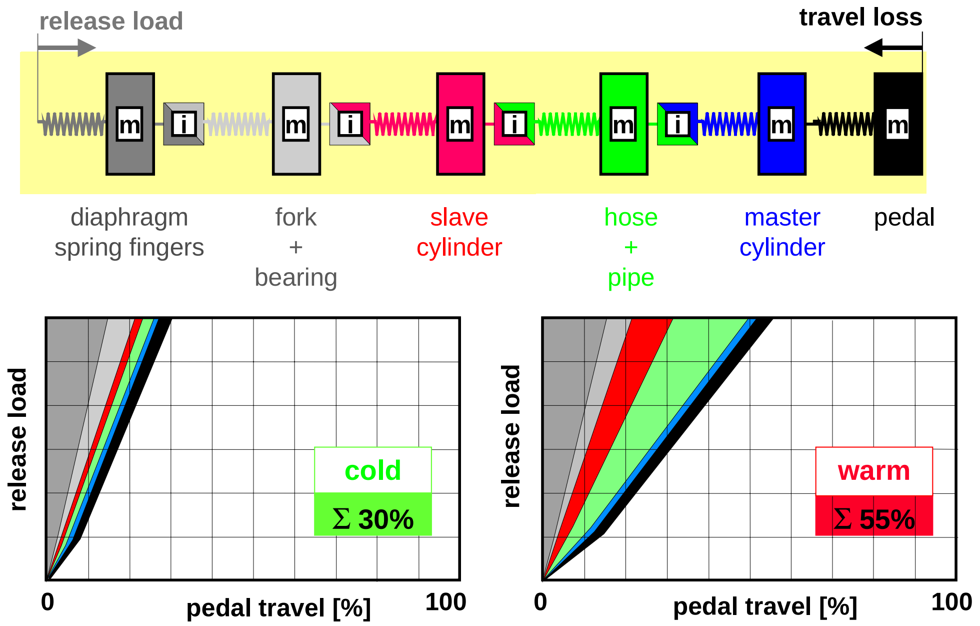

Figure 5: Elasticity's (reduced to pedal travel, cover stiffness not shown)

The illustration of individual elasticity's, reduced to the pedal travel, shows which elasticity's influence the total travel loss the most.

Depending on the quality of the system, the travel loss amounts to up to half of the pedal travel. The components (diaphragm spring fingers, release fork) located on the high force level and low ratio stage have the most influence on the position of the coupling points, hence on the beginning of the torque build-up and on the separation behavior. At the same time, it is apparent how the elasticity's change due to temperature. In the above example, the components – slave cylinder and hydraulic line – exhibit the most potential for improvement.

With this illustration, it is possible to usefully evaluate the different elasticity's in the mechanical system, the semi-hydraulic system and the central hydraulic system.

A comparative consideration can also be made with the frictions in the entire system. The combination of both considerations allows for the study of the influences of force and travel hysteresis.

Vibrations in the Clutch / Operation System

There are more ways to identify pedal vibrations and actuation noises (e.g., eek, whoop, scratch, etc.) than for almost any other phenomenon. This provides a clear indication that the types of excitation and the vibration transmission are numerous for this type of complex system.

Examples for excitations of vibrations in the system clutch and operation are:

- Axial or bending vibrations from the crankshaft and flywheel
- Unperpendicular release bearing
- Vibrations of the engine-transmission assembly
- Alignment between engine and transmission
- Actuation alignment

To understand the entire vibrational system, to separate the various influence variables and to be able to represent the corresponding remedies in both computation and practice, the entire system was set up as a vibration model at LuK.

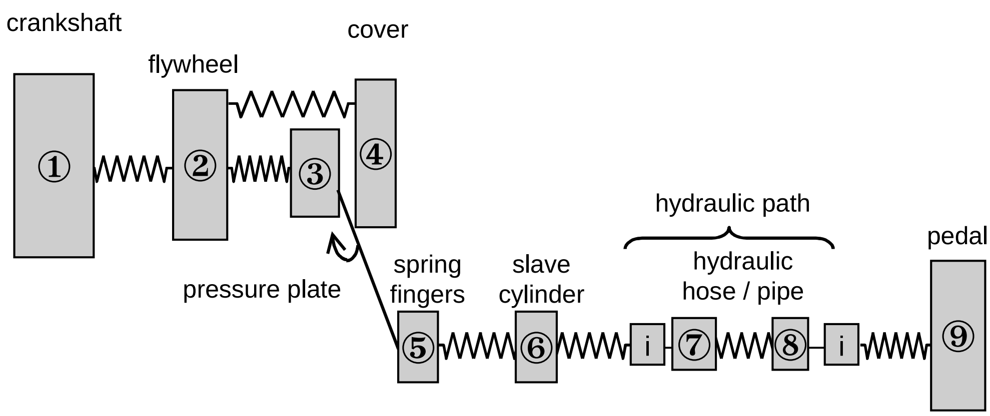

Figure 6: Vibration model

In the following case, this model provides information that is representative of many problems that can be solved in this way.

A strong, high-frequency and pedal travel-dependent actuation noise occurred in one vehicle.

A natural frequency analysis of the vibration model results in a correspondence of the cover natural frequency and the frequency of a standing wave in the fluid column of the hydraulic travel, which leads to good noise transmission in the release system.

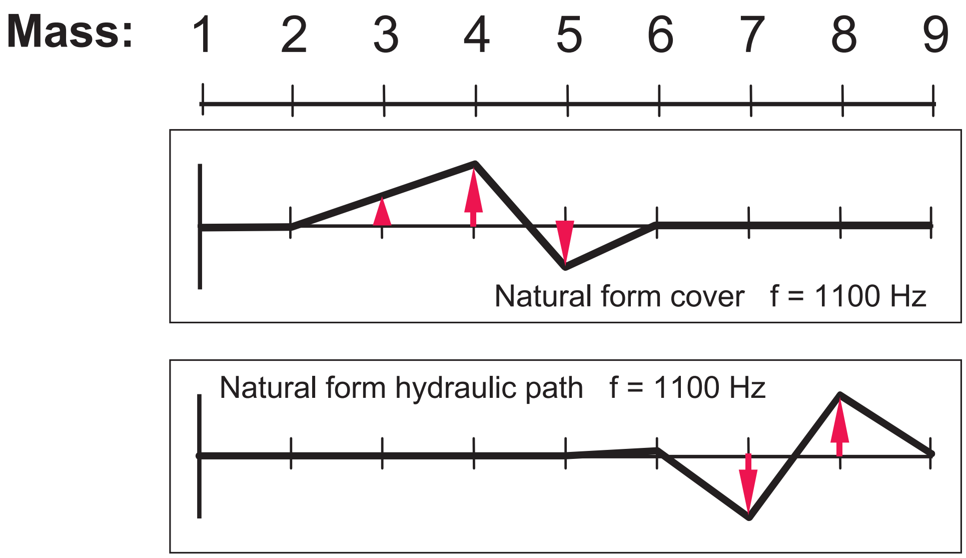

Figure 7: Natural modes of the release system

The natural mode of this standing sound wave in the fluid corresponds for a mechanical system to a string clamped on both ends.

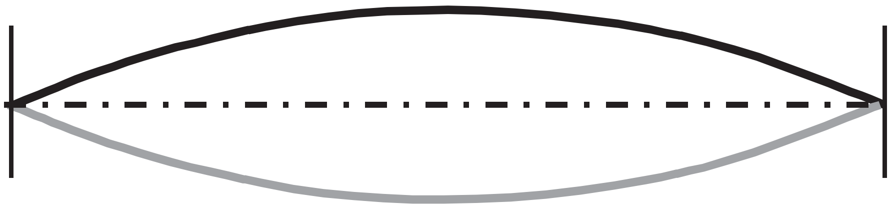

Figure 8: Natural mode of a clamped string

Theoretically, this can be avoided by detuning the two frequencies, thus by changing the cover stiffness or the length of the hydraulic line. In the above case, the steel hydraulic line was lengthened by approx. 20 cm, which is the simplest solution. This solution completely eliminated the actuation noises, without having a negative effect on the elasticity in the release system.

This example illustrates how vibrations, that are excited and transmitted in the clutch and release system, can be described and how improvements can be developed using the simulation program. The critical areas of the individual components were shown. For the pressure plate for example, the simulation program also considers a release travel-dependent natural frequency, because of the surrounding springs. Remedies can be provided depending on the problem by using a “soft connection” for the pressure plate.

The effects of the crankshaft dynamics (axial and bending vibrations) or excitations from out of perpendicular release bearings (“slanted position of the diaphragm spring”) result in reactions on the clutch that are recognized in the timeframe through simulation and thus, can also be prevented. Hence, it is possible to depict the influences of friction and damping. The simulation makes it possible to design the damping elements in the pressure plate as well as to define the friction and damping equipment in the hydraulic or mechanical actuation system.

The “rapid engagement” procedure can also be simulated. In addition to friction and damping, the distribution of the masses and ratios in the release system play a decisive role here.

Clutch Modulation During Driveaway

Changes to the driving profile due to a higher proportion of city driving or traffic jams, or major changes to the entire vehicle, lead to a critical evaluation of the driveaway performance in many vehicles, particularly at idle.

Significant changes to the entire vehicle that stress the clutch and release system layout with regard to modulation are:

- Small capacity, super-charged engines that reach a high maximum torque, but have low idle torque.
- High maximum engine torque's result in high clutch torque's and high release forces.
- Lowering the idle speed as well as reducing the engine-side rotating masses decrease the flywheel energy during driveaway.
- New injection technologies (particularly for diesel engines) change the engine speed stability during driveaway.
- Clutch systems with reduced forces offer potential in assembly standardization, in the area of clutch and release systems with lower release system ratios.
- The introduction of longer axles results in an increase of the effective vehicle mass reduced to the transmission input shaft.

An added difficulty is that vehicles have predominantly been subjectively classified as having either good or poor driveaway performance, because there were no objective parameters and insufficient measurement and simulation options to describe the driveaway characteristics.

Problems during driveaway are generally attributed to the clutch because it connects the "rotating engine mass" and the "standing vehicle mass". The factors influencing driveaway characteristics and how these factors can be measured and evaluated using measurement and simulation, will be illustrated below.

The factors involved in driveaway can be classified as shown in Figure 9:

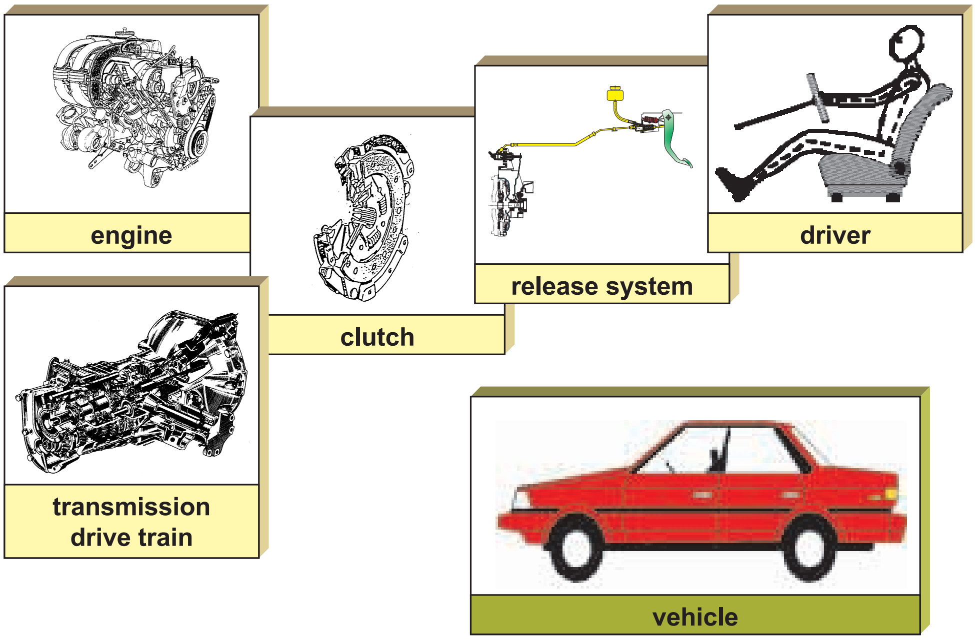

Figure 9 : Influencing parameters during driveaway

In practice, the clutch is almost always proven to be a part of the influencing variables. Various measurement and simulation options were developed at LuK in order to better understand, evaluate and effect positive changes in the individual factors and their interactions.

The driveaway performance of three vehicles will be compared and studied as an example of this systematic procedure. Three different vehicles with similar piston displacement, but with different actuation systems, are used.

To conduct this study, the vehicles **A**, **B** and **C** are broken down into their corresponding sub-systems (clutch, release system, engine, vehicle).

First the clutch is considered. This corresponds to the current “classic” scope of the task of a clutch manufacturer with regard to its options for influencing driveaway characteristics.

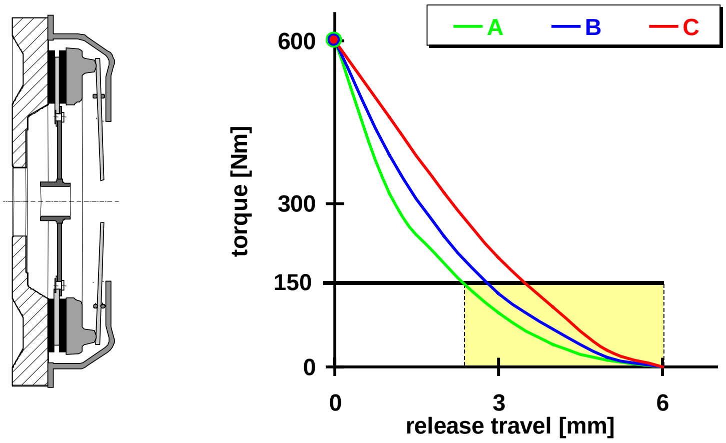

Figure 10 : Clutch torque as a function of the release travel

Figure 10 illustrates the three fundamental clutch torque curves (**A** / **B** / **C**), which all lead to the same maximum clutch torque with the same release travel.

Clutch **C** is shown here as having the steepest characteristic in the torque range (≤ 150 Nm) that is decisive for driveaway. Clutch **A** is shown having the shallowest. Hence, clutch **A** would initially be classified as the easiest to control.

These three clutches were installed and measured in the corresponding vehicles.

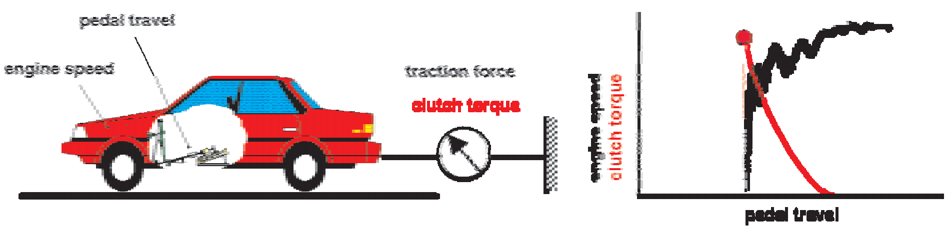

Figure 11 : Determination of characteristic values during engagement to describe the release system

To determine the influence of the actuation system, the clutch is engaged by a spindle unit, which acts upon the clutch pedal; the vehicle is fixed by means of a load cell. This load cell records the torque build-up of the clutch as a function of the pedal travel, which is influenced accordingly by the ratio and the elasticity of the release system. As this measurement is taken, the engine speed is also measured. The engine behavior as well as the level of the stall torque attained when closing the clutch at different speeds while in idle provides information about the quality of the engine and the engine control.

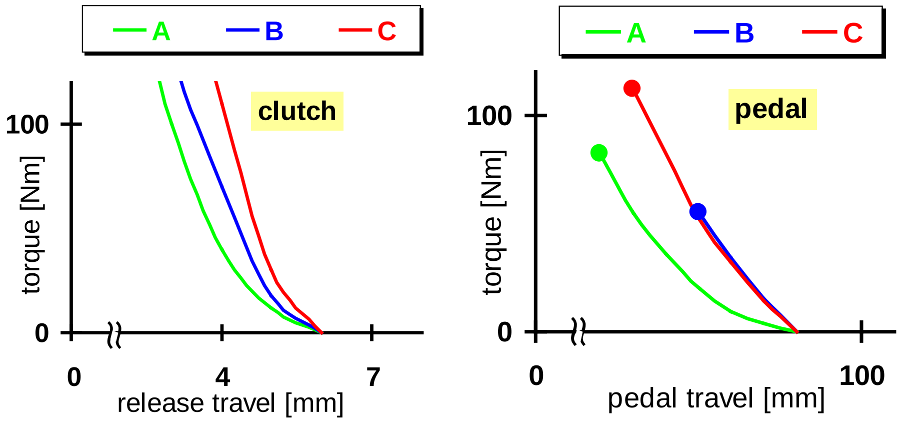

Figure 12: Clutch torque over release travel and pedal travel

Figure 12 (right) illustrates curves **A, B** and **C** of the three different clutches with the applicable release systems in the vehicles. The curves show the modulation travels for the clutch torque on the pedal. All three curves are standardized to the same engagement point, and thus all have the same pedal travel at zero torque. The curve runs over the pedal travel up to the "STALL torque" attainable at idle, at which point the engine stalls. This value is indicated as a bold dot.

System **A** still shows the shallowest torque curve over the pedal travel. The torque curves **B** and **C** are practically identical along the pedal travel due to the higher release system ratio of vehicle **C**.

Hence, systems **B** and **C** now pose identical requirements on the driver and on the engine, although the applicable clutches are laid out very differently along the release travel (see Figure 12, left).

The engine influences result from the STALL torque achieved. Vehicle **C** has the highest engine torque available at idle. Vehicle **B** reaches only half this value, therefore stalls more easily during driveaway.

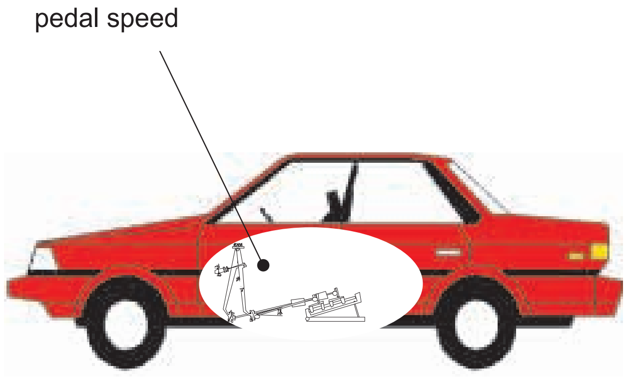

Figure 13 : measurement of driveaway characteristics

This is confirmed by the driveaway measurement (Figure 13), in which the clutch is engaged by a spindle unit connected to the clutch pedal. The vehicle, however, is not fixed. The limit engagement speed [mm/s] determined in this test - at which the vehicle is still able to drive away at idle - provides a characteristic value upon which vehicles could be compared objectively with regard to driveaway performance.

Studies conducted so far on 20 different vehicles at LuK show that a good subjective evaluation (> Rating 6) is achieved after a limit engagement speed of 25 mm/s. At higher limit engagement speeds, the vehicle drives off without any problems.

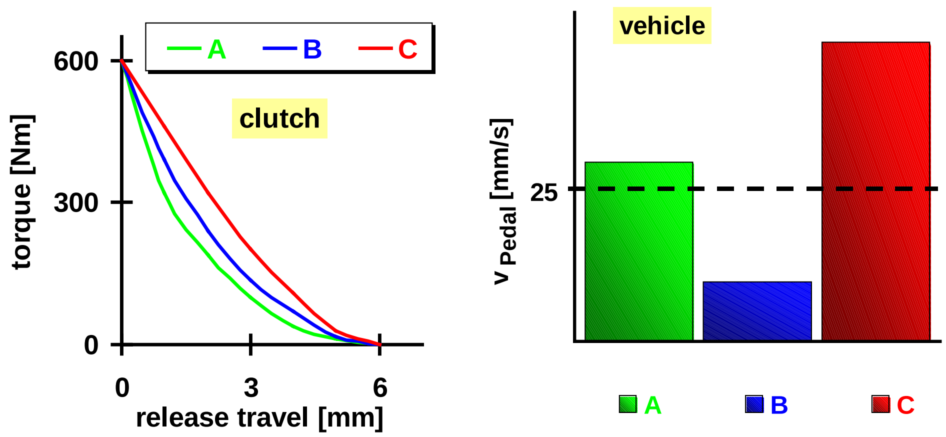

Figure 14: Comparison of the clutch torque, pedal limit speed

Figure 14 illustrates the maximum engagement speeds possible on the clutch pedal of the three comparison vehicles. Vehicle **C** was subjectively rated with Rating 10, vehicle **B** with Rating 3, vehicle **A** with Rating 7. This corresponds to the characteristic value achieved in the driveaway performance test. The result is initially surprising because a better driveaway performance had been expected in vehicle **A** than in vehicle **C**, based on the clutch characteristic curves. This further proves that it is wrong to design the individual components without considering the entire vehicle.

The band width from 5 mm/s to 60 mm/s, i.e., from rating 1 to rating 10, for all of the pedal limit speeds measured so far on different vehicles shows that there is a need for action here.

The information obtained from these measurements is sufficient to systematically compare vehicles and to dispose of the subjective evaluation technique.

In order to replicate the behavior of the real driver and of the real actuation speed on the clutch pedal, basic observations were carried out at LuK. The extent to which ergonomic considerations and the characteristic of the pedal force curve influence the engagement process were studied.

The following experiment comes to mind (Figure 15).

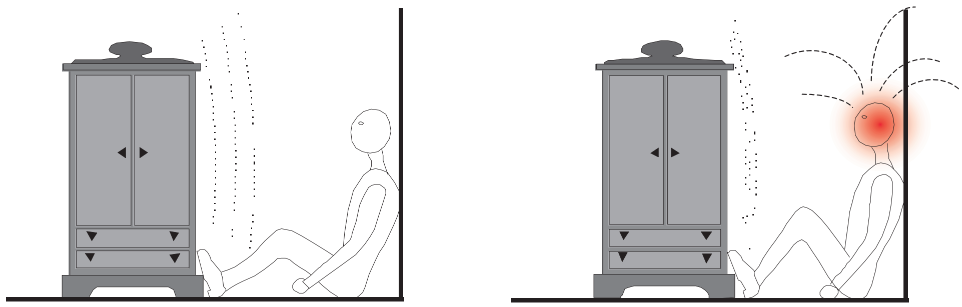

Figure 15 : Simple ergonomic consideration

Although equal displacement forces are required in both cases, in the seated position on the left, subjectively less exertion is required than on the right. From this biomechanical perspective, the “controlled variable” necessary for actuation in the following consideration was not assumed to be pedal force on the foot, but rather the torque at the hip point.

To this extent, the driver’s leg is thus a part of the release system. The weight of the leg acts as a preload on the pedal.

A study on the driver seat position is illustrated in Figure 16. The comparative consideration of two different seat positions shows the extent to which this torque can be influenced by ergonomic considerations for the same pedal force characteristic. Seat position 1 leads to a sharp drop-off in the torque characteristic curve. It is easy to imagine that the driver cannot modulate the pedal with this sharply dropping or rising gradient as well as with a horizontal torque curve.

Therefore, the seat position must also be considered in the future.

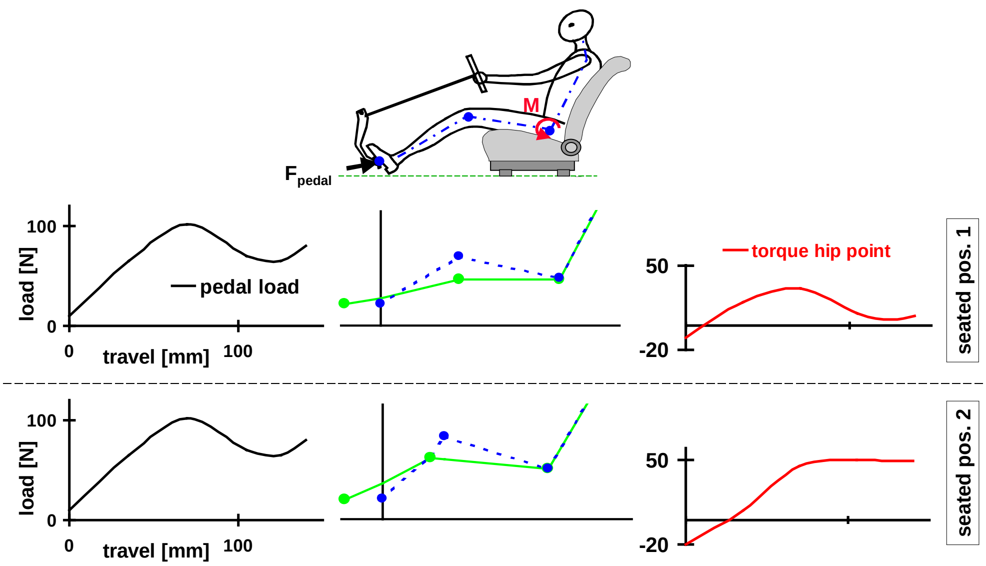

Figure 16 : Simple driver model

All of the influential parameters described for engine speed control via the clutch, the release system and up to the drive train were combined by LuK into a single simulation program. The reliable torsion vibration calculation program was used as the basis. This allowed for an important step to be taken in the simulation of real driveaway processes. With this simulation tool, it is now possible to vary and evaluate each of the factors that influence the driveaway performance listed in Figure 9.

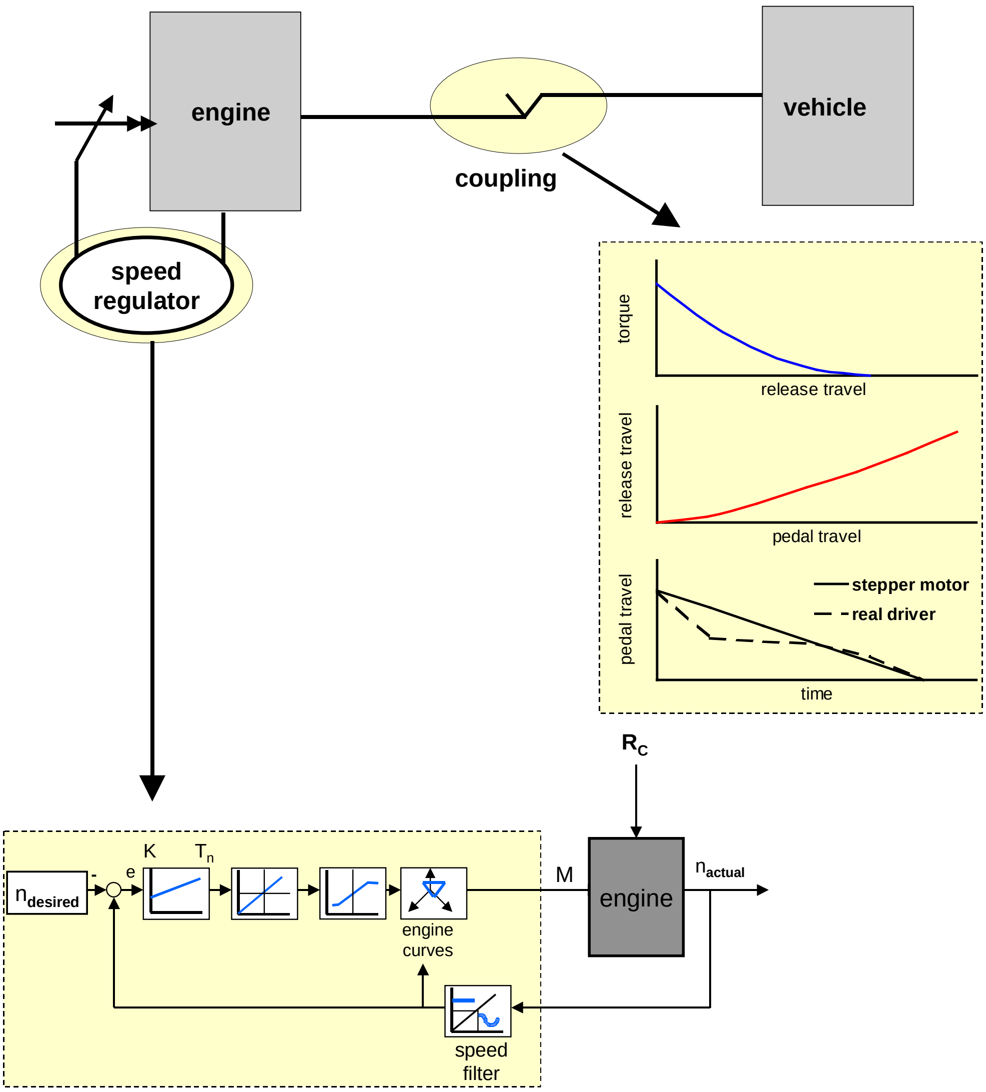

Figure 17 : Simulation model of the driveaway process

Figure 17 illustrates the individual program modules. The friction coupling depicts the interface between the engine and the vehicle. Hence, it represents the function of clutch, release system and driver. Elasticity's, ratios and frictions of the release system are integrated at this point.

The clutch is thus closed by the driver or by a spindle system (see measurement) at the pedal with "pedal travel over time" input. This function can also assume any characteristic, for example, the influence of force and travel hysteresis on the driveaway performance, during reversal of the pedal movement direction, can be realized.

The engine speed control can be achieved in two different ways. The speed can be controlled by means of a PI or PID regulator or the speed can be controlled by means of the real engine map. In this case, the controller is subject to the dependencies on the throttle position, speed and torque that are specified in the map.

Currently, the data to replicate the engine controls in the simulation model on the vehicle are obtained by a simple test. To do this, the reaction of the engine is measured when subjected to defined torque jumps, similar to the tractive force measurement in Figure 11. Furthermore, this allows for a comparative consideration of the engine controls independent of the clutch.

Figure 18 and Figure 19 illustrate two examples. The quality of the engine control can be determined from the engine's reactions to the torque jumps.

Figure 18 shows strong engine speed oscillations after each torque jump. The engine in Figure 19 exhibits short breaks in the speed, however, adjusts immediately thereafter.

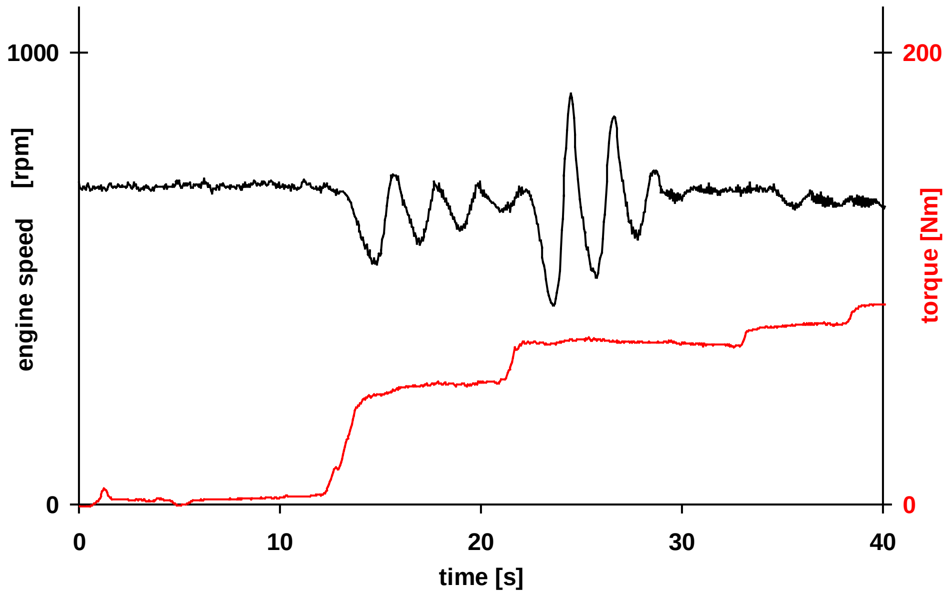

Figure 18 : Identification of an idle controller (Vehicle 1)

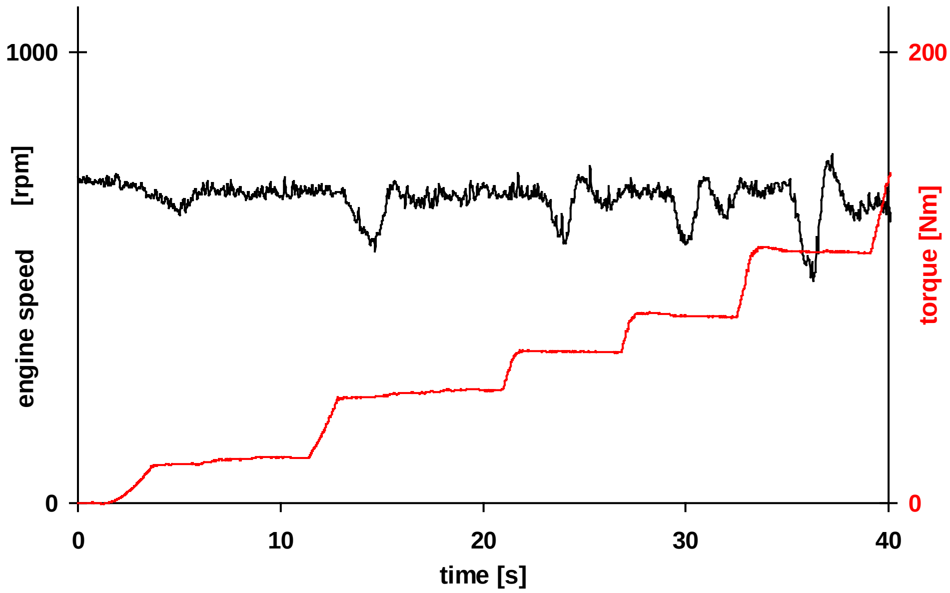

Figure 19 : Identification of an idle controller (Vehicle 2)

Of course, the engine control with all of its special cases can only be mapped and influenced with limited accuracy. However, concrete suggestions on how the driveaway characteristics of a vehicle can be improved can be made based on the simulations.

The other influences from the drive train and release system can be simulated very precisely and can be evaluated with regard to their influence.

Several examples are illustrated in Figure 20 to Figure 22. A real vehicle was used as the basis. Three different engine controls were simulated for the variations.

The limit engagement speed on the clutch pedal (PGEG), which is shown as the y-axis in the bar chart, was evaluated. The limit engagement speed should lie above the limit value of 25 mm/s so that the engine will not stall too easily during engagement.

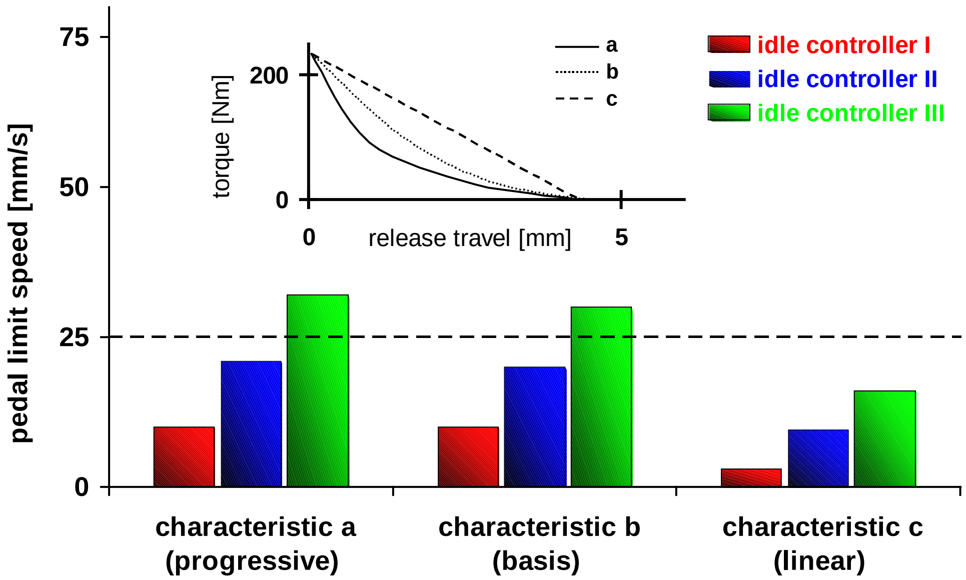

Figure 20: Variation of the clutch torque curve

Figure 20 illustrates the influence of different clutch torque curves. In this case, even the very shallow torque curve in characteristic curve **a** could only have been managed with the control parameters of controller III.

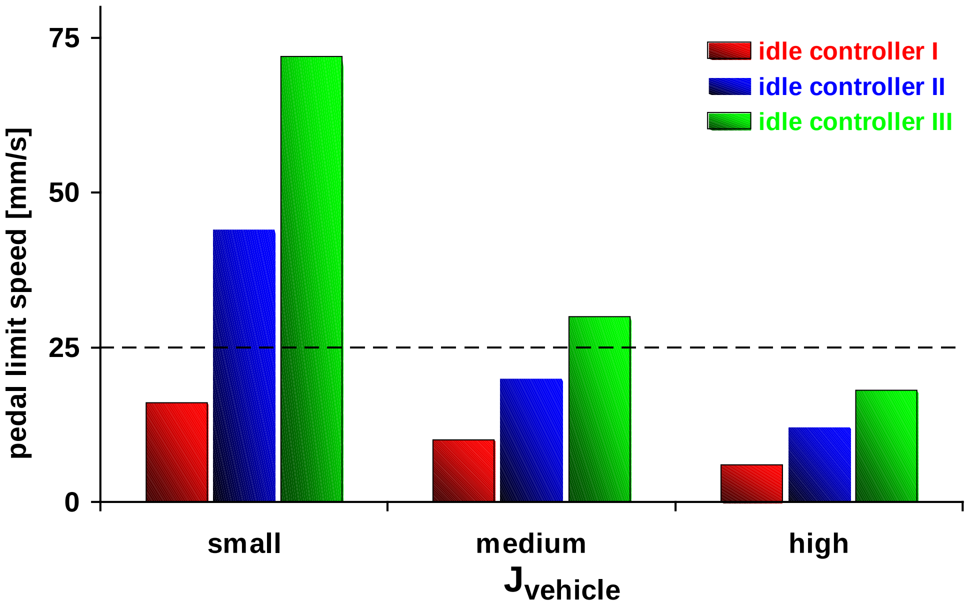

Figure 21: Variation of the reduced vehicle mass

Figure 21 illustrates the influence of the vehicle mass reduced to the transmission input shaft. The translational vehicle mass was converted along with the transmission and differential ratio as well as with the rolling radius of the tires, into a rotary mass. In particular, a “long axle”, which leads to high reduced vehicle mass moments of inertia, leads to driveaway problems here insofar as this was not compensated for by other parameters.

Figure 22 illustrates the influence of the engine side rotating masses on a vehicle’s pedal limit speed. The lower the engine-side rotating mass, the less centrifugal energy available for the driveaway process.

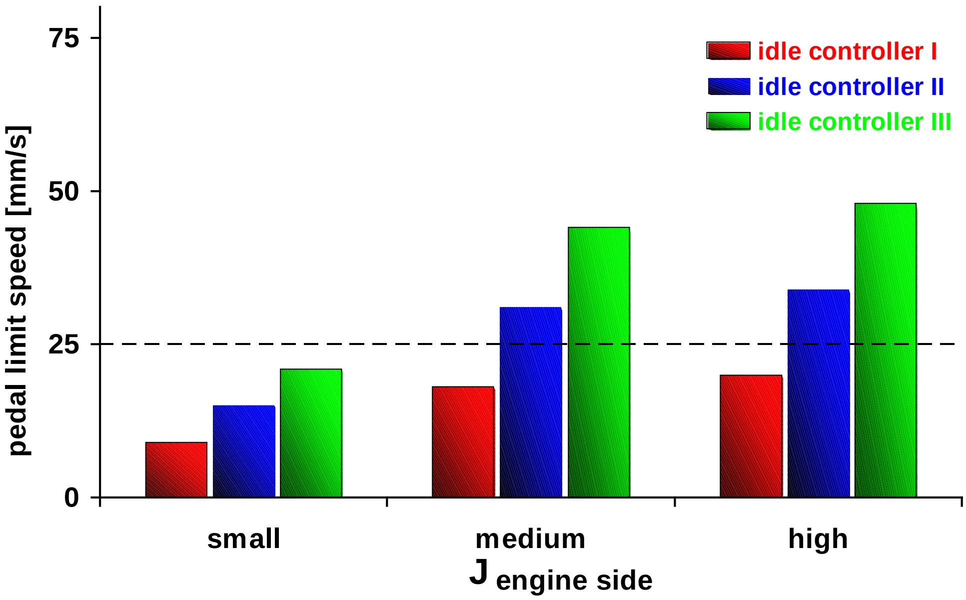

Figure 22: Variation of the engine-side rotating mass

These examples should illustrate that the effect of all of the important parameters were included in the simulation and that predictions can thus be acted upon.

Summary

The experiences in recent years at LuK show that a rigid separation of the clutch and the release system cannot lead to an optimal, technical solution.

Only by carefully analyzing all of the elements involved in the flow of force from the clutch and release mechanism and considering specific vehicle and engine data, is optimal function of the clutch ensured. A system consideration is almost mandatory.

In the future, project management should be arranged for all clutch optimizations and new designs so that all of the elements in the functional chain are tuned properly with one another.

Dual Mass Flywheel

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Dipl.-Ing. **Roland Seebacher**

Dr.-Ing. **Ad Kooy**

Introduction

The first mass-produced dual mass flywheel (DMFW) in automotive history went into production around 1985. A brief historical review (Figure 1) shows the development of the DMFW. In the beginning, unlubricated dampers were used, which had heavy springs located far to the outside that exhibited wear problems. Around 1987, the first grease-lubricated DMFWs were used and service life was no longer an issue.

The introduction of the arc spring damper was a breakthrough for the DMFW in 1989 that solved almost all of the DMFW resonance problems at once. In addition, costs were continually reduced [1-4].Initially, the primary flywheel mass was a casting or forged steel. Later, the metal-forming specialists at LuK were successful in forming all of the parts except for the secondary flywheel mass from formed sheet metal parts. To increase the primary mass moment of inertia, folded masses were developed from sheet metal (1995). This formed the basis for broad usage of the DMFW. This intensive detail work paid off with a large increase in DMFW production (Figure 2).

With an estimated production volume of approximately 2 million DMFWs for 1998, the noise and comfort behaviour of every fifth car with manual transmission in Europe will be improved by the DMFW. Figure 3 illustrates the allocation according to engine displacement and gasoline / diesel engine class. It is apparent that engines with more than 2.0 liter engine displacement, particularly gasoline engines, are predominantly equipped with DMFW. The use of DMFW for mid-size engines only began a few years ago. There are currently only a few projects for engine displacement less than 1.6 liter.

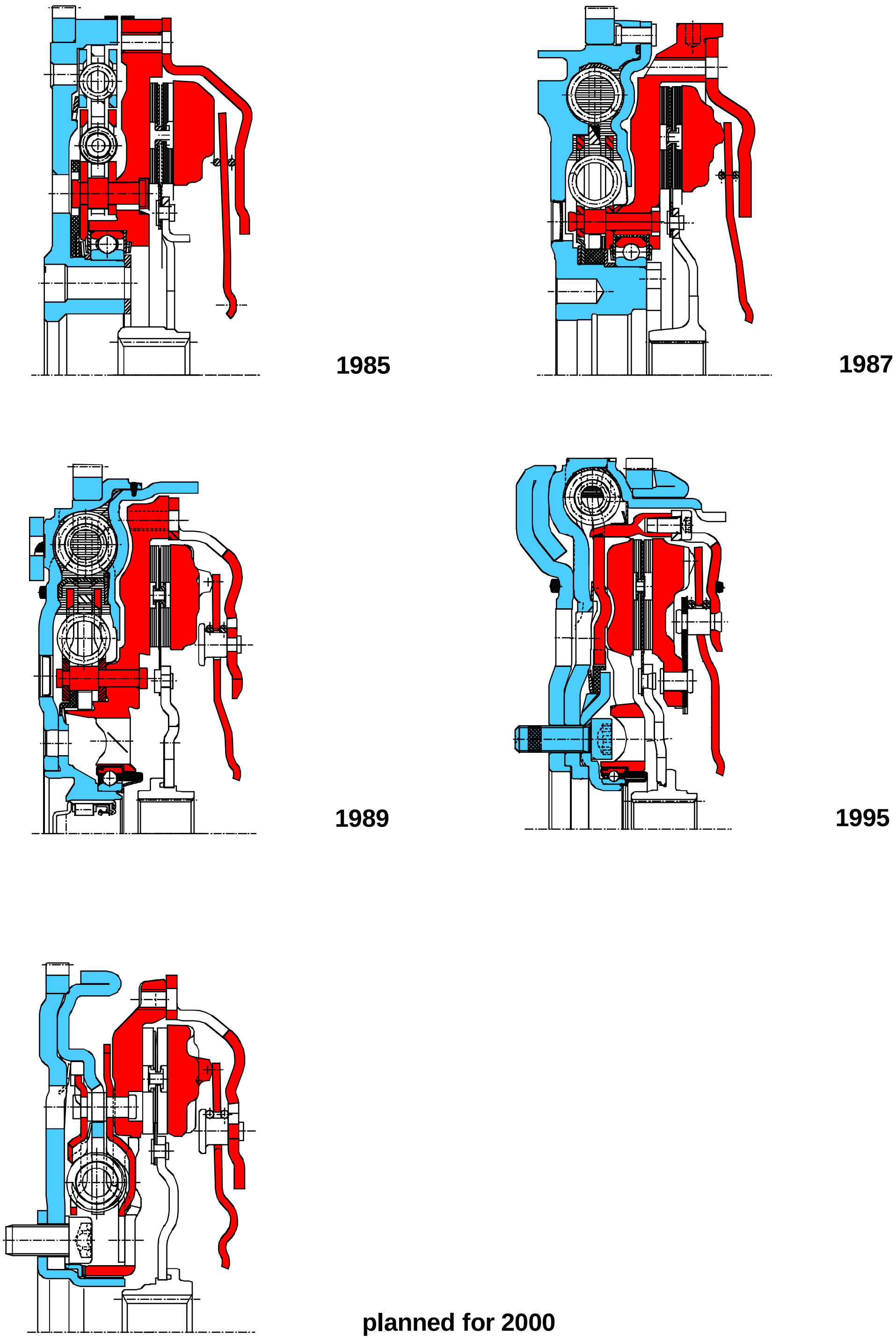


Figure 1: Development history of the DMFW

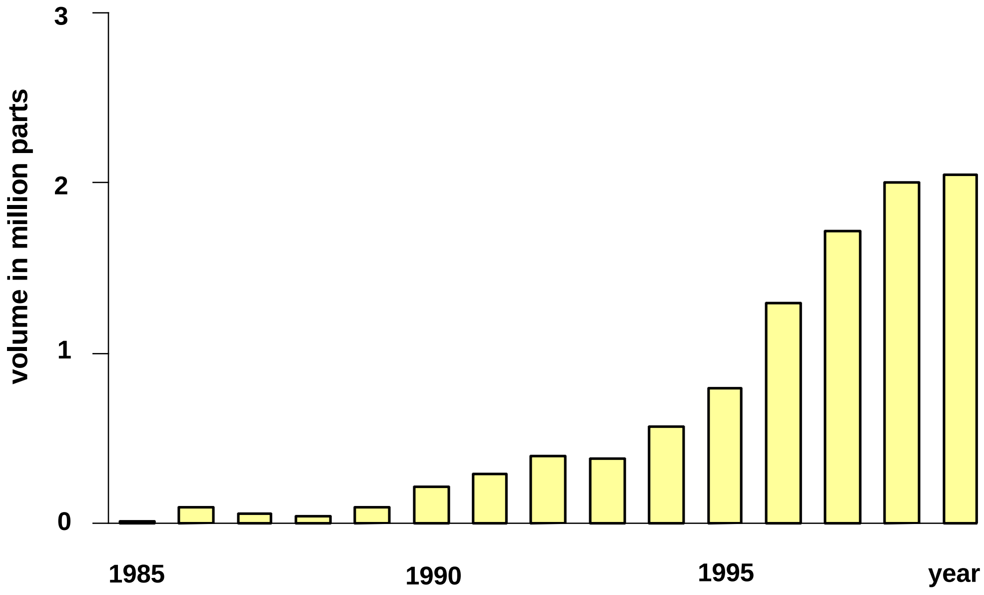


Figure 2: Development of DMFW production for German car manufacturers

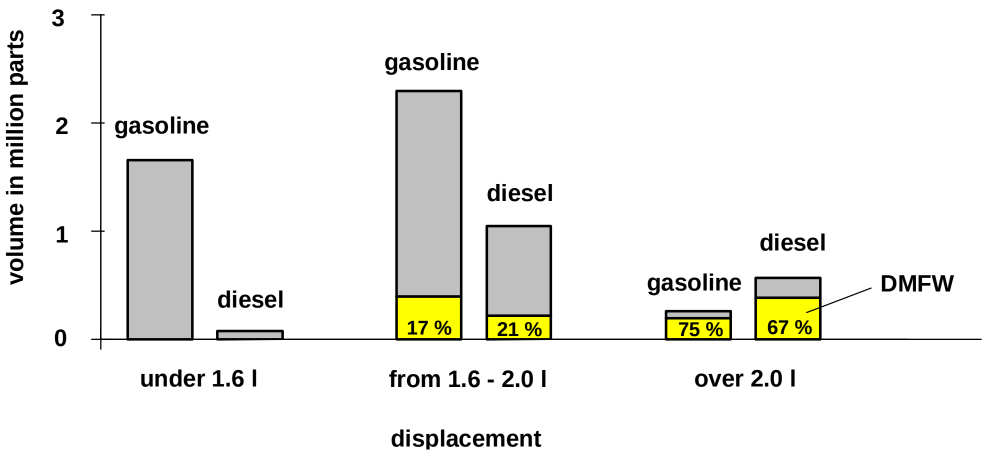


Figure 3: Portion of vehicles with DMFW for different engine classes (1997)

LuK expects that in a few years the saturation will be similar for at least mid-size engines as it is today for powerful engines because a DMFW shows its advantages in all vehicles. It has not been used thus far in smaller cars due to its high costs.

Hence, a critical point in the development of DMFW is reducing costs. A later part of this presentation will cover this issue.

Advantages of the DMFW

Although not everyone wants the DMFW due to the costs, the achievable improvements are so clear that it is being used extensively in large vehicles. The most important advantages will be outlined below.

Isolation from Torsional Vibrations

The primary feature of the DMFW is the almost complete isolation of torsional vibrations. This has been discussed extensively in earlier presentations and will only be summarized here.

Figure 4 illustrates the angle accelerations at the transmission input for a conventional system with a torsion damper in the clutch disc (left) in comparison to a DMFW (right). With the torsion damper in the clutch disc, there is no significant vibration isolation achieved at low speeds. Resonance can be avoided by selecting appropriate damping.

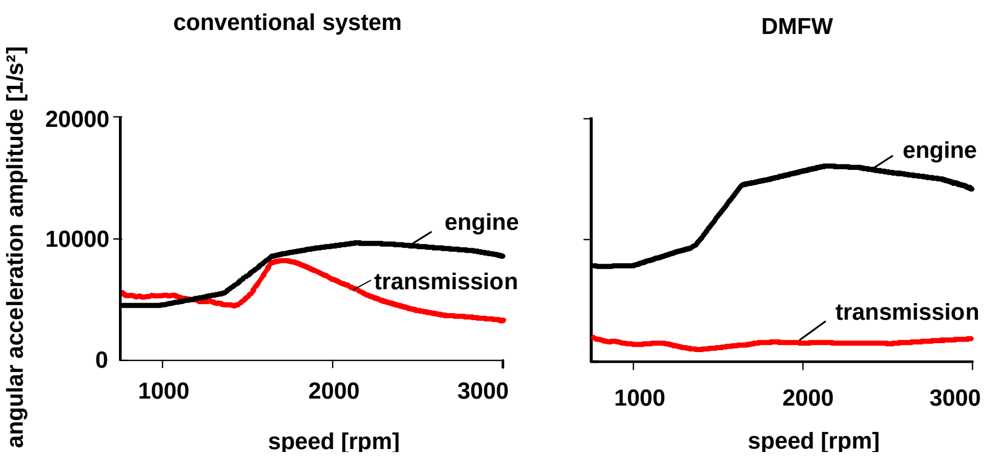


Figure 4: Comparison of vibration isolation of a conventional system to a dual mass flywheel

In contrast, the DMFW almost completely filters out the engine irregularity. Resonance generally no longer occurs in the driving range. Gear rattle no longer occurs due to the almost uniform operation of the secondary flywheel side and thus also of the transmission input shaft. Annoying droning can also be almost completely eliminated.

The irregularity of the engine itself becomes greater with DMFW because the primary flywheel mass is lower than the conventional flywheel mass with a clutch. Therefore, the accessory drive must occasionally be retuned. The smaller primary flywheel mass also has advantages, as will be presented later.

Good vibration isolation, particularly during low-speed driving, often leads to low-consumption operation, which saves fuel due to the predominantly low engine speeds used. Many modern engines with a relatively flat torque curve favor this consumption-reducing operation.

Transmission Relief

Another positive effect results from the transmission relief. The drive train and hence also the transmission are relieved of stress due to the elimination of engine irregularities.

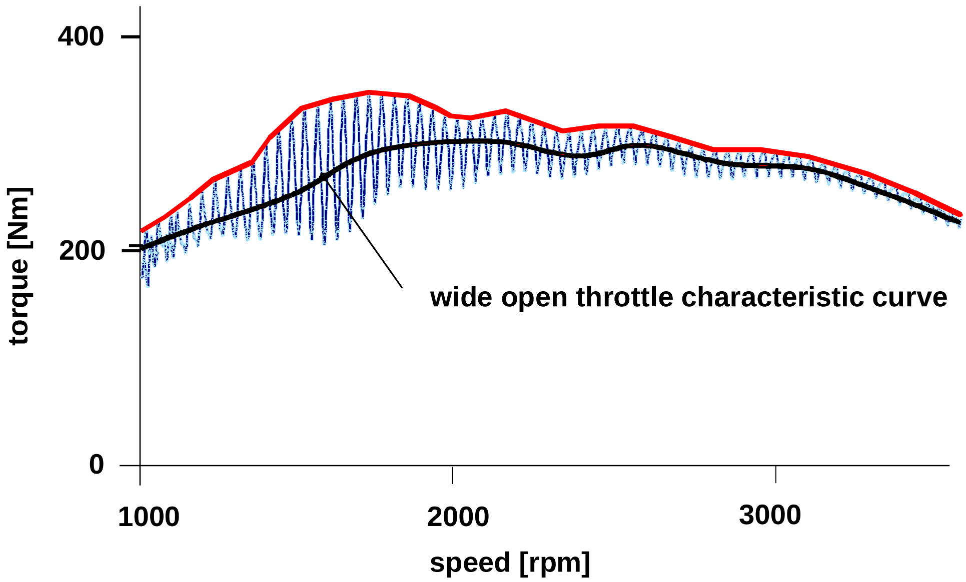


Figure 5: Increase of the actual effective torque in the transmission due to the engine irregularity

Figure 5 illustrates the wide open throttle characteristic curve of a typical diesel engine. For a conventional drive train, the additional dynamic torques as a result of the irregularity are superimposed. Depending on the speed, they can generate more than 10 % additional load.

The DMFW almost completely eliminates the additional high-frequency torques. Since the transmission is relieved, a higher static torque can be transferred, particularly with diesel engines with DMFW (figure 6).

	gasoline	diesel
conventional	100 %	100 %
DMFW	105 %	110 %

Figure 6: Increase of the transmission load capacity when using a DMFW. The load capacity for the conventional drive train is assumed as 100 % for both gasoline and diesel vehicles.

Crankshaft Relief

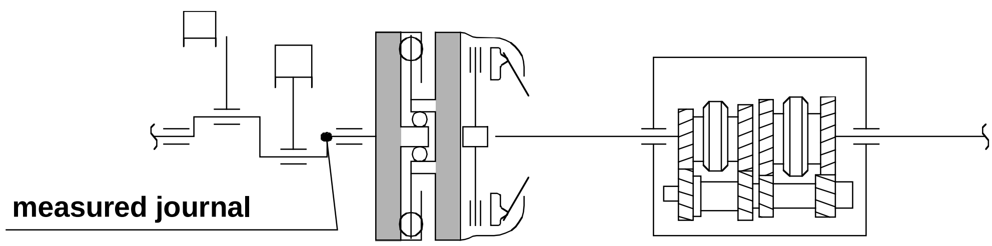
The DMFW permanently alters the vibration system of the crankshaft. In the conventional system, the heavy flywheel including the clutch is rigidly connected with the crankshaft. The large inertia of the flywheel generates high reaction forces on the crankshaft.

The DMFW system behaves more favorably because the secondary flywheel mass can be disregarded for the bending load. It is only very loosely connected via the torsion damper as well as via the roller bearing to the primary flywheel mass and therefore generates practically no reactions.

The primary flywheel mass is much lighter than a conventional flywheel and is also elastic, like a flexplate for a torque converter.

Inherent bending and torsion resonance forms change with the DMFW in comparison to a conventional system. The crankshaft is mostly relieved.

Figure 7 illustrates a measured example. Both torsion and bending vibrations are lower with the DMFW. In individual instances, it must be decided whether the crankshaft damper can be omitted or if a simpler material can be used for the crankshaft, such as a casting.



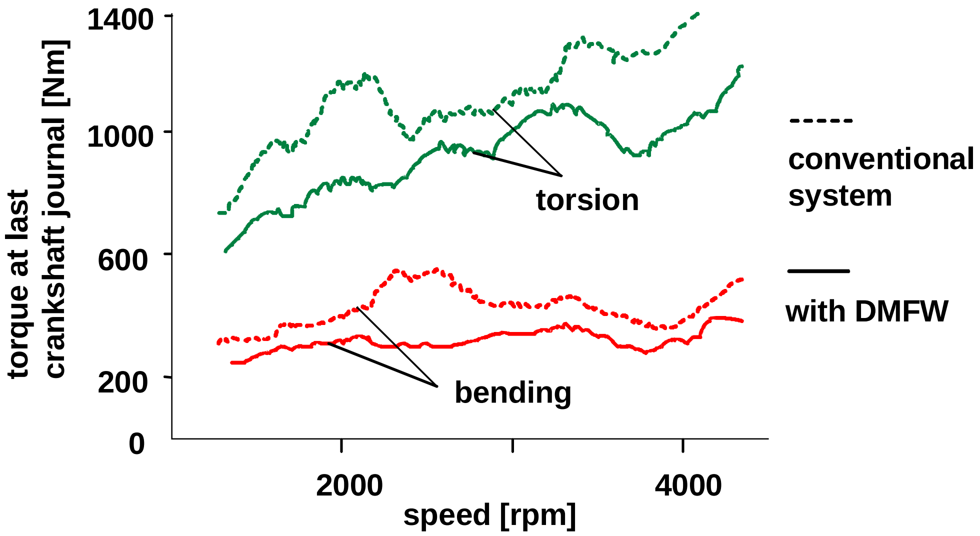


Figure 7: Reduction of the torsion and bending vibrations in the crankshaft using DMFW

LuK recommends that these opportunities for optimization be used in further vehicle developments. This could generate considerable savings. LuK is convinced that there are no additional costs from the DMFW if the secondary effects are taken into consideration.

Warranty

One of these secondary effects is the warranty. The DMFW was designed in the beginning to last the entire life of the engine. The replacement parts deliveries for the DMFW are actually few. Hence, the DMFW is a fully developed component for automotive drive trains.

Figure 8a illustrates the field complaints for a vehicle with a conventional drive train. Apparent is the disproportionately high number of complaints in the clutch area for which the clutch is not the actual cause. This is attributed to the fact that frequently, clutch discs are replaced along with the entire clutch because the customer complains about transmission rattle. Hence, the garage, which has no solution, replaces the entire system to pacify the customer. Generally, the replacement was not successful. Therefore, clutch discs were sometimes changed multiple times. Since not only the costs for the replaced parts, but also the numerous high disassembly costs were often covered at the company's expense, there were exorbitant warranty costs, which transferred to the total production and partially added to the costs for the new clutch parts.

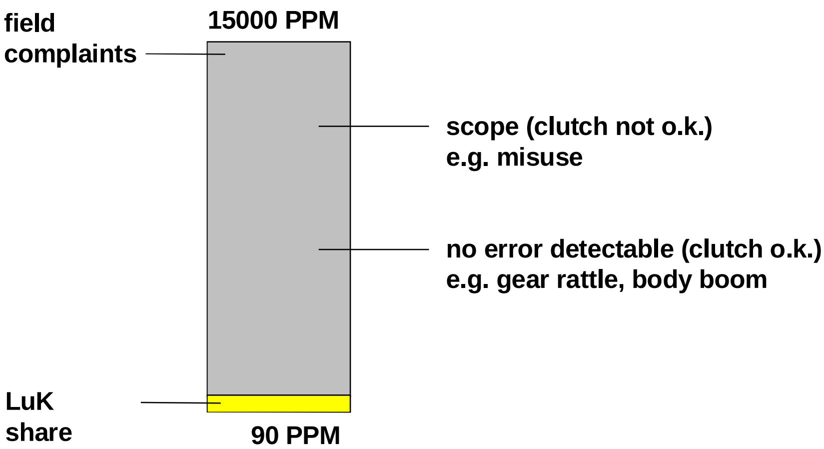


Figure 8a: Field complaints for a vehicle with a conventional drive train

The DMFW has put an end to this custom (Figure 8b). The complaints of this type have dropped so significantly that attention can finally be given to the actual cases of damage. Likewise, on site studies can be conducted instead.

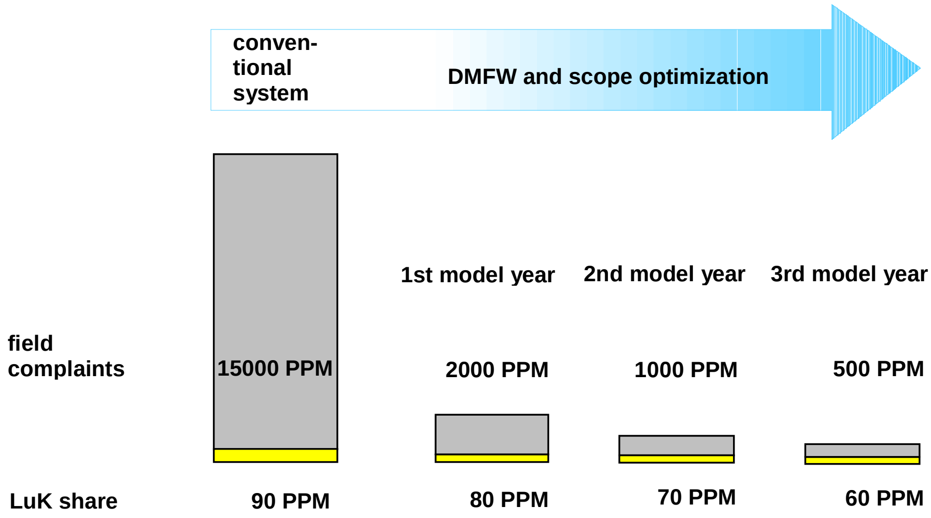


Figure 8b: Improvement in field complaints after using a DMFW

A few manufacturers are already considering this cost effect in the economic calculation if the issue is whether or not a DMFW should be used.

The DMFW is a mature product, but there is still further development potential.

Two aspects shall be discussed below.

Engine Start

The problem of resonance breakthrough when starting the engine was a primary issue from the very beginning of DMFW development. The good vibration isolation of the DMFW during driving operations was achieved in that the resonance frequency was shifted into the range below idle speed by the large secondary flywheel mass.

With each start of the engine, however, it must pass through the resonance frequency. This can lead to torques that are too high due to large inertias. The development of the DMFW was therefore characterized by a constant battle against resonance amplitudes.

It is known that resonance amplitudes are greater the higher the excitation from the engine. Hence, diesel engines with only four or even three cylinders place the highest demands on a DMFW. Every type of damping, such as basic friction, load friction devices, and arc spring friction has a favorable effect. Since these damping factors can diminish the isolation in varying degrees, naturally there are limits.

We have, however, received significant assistance because many newer, electronically controlled engines have an improved starting behaviour. The starting torque of the engine has turned out to be a determining factor that is significant for the formation of resonance. That is the torque with which the engine accelerates from the starter speed. The faster the resonance speed is passed through, the less the inertias can begin to vibrate.

Figure 9 illustrates simulations of a poor starting behaviour as speed over time.

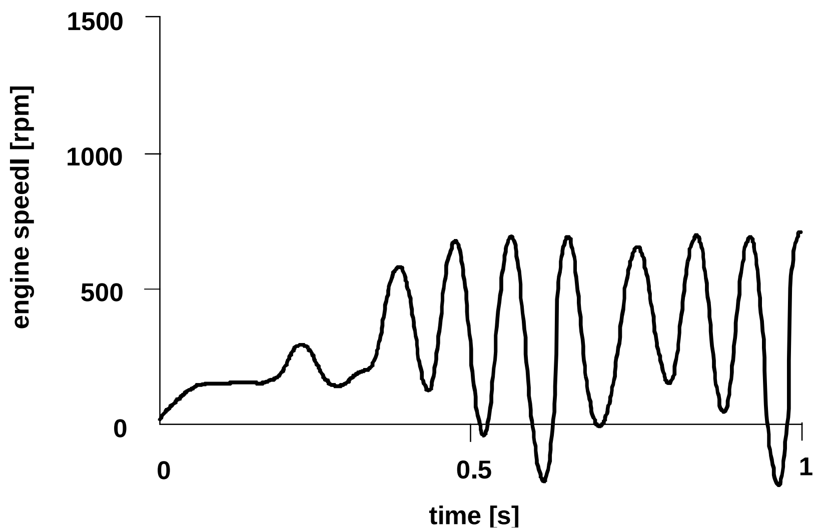


Figure 9: Poor starting behaviour (suspended start)

The cases (as in Figure 9) in which the engine remained in the resonance for a longer period of time or that did not even rev up on their own power are all critical. This is always the case if the engine power at starting rpm is so low that the entire energy is sapped by the highly vibrating system and there is no energy left over for revving up. This condition is also designated as suspended start and must absolutely be avoided with the DMFW because the long-lasting high amplitudes can cause mechanical damage to the components.

Figure 10a illustrates several start simulation calculations compiled in a matrix. The starting torque increases toward the top and the torque amplitude increases toward the right.

The matrix clearly shows how the engine starts well at higher starting torques and even manages for highly irregular engines.

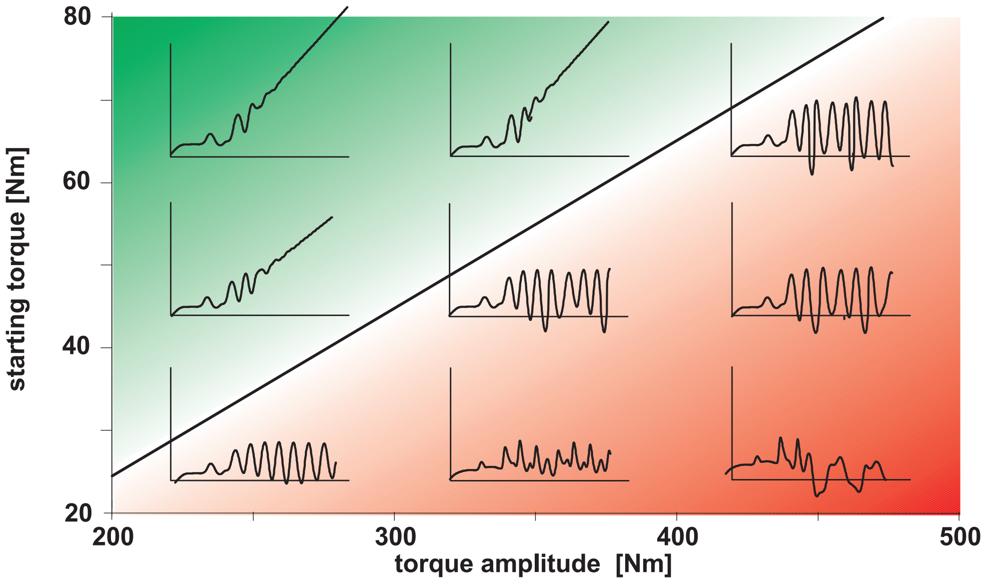


Figure 10a: Influence of torque amplitude (irregularity) and starting torque on the starting behaviour

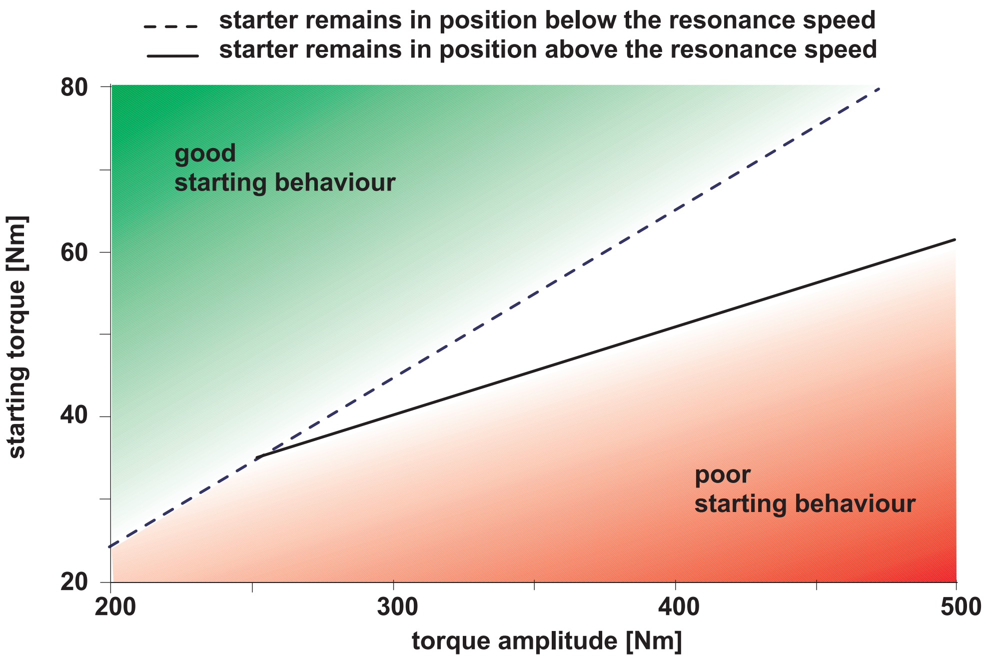


Figure 10b: Border lines for a good (upper left) and a poor (lower right) starting behaviour

Between this good starting behaviour and an unacceptable starting behaviour, there is a diagonal separation line, which is repeated again in Figure 10b. The range that is safe for starting lies above the separation line.

If the starter remains in position above the resonance speed, then the starter is prevented from immediately going out of position again by briefly tipping the ignition key. This results in a still more favorable condition. The limit between a good and a bad starting behaviour shifts downward to smaller starting torques. The large starter inertia (reduced on the crankshaft) reduces the irregularity of the engine.

Many modern engines exhibit a starting torque of 70 - 80 Nm, whereas only approximately 40 Nm were customary earlier. Therefore, current DMFW concepts also work without problems for many three cylinder engines although these problems seem critical from the viewpoint of irregularity.

The starting behaviour can be improved by the measures cited in Figure 11.

- **High engine starting torque**
- **Starter up above resonance speed left in position**
- **High starter speed**
- **Damping (friction hysteresis in the DMFW)**
- **High primary flywheel mass**
- **Small secondary flywheel mass**
- **Flat spring rate of the rotation damper**

Figure 11: Measures to improve the starting behaviour

In the early years of DMFW development, the high load of the components from the resonance breakthrough was a primary issue. Since the components had large dimensions in comparison to a conventional clutch disc, the flange for example, another cause for excess torque was recognized relatively late. Only as the occurrences of resonance were gradually improved and when attempts were made to design the components somewhat smaller for cost reasons was it discovered that a sudden load generated similarly high peak torques.

When the clutch was engaged very quickly, impacts occurred if there was a great difference between the speeds of the engine and the transmission shaft. These types of quick engagements occur during very sporty, fast shifting, but also during incorrect operation, such as slipping from the clutch pedal.

The result is illustrated in several phases in Figure 12. The rotational movement of the drive train is depicted as a linear model to obtain a better overview.

Assuming that the two flywheel masses of the DMFW, which are coupled together via the DMFW torsion damper, move toward the right at a high speed and the remaining drive train stands still, the clutch is closed suddenly. The secondary flywheel mass is thus quickly slowed, while the primary flywheel mass is only slowed later because of the very weak torsion damper. Therefore, a relative movement occurs between the two flywheel masses that can become so large that the masses can impact upon one another with high speed. This can lead to very high peak torques.

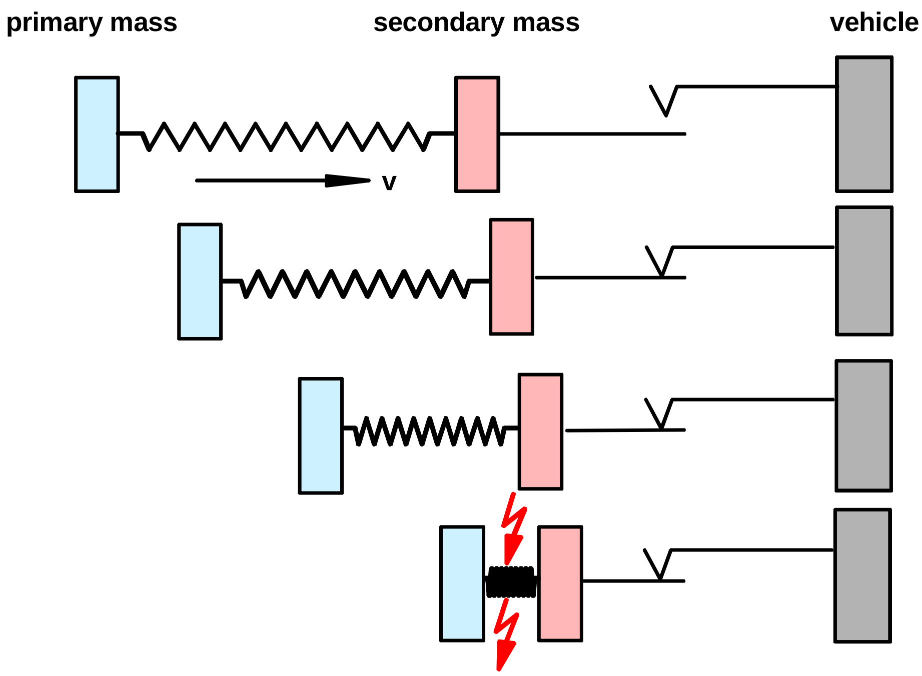


Figure 12: Impact load after quick engagement

Figure 13 illustrates the torques occurring between the flywheel masses immediately after the clutch closes as it would occur for an ideal torsion damper with a very long characteristic curve without impact. These can be more than double the engine torque depending on the distribution of the mass.

Typically, the characteristic curve of a DMFW torsion damper ends at approximately 1.3 times the engine torque. The damper then blocks and an impact occurs that reaches up to 20 times the engine torque.

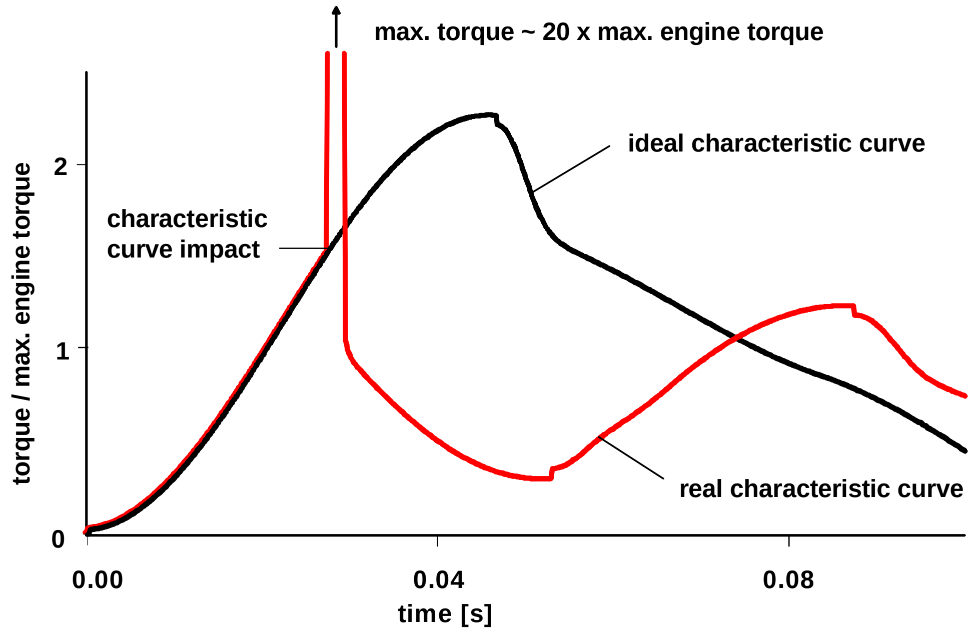


Figure 13: Torque curve between the primary and the secondary flywheel masses after quick engagement for ideal, infinitely long characteristic curve, as well as for the real characteristic curve with an impact torque of approximately 1.3 times the engine torque

Figure 14 illustrates the influence of the engagement time and the impact torque on the peak torques. The engagement time was the parameter varied. The figure illustrates that the peak torques are highly dependent on these parameters. During slow engagement and/or high impact torques, impacts are practically avoided. Hence, the goal is to lengthen the engagement time, for example, by installing a valve in the hydraulic release system.

A peak-torque-limiter is suited for this purpose. It can greatly reduce the impact torques by acting as a relief valve.

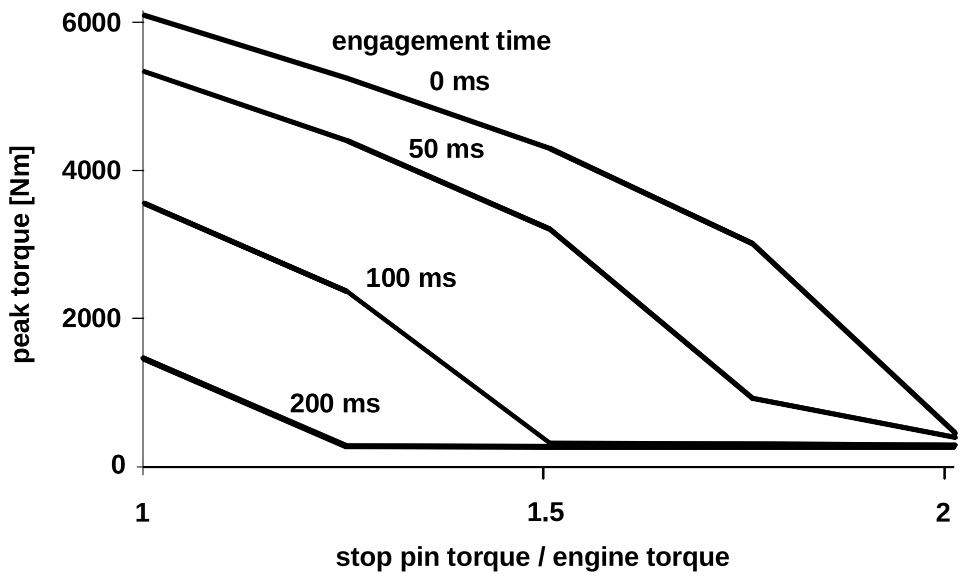


Figure 14: Influence of the impact torque and the engagement time on the peak torques

	time [ms]
• **Mechanical release system with steel pedal**	**15 - 20**
• **Mechanical release system with plastic pedal**	**3 - 7**
• **Hydraulic release system warm**	**30 - 70**
• **Hydraulic release system cold**	**400 - 1000**
• **Hydraulic release system warm, with damping**	**100 - 250**

Figure 15: Typical engagement times for mechanical and hydraulic release systems

Figure 15 illustrates typical minimum engagement times during rapid engagement. Mechanical release systems can engage without slowing, particularly if they have a light plastic pedal. Even a heavier steel pedal reduces the peak torques notably. The favorable, minimum – i.e., long – engagement times are produced by the hydraulic release systems.

If the peak torques cannot be limited over the engagement time, other measures must be used. Figure 16 lists the known measures. A torque limiter, which is connected in series to the DMFW damper, has proven the most effective.

- **Stop pin torque high**
- **Peak torque limiter damping in release system**
- **Spring rate high (shorten angle)**
- **Torque limiter**
- **Reduce clutch torque**
- **Automated clutch instead of conventional clutch**

Figure 16: Measures to reduce peak torques during rapid engagement

New Generations

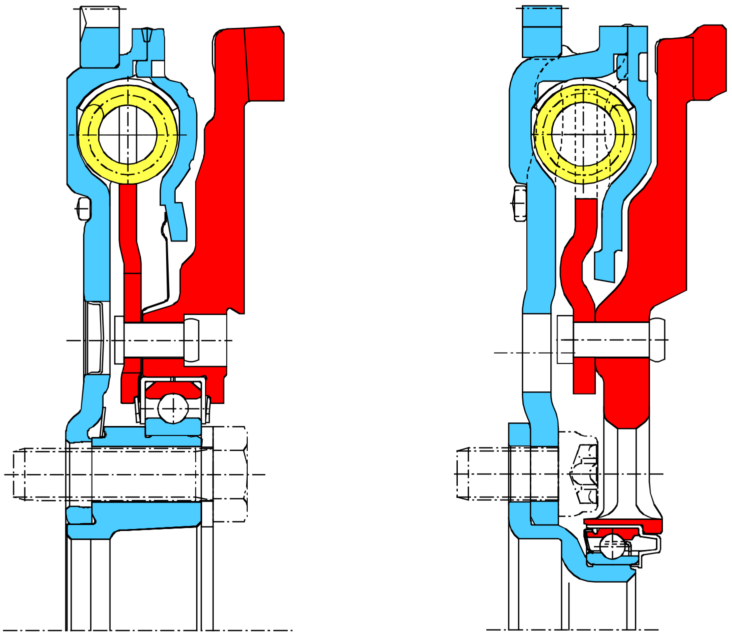
In the beginning, it was indicated that a DMFW could result in significant noise and comfort improvements in all vehicles. The apparent additional costs have prevented broad application of DMFW thus far, particularly for smaller vehicles, because the numerous secondary advantages were not yet considered.

Therefore, cost reductions have been the focus of development in recent years. The most important of these reductions will be covered below.

General Cost Reductions

The metal-forming process for the sheet metal parts was improved, leading to machining being rendered almost unnecessary in newer designs. In addition, other cost reductions were achieved through FEA calculations and optimum material selection.

The introduction of the smaller ball bearing was difficult, but in the meantime has proven itself worthwhile in production (Figure 17). There is no intermediate size between the large and the small bearing due to the bolt hole configuration of the crankshaft (either inside or outside of the bearing). This resulted in a large step, which was difficult for many customers.



series DMFW **reduced cost DMFW**

Figure 17: Reduced-cost DMFW with small bearing

In addition, modular construction systems were developed for different customers for which only small modifications to individual DMFW components, such as arc springs or friction control devices, had to be made between the individual engine sizes.

These types of modular construction solutions also require the vehicle manufacturer's assistance, who must also undertake standardization measures, for example with regard to ring gear position.

Bushings

Another cost reduction could be accomplished with bushings (Figure 18). It seemed indispensable to arrange this bushing inside the bolt hole configuration of the crankshaft. When the clutch releases, the entire release force must be supported by the bushing. In connection with the large friction radius, the friction moment was too high, which impaired the isolation. Therefore, LuK recommended that the bushing must be designed to the smallest diameter.

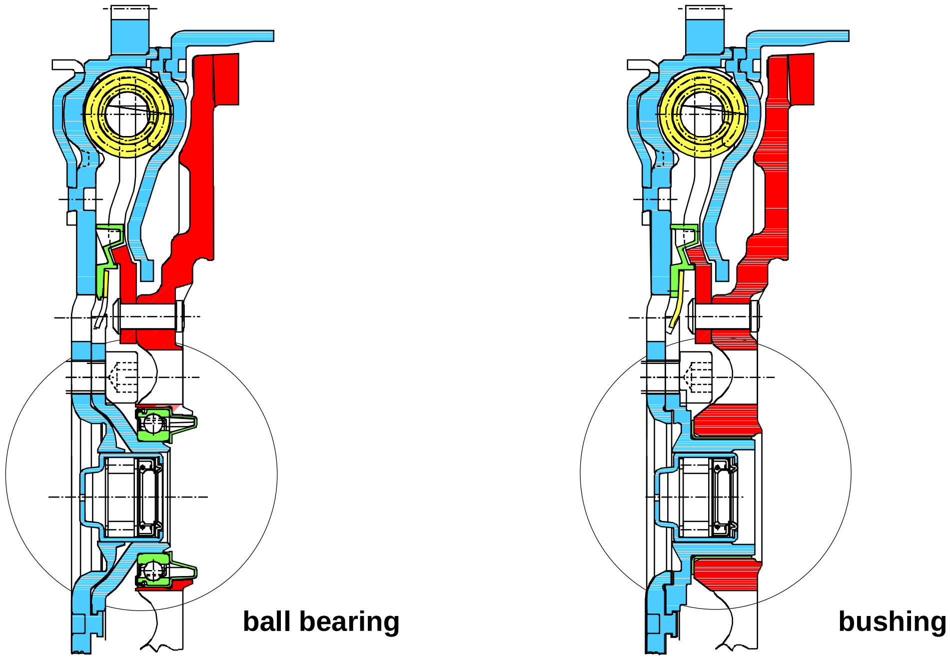


Figure 18: DMFW with bushing

Tests conducted thus far with various bushing designs are promising and a satisfactory service life is expected with sufficient centering accuracy.

DMFW with Dry Damper

In the initial DMFW development, efforts were made to design the torsion damper similar to those in clutch discs. Since the DMFW torsion damper exhibits substantially better vibration isolation than a torsion damper on a clutch disc, the springs in the DMFW had to embody a larger relative vibration angle. The higher wear on the spring guides associated with this change required a switch to grease-lubricated dampers.

Due to the related costs for lubrication for the grease, the seal, etc., LuK conducted a new study to develop a dry-running damper for the DMFW.

We still are not able to say with certainty that the service life will be achieved. But there is reason to believe that our chances are better now than in 1985. LuK has better theoretical and technical testing facilities to analyze the arrangements that are created and to introduce measures against wear.

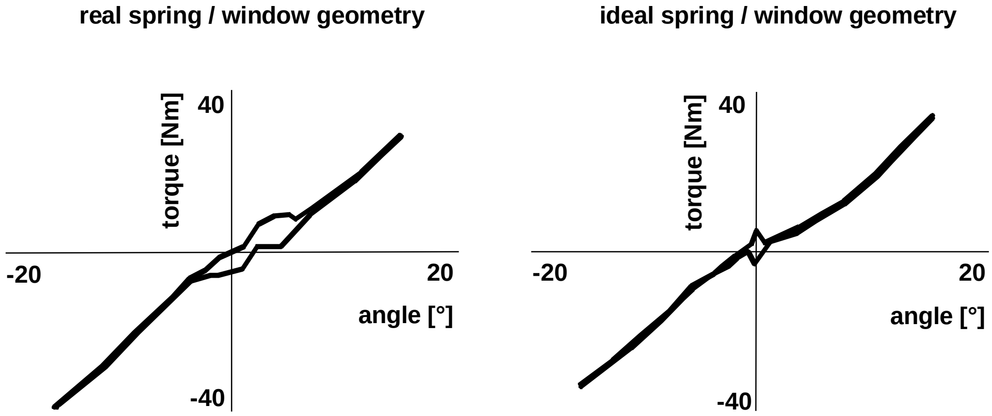


Figure 19: Hysteresis partial loop during torque change

By optimally designing the spring guide, i.e., of windows and spring end coils, the friction work and thus the wear can be reduced considerably, for example, as shown in Figure 19.

Another improvement is achieved if the springs are configured on as small a diameter as possible to keep the centrifugal force low. Based on experience from the early development of the DMFW, a sufficient service life can be expected from the dry torsion damper in the DMFW. The DMFW design is simplified significantly by omitting the grease lubrication.

If all of the savings potentials are combined, then a DMFW results as illustrated in Figure 20. The inner coil springs no longer permit the customary flat characteristic curve. However, inspections in several vehicles have shown that these versions of the DMFW for four and six cylinder gasoline engines exhibit good vibration isolation. The realizable spring rate does not seem to be adequate for four cylinder diesel engines. The dry DMFW in this form is not currently possible for this type of engine.

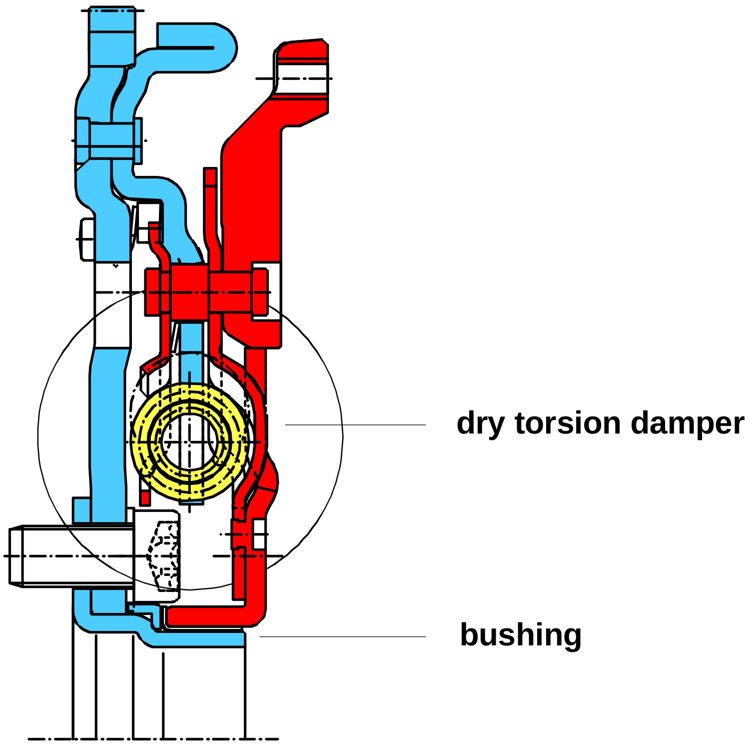


Figure 20: Future DMFW

Alternative Possibilities for Elimination of Torsional Vibrations in the Drive Train

Alternatives to the DMFW are constantly being sought – even at LuK.

The torsional vibrations can be filtered out, for example, via a slip clutch. This does not, however, achieve the vibration isolation of the DMFW, as Figure 21 illustrates. This is explained in the presentation on the automation of clutches [5].

Another theoretically interesting possibility that has recently met with great response in the popular scientific press shall now be explored [7, 8].

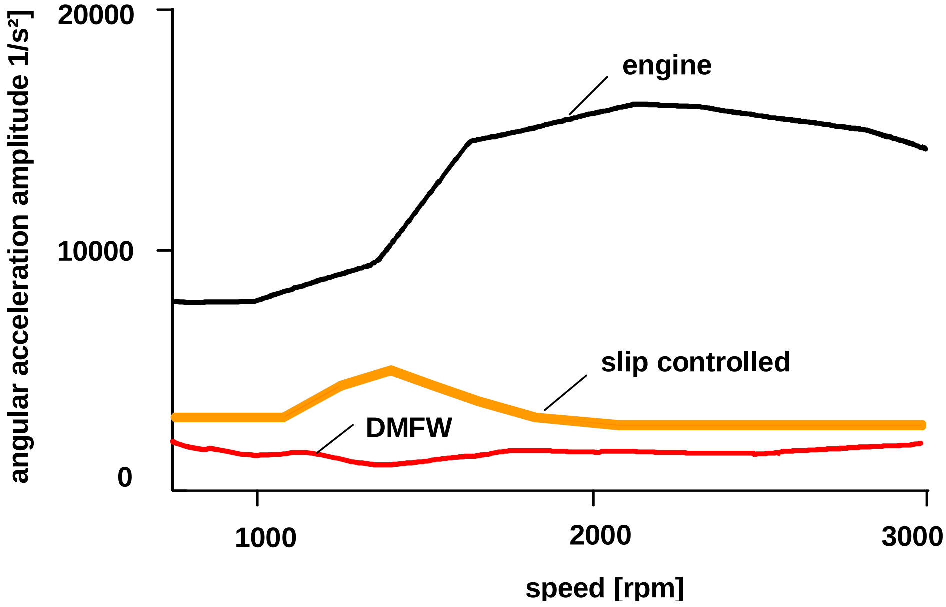


Figure 21: Comparison of the vibration isolation of a dual mass flywheel with a controlled slip isolation system

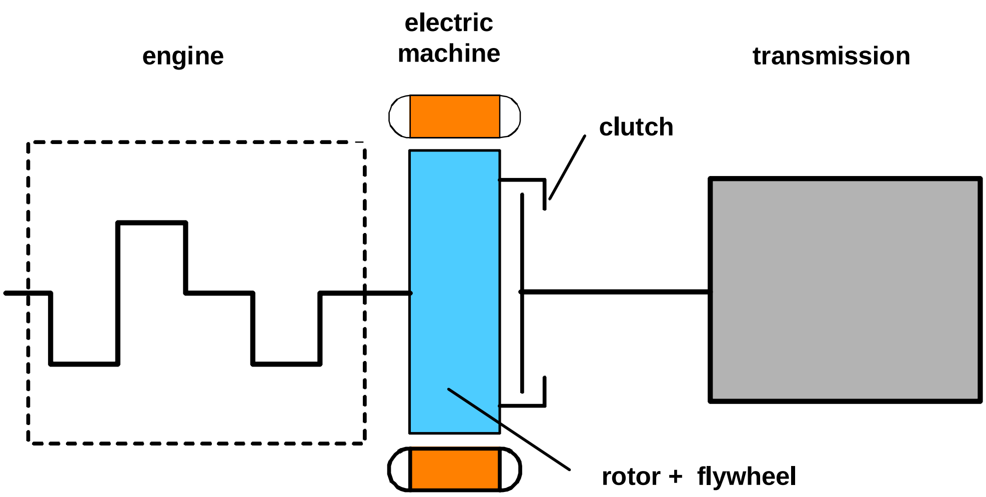


Figure 22: Schematic representation of the crankshaft starter generator

It deals with the possibility of reducing the torque variations by deliberately generating counter-torque with an electric machine. This seems easy to do if - for a completely different reason - a crankshaft starter generator should be used. These crankshaft starter generators, which have been developed by different companies, combine the starter and generator in one machine, which is installed between the engine and transmission instead of the flywheel (Figure 22). Such an electrical machine could work as a generator and draw the corresponding amount of torque from the crankshaft whenever the engine delivers too much torque after ignition, in order to give back the torque as an electric motor during the compression phase. Theoretically, the torque curve could be completely flattened out.

Figure 23 illustrates the torque curve over the crankshaft angle for a four-cylinder diesel engine. Based on a speed of 1500 rpm, a crankshaft angle is determined for one half rotation.

It can be seen which energies must be considered based on the areas above and below the middle torque line.

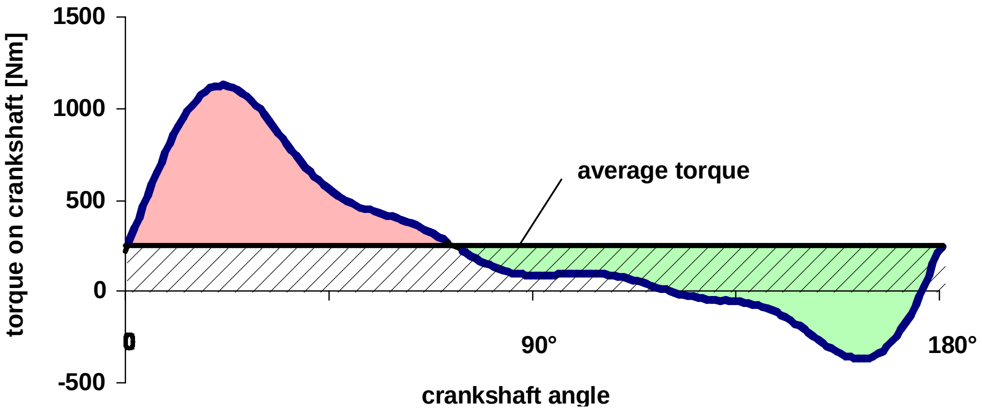


Figure 23: Torque curve over crankshaft angle

The energy to be taken away by generating a current in case of excess torque is represented by the red area. This must be stored for a short time, for example, in a capacitor.

During the compression phase, this energy must be returned via the electric motor (green area).

To estimate which amounts of energy are involved, the area corresponding to the average torque, which corresponds to the work of the combustion engine, was also shaded.

A simple comparison of factors shows:

The electric power, which must be transported back and forth, achieves the same order of magnitude as the average engine power. In other words: some 10 kW are constantly being generated in the electric machine, balanced out, stored in the capacitor, released again, sent via an inverter to then drive the electric motor. Even if an implausibly high efficiency of 98 % is assumed for each individual step, the overall efficiency would only be approximately 88 %. That means 12 % of the electric power transported back and forth, which is in the order of magnitude of the combustion engine power, or a significant amount of kW, would be converted to heat.

Even if it is assumed that full torque compensation is unnecessary because the DMFW cannot completely eliminate the vibrations, the energy balance of a corresponding electrical machine would fall extremely short with a diesel engine.

Gasoline engines are more favorable here (Figure 24). Nevertheless, this does not change the fact that there are enormous electrical losses with active torsion excitation damping from an electrical machine.

vehicle	**amplitude of the torque variations [Nm]**	
	idle	**drive at 1500 rpm**
four cylinder diesel	**300**	**700**
six cylinder diesel	**280**	**700**
four cylinder gasoline	**35**	**290**
six cylilnder gasoline	**35**	**300**

Figure 24: Typical torque amplitudes on the crankshaft

By contrast, the DMFW achieves fantastic values in this regard. Due to the vibration angle within the DMFW, some energy is also lost because of the friction generated. For the unfavorable case of a four-cylinder diesel engine at 1500 rpm, the loss from the hysteresis cycle is the same as for the corresponding vibration angle. This is a loss of approximately 50 W, or a factor of 100 less than with the above-described active damping.

LuK therefore goes on the assumption that a mechanical damping system similar to the DMFW is still needed with the use of crankshaft starter generators.

Summary

In the European High Group, the DMFW has become quite successful and is just about to penetrate into the mid-size class. Early developments for small engines below 1.6 liter engine displacement give reason to expect that there will be numerous DMFW applications for this area in a few years.

The DMFW offers the best vibration isolation, which cannot be provided by any other system today. In addition to the familiar advantages, the elimination of gear rattle and droning are additional advantages, which were not considered as much in the past.

Lower transmission loading can be expected by filtering out changing torque portions, particularly for diesel engines. Crankshaft vibrations (torsion and bending) are reduced. This allows for a new crankshaft design. It must be noted that the engine irregularity is likely to increase due to the low flywheel mass of the DMFW.

The elimination of gear rattle prevents numerous customer complaints who fear that their transmission could be damaged, causing expensive disassembly costs during the warranty term.

Fuel consumption and emissions are reduced by driving in a lower speed range.

The additional system costs for an optimized DMFW are still higher than for a conventional solution. If the secondary advantages are considered, many DMFWs are already cost-neutral.

Nevertheless, LuK is still trying to reduce the cost of the DMFW. This should lead to new fields of application. In addition to the customary rationalization measures and fine tuning, a transition will be made in the coming years from a ball bearing to a bushing. Furthermore, work will be conducted on dry DMFW dampers, which are not being considered for critical engine sizes, but which can produce a cost efficient DMFW concept for a large portion of vehicles.

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Automation of Manual Transmissions

Dr. techn. **Robert Fischer**

Dr.-Ing. **Reinhard Berger**

Introduction

Many people complain about increasing traffic, more and more regulations as well as higher gasoline prices. All of these factors point to an increase in the automation of the drive train. The automation solutions that will be accepted will depend on how economical and comfortable the new systems are. The automation of the manual transmission promises to be an economical and comfortable solution.

In which market segments do we expect manual transmission automation to gain acceptance?

With Electronic Clutch Management (EKM), the driver determines when and how the gears change. In this respect, the behavior is very similar to a conventional manual transmission. LuK mastered the clutch strategies for all vehicle and torque classes. Therefore, the EKM can be offered in every torque class for drivers who like to shift gears themselves.

The gears are changed automatically with the automated shift transmission. In contrast to the automatic step transmission, the ASG must interrupt the tractive force when shifting. This is more clearly detectable the higher the tractive force that is interrupted. For this reason, it can be surmised that ASG will be accepted mainly in small cars. The increasing use of electronic throttle simplifies the introduction of ASG.

This report can be divided into three major sections. First, the current production status of the EKM will be presented. The second section deals with the further development of this system. The third part deals with the automated manual transmission.

Electronic Clutch Management: Production Status

Basic Layout

Figure 1 illustrates the basic layout of the electric motor-driven EKM. This picture was presented already at the last colloquium [1] and illustrated the direction of development at that time. The basis of the EKM here is the self-adjusting, reduced-load clutch (SAC) [2], which allows a small electric motor as an actuator in combination with the torque tracking (see Chapter 2.3). This small electric motor has low heat build-up, so that the actuator and controller can be integrated into an "intelligent actuator". This replaces the clutch pedal and delivers the highest clutch comfort. Changes to the release system and to the transmission are not necessary. The only additional expenses required are sensors to detect the intention to change gears and to recognize the gear. All other signals are already available in the vehicle.

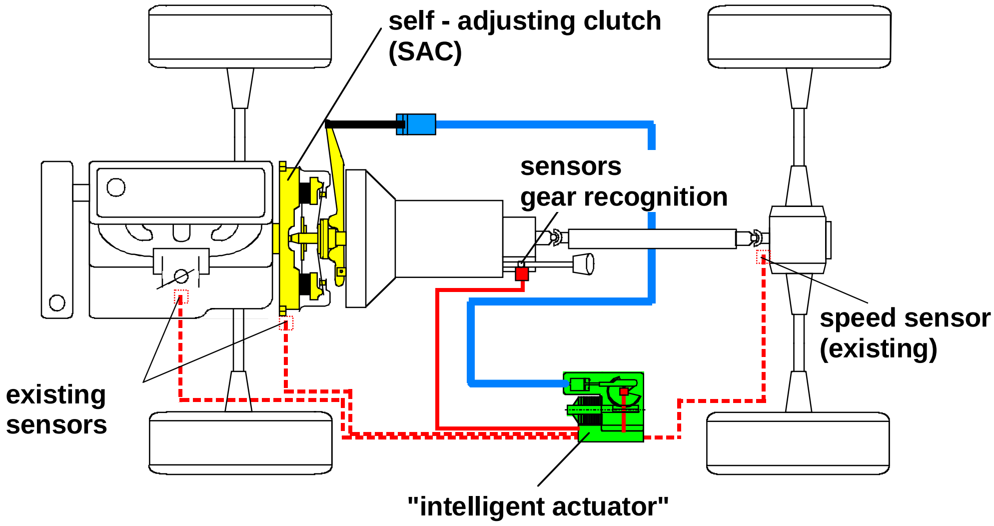


Figure 1: System overview: electric motor-driven EKM

LuK is cooperating with BOSCH in the development of EKM and LuK is responsible for the entire system.

Minimizing Costs

One very important goal in the development of the EKM was keeping costs down. The system was supposed to be purely an add-on system, i.e., changes to the transmission, gearshift mechanism and release system should be avoided. In addition, the number of sensors needed should be limited to as few as possible and the cabling expenses should be reduced (see Figure 2). It was thus determined to eliminate the clutch travel sensor and the transmission input speed sensor. The goal for the intention to change gears was to detect the driver's intention by using a travel measurement and without changing the gear-change feeling or modifying the gearshift. All this could be realized through intelligent software strategies [3].

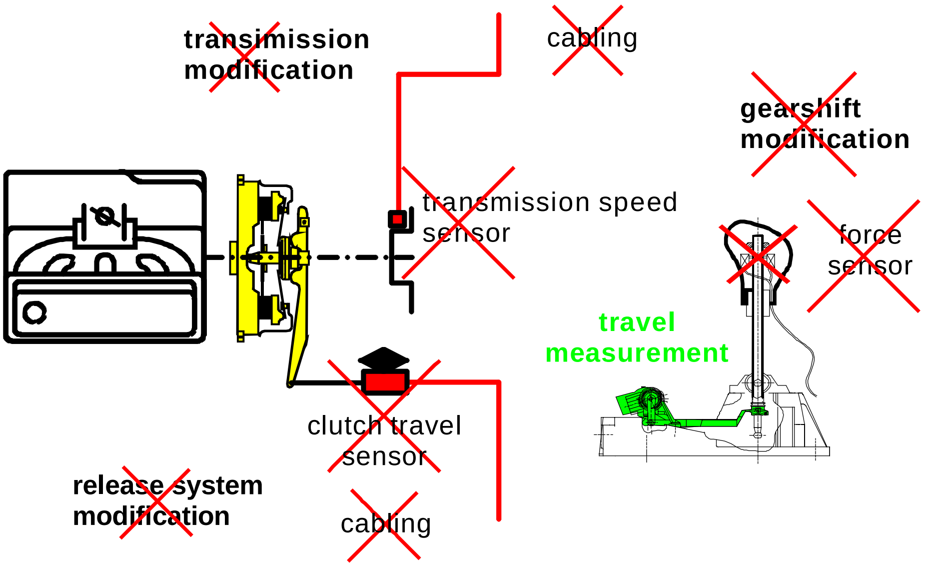


Figure 2: Add-on EKM: Minimizing costs

Another point is the integration of actuator and control that were already mentioned.

Torque Tracking

Torque tracking is the important basis for fast clutch times despite a small electric motor, and for good load cycle comfort. The function of torque tracking is explained with the help of Figure 3.

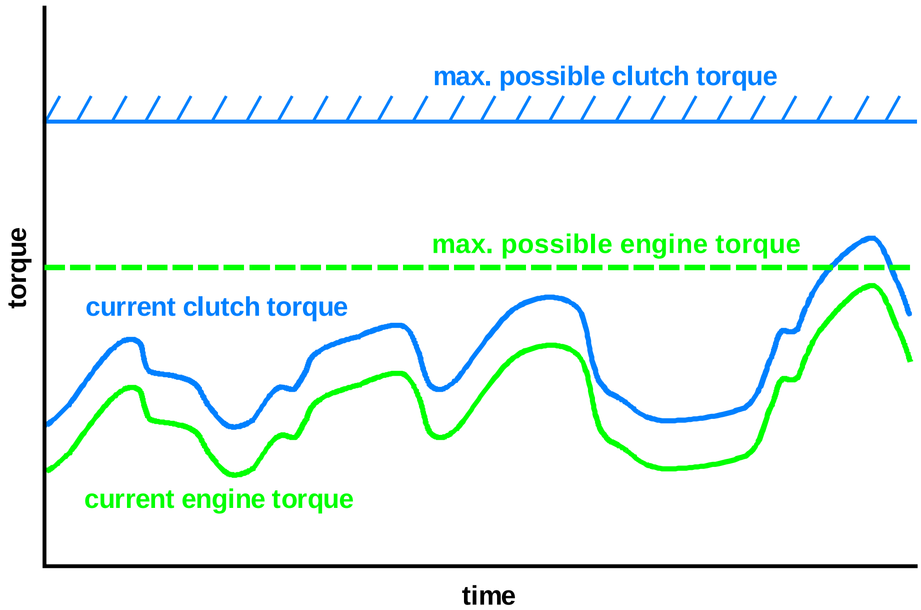


Figure 3: Time curve torque tracking

A clutch must be able to safely transfer the engine torque even in the worst case and must thus have sufficient additional reserves. In practice, a fully closed clutch can transfer 1.5 to 2.5 times the maximum possible engine torque. Torque tracking is based on the idea of adjusting the clutch torque to the current engine torque and only allowing a small safety margin.

Figure 4 illustrates the advantage during a shifting cycle. With a conventional system without torque tracking, the clutch torque remains much higher than the engine torque. If the driver wants to change gears and lets up on the gas pedal, the engine torque decreases. When he moves the shift lever, the intention to switch gears is triggered and the clutch must now go from “completely closed” to “completely open”. This defines the disengagement time. This must not be too long or else the clutch still transfers torque during the synchronization of the next gear, which can lead to transmission chatter or damage. Figure 4b illustrates the same process with torque tracking. The clutch torque is only slightly higher than the engine torque. Thus the travel to “completely open” is already significantly less than with a conventional sequence. If the driver then lets up on the gas because he wants to change gears, the engine torque and thus also the clutch torque decrease immediately. By triggering the intent to switch, the clutch is thus already almost open and the rest of the disengagement occurs very quickly. Even very sudden gear changes are thus possible without transmission noise or damage.

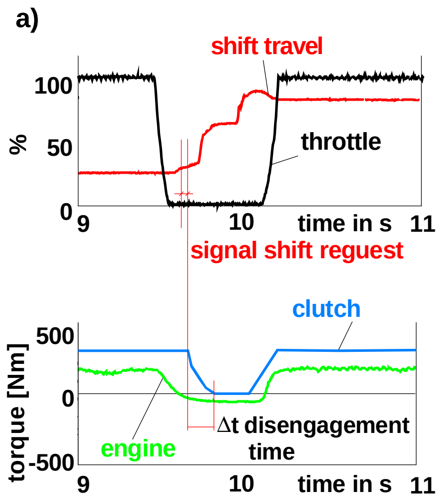


Figure 4: The shifting process a) without torque tracking and b) with torque tracking

The alternative to the electric motor-driven EKM is the hydraulic EKM. Such a system has already been produced by LuK. It is considerably more expensive than the electric motor-driven version, but in exchange it is theoretically faster. Torque tracking makes extremely short electric motor-driven clutch times possible, making it comparable to the hydraulic solution.

Load cycling is another advantage of torque tracking. A faster gas tip-in generates torque peaks and thus surge oscillations, which occur with very different intensities depending on the vehicle (Figure 5a). These are prevented by torque tracking through a very short slip phase, which is not relevant to use and wear.

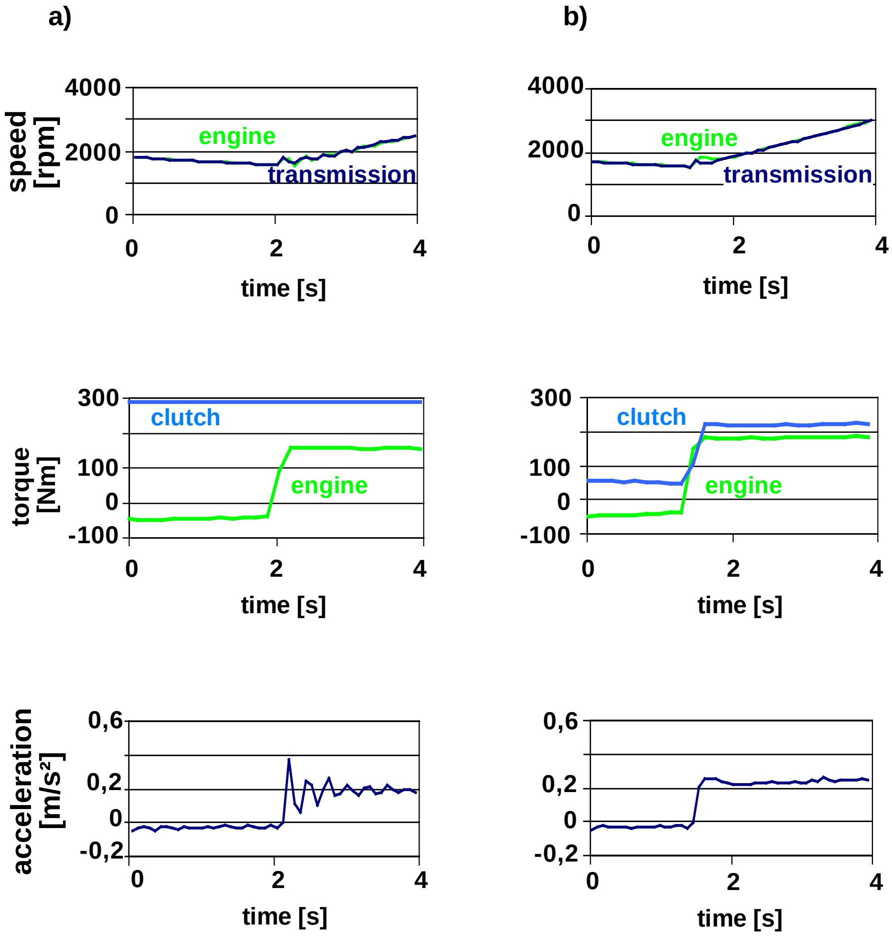


Figure 5: Load cycle a) without torque tracking and b) with torque tracking

The demands on the drive train are also limited via EKM with torque tracking:

- With jack-rabbit starts there are no jerks due to sudden snapping of the clutch (valid for all EKMs).
- The maximum torque transfer reserve of the clutch is usually not used; the clutch works as a torque limiter.
- For this reason, even with output-side impacts the peak torque value is reduced.
- The clutch wear tends to be lower than with pedal actuation because the electronic systems act optimally in every situation, in contrast to the driver (applies to all EKMs).

The drive train would not require as high of a safety factor with 100% EKM use.

Because of the torque tracking, the release system with its seals, lines and the release bearing is constantly under load. This has proven not to be critical in more than 4 million test kilometers because:

- The low release force of the SAC leads to a comparably low maximum load.
- The additional travel will be compensated with clutch torque modulation by shortened actuation travel when opening the clutch so that the total actuation travel is not longer than with a conventional system.

Production Design

Figure 6 illustrates the current production components, the intelligent clutch actuator, which replaces the clutch pedal, the release force-reduced self-adjusting clutch (SAC) and the sensors offered as add-ons for gear-change intention and gear recognition.

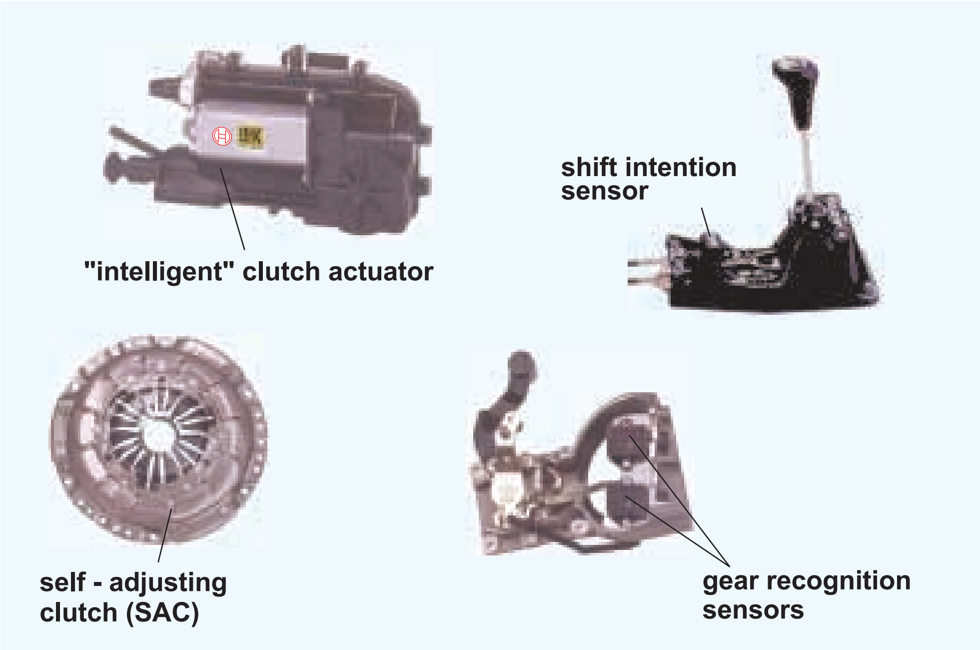


Figure 6: Add-on EKM: Components

The clutch actuator includes the master cylinder, which is otherwise integrated into the pedal block. A semi-hydraulic release system could serve instead of the hydraulic release system with a central release bearing. With very little modification, it could be adapted to purely mechanical clutch actuation.

The simplicity of the components means that high functional reliability can be expected. The components are also so simple because of major software development expenses for additional sensors, which means, for example, that a transmission input speed sensor and clutch travel sensor could be avoided. The following press statements prove that this does not mean that comfort is compromised in any way:

...during the test drives, it was equipped with a semi-automatic (only shifting, not using the clutch), which harmonizes excellently with the concept...
Autoflotte 7/1997

...Shifting made fun - especially if the automatic clutch is ordered. It functions so perfectly that we would abandon all tip and steptronics in this world for this clutch...
FAZ 10/1997

...Another advance in comfort is provided by the automatic clutch developed by the specialists LuK...
mot 17/1997

...The recently developed automatic clutch...Shifting has become a true pleasure and it can even lead to lazy shifters using the best gear with regard to comfort and fuel consumption...
Handelsblatt 26.06.97

...or semi-automatic with manual shifting without using the clutch (David Coulthard would buy this version)...
Die Welt 28.06.97

Figure 7: Press statements on the LuK EKM (translation)

Electronic Clutch Management: Further Development

Goals

A truly economical and efficient solution has already been achieved with the current production status. The market penetration will increase as the system becomes less expensive and more compact and as additional functions are realized. LuK has set the following goals for itself:

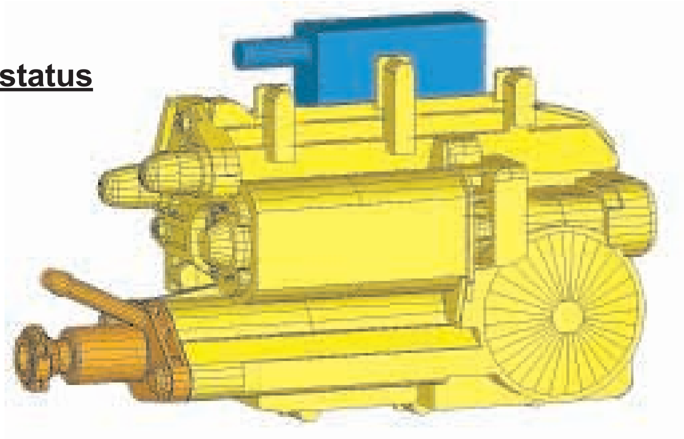
- Minimizing size and weight
- Improving applicability
- Offering higher torques
- Expanding functions
- Decreasing costs

Clutch Actuator

The current production status includes components that have been proven in other vehicle applications and thus have been used. These include a worm gear transmission and a crank mechanism among others. A screw drive offers greater flexibility and the possibility to balance out the tolerances without initial adjustment.

In addition, the control and electronic power units are smaller. Overall, therefore, the size and weight are reduced and a higher efficiency is achieved.

production status



new development

size / weight	- 30%
applicability	⇑
performance	+ 50%
costs	- 10%

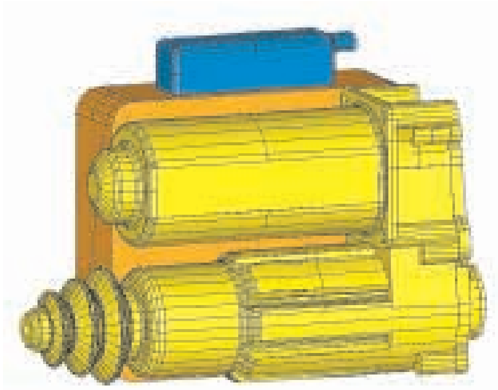


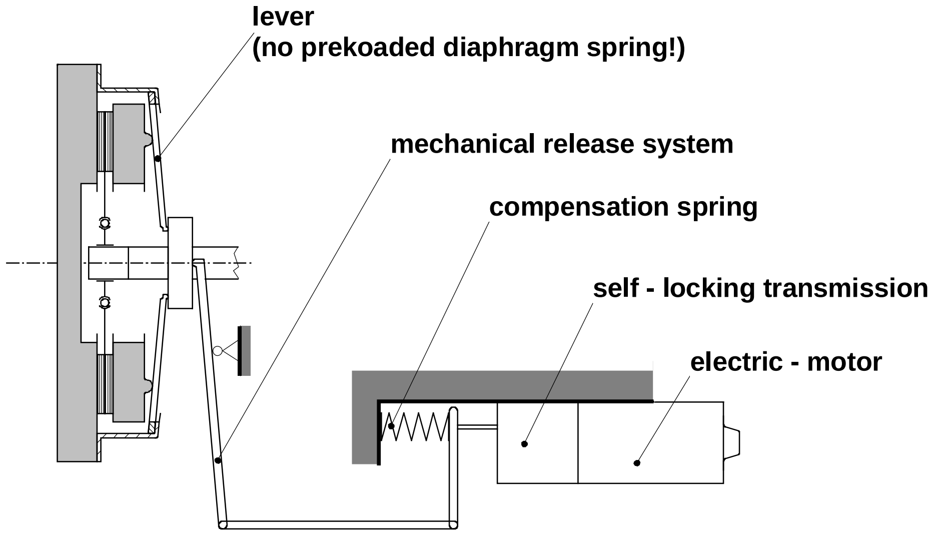
Figure 8: New development - clutch actuator

By following a modular concept, the applicability of the clutch actuator has also been improved, particularly by building it directly on the transmission.

Clutch

The SAC used with the EKM already reduces the load needed and thus the work the actuator must perform in comparison to the conventional system. The development goal is better tuning of the entire "clutch, actuator and software" system in order to achieve simpler and less expensive components. approach is that a clutch can be closed by an outside force, which LuK calls an "active clutch" (AC), see Figure 9.

In the form presented here, the AC is simpler than the SAC or a conventional clutch. A simple lever is used to close it. This results in a direct relationship between actuation travel, actuation force and clutch torque, which improves the controllability and the ability to modulate the clutch.



size / weight	⇓
applicability	⇑
costs (clutch)	**-20%**

Figure 9: Active clutch (AC): Basic diagram

Why can the clutch be simplified in this way?

Because of the torque tracking, there is a characteristic frequency distribution of the clutch torque and thus also of the actuation travel. It is known that the idle stage and low-load portions are relatively high not only in the specified driving cycles but also in practical operation. Correspondingly, the maximum in the frequency distribution is at a relatively low clutch torque (partial load and coasting operation) see Figure 10a.

If the actuation force of the SAC is then considered, one sees that the greatest force occurs mostly at the maximum frequency (Figure 10b). The idea was to set the actuation force to “zero” at the maximum frequency with a new clutch design. The AC fulfills this condition, see Figure 10c.

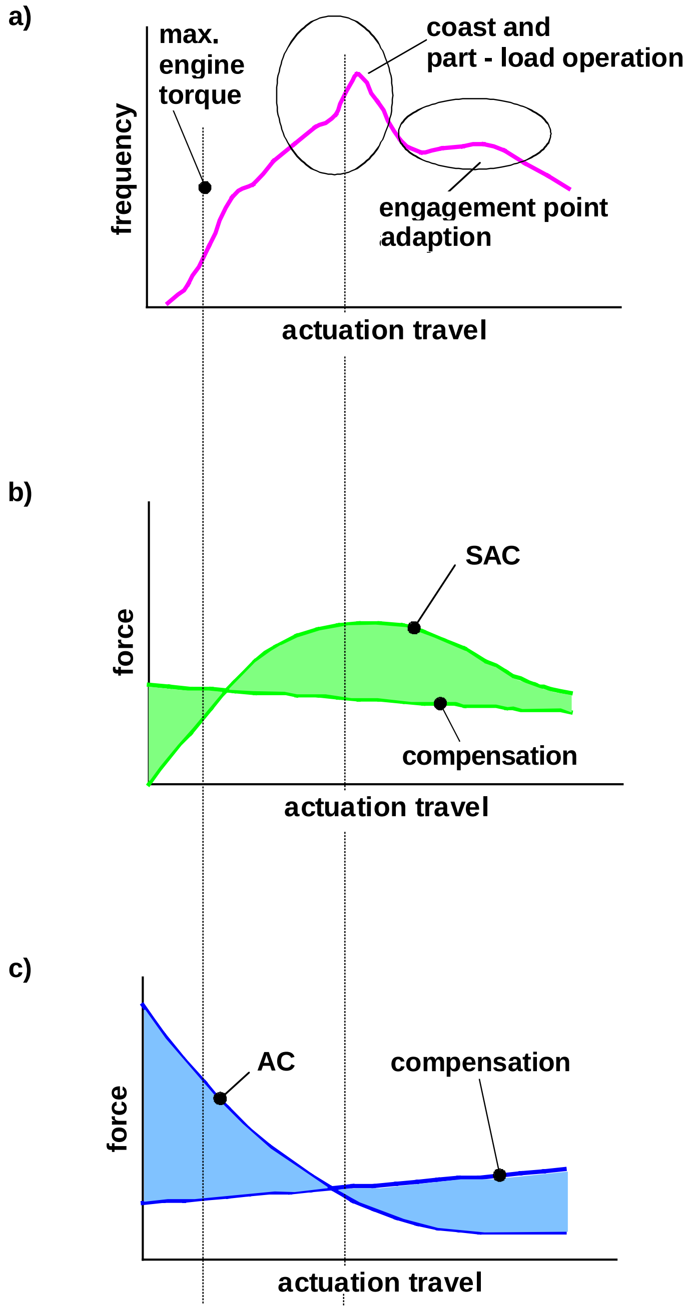


Figure 10: Active Clutch (AC), load population with torque tracking

The total energy load (actuator work) limited by the performance capacity of the actuator is the same for the AC and SAC, but clearly better than with a conventional clutch. Figure 11 illustrates this. Through suitable wear compensation, it is again possible to achieve a clear decrease of the required force with the AC and thus the actuator work (hatch-marks in Figure 11). The result is that it is possible to transmit higher torques or use a smaller electric motor.

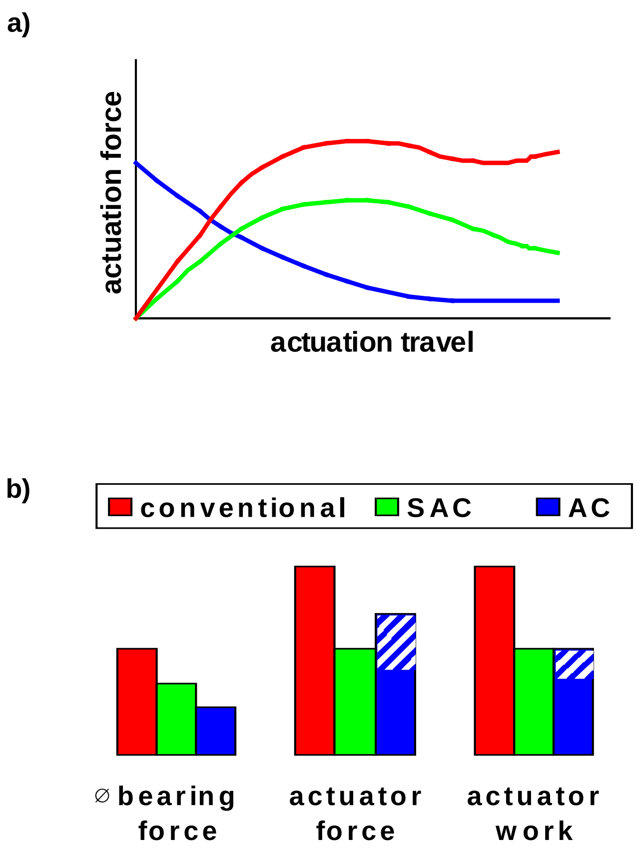


Figure 11: Active Clutch (AC) - system comparison

Recognition of Gear and Intention to Shift

Today, one sensor is used for the intention to shift and two sensors for gear recognition.

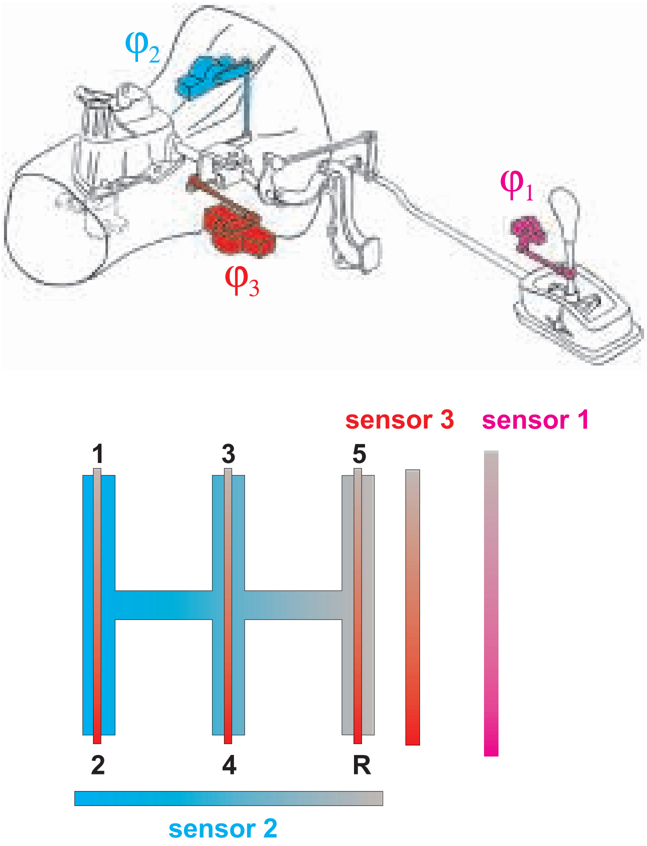
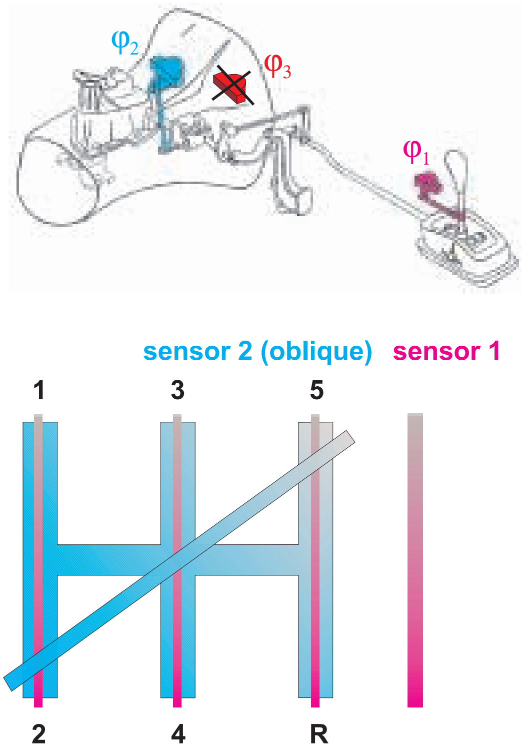


Figure 12: Recognition of gear and intention to shift with three sensors

Sensor 3 for gear recognition in the shift direction and the shifting intention sensor (1) actually operate in the same movement direction. Whether two sensors are needed here depends significantly on the external shifting. Because there is both play and elasticity there, the distinction between shifting intention recognition and gear sensing is unclear.

In order to get by with only two sensors for external gearshifts with play and elasticity, the selection sensor is disposed at an angle. In this way, it covers both the selection and the shift direction. The less precise signal of the shift intention sensor related to the transmission is sufficient in this combination to clearly identify the gear position.



size / weight	⇓
applicability	⇑
costs	-30%

Figure 13: Recognition of gear and intention to shift with only two sensors

In the best case, two sensors will be sufficient in the future for recognition of gear and intention to shift functions.

Slip

Previous EKM development focused mainly on the goal of vibration damping through slip [4]. This goal was abandoned because the overall wear was very high with continuous slip systems. This was not only because of the high dissipated energy but also the specific wear from continuous slip was clearly higher than during launching and shifting.

The SAC has an increased wear reserve and thus offers a favorable perspective. Nevertheless an effort has been made to minimize the wear that occurs in slip regulation.

Another problem area with constant slip systems is the increased fuel consumption. The fact is that with a quieter drive train, drivers drive in higher gears, resulting in better fuel consumption, as has been confirmed with the DMFW.

For this reason, LuK has optimized the entire system.

In Figure 14, the relationships are first shown without slip. Then it is possible to hold the amplitude variation at the transmission input in drive below the engine excitation (14/1) using conventionally optimized torsion dampers (14/2). If slip is then used, it can be assumed that the torsion damper can be omitted. However, the variation amplitude when using a rigid clutch disc (14/3) is, however, significantly higher than with an optimized torsion damper. Thus, in this example, the variation amplitude with the rigid clutch disc at 1600 rpm is approximately four times higher than with the normal torsion damper.

In order to achieve an improvement with slip, one can assume as a rough reference value that slip must be generated in sizes of the otherwise available variation amplitude. Hence, a rigid clutch disc requires a very high slip in all driving ranges.

For the optimal system with a slip clutch, a simplified torsion damper is used without a friction control device and with a relatively low spring rate. It is apparent in the picture that non-slipping (14/4), the isolation from a speed of 1300 rpm is better than with the optimized torsion damper.

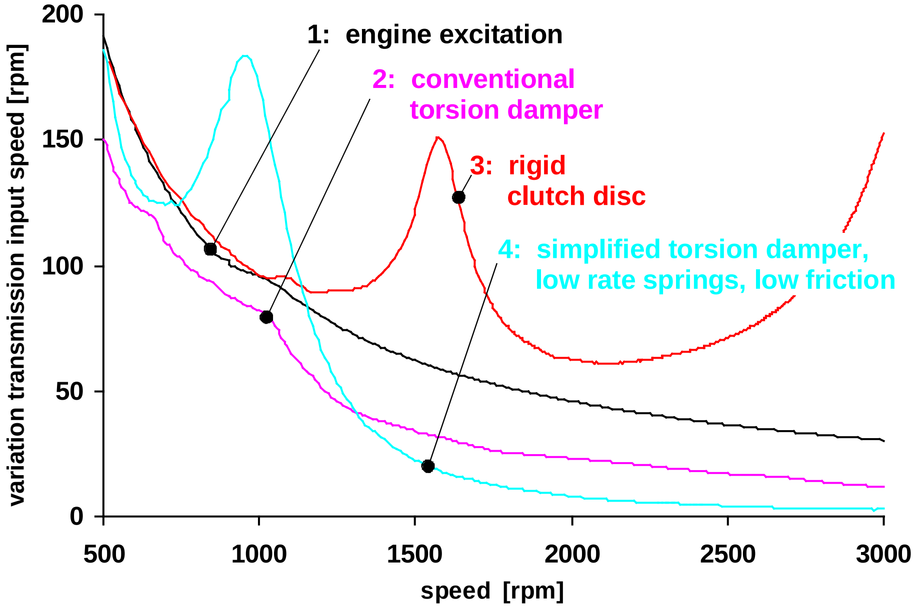


Figure 14: Drive train vibrations with non-slip systems

How is optimization achieved? If slip is used with a simplified torsion damper, the resonance can be eliminated in the lower speed range. Interestingly, a new resonance peak occurs (shown as a thin line in Figure 15), which is higher than the resonance without slip. This can be explained by the change in the distribution of the rotating masses. The drive train in the vibration system without slip is balanced between the heavy engine rotating mass and the total vehicle mass. In a slip system, the engine rotating mass is replaced by the significantly smaller rotating mass of the clutch disc. It would thus make no sense in the example shown to slip in the range from approximately 1600 rpm. Here the variation amplitude without slip is clearly smaller (see Figure 15). It remains to be mentioned why this resonance is excited at all in a slip system. The reason for this is that a slip system can never ideally isolate. Excitations come into the drive train via the slipping clutch through the friction coefficient curve, through varying slip speed, but also through flat geometric deviations like runout and similar factors.

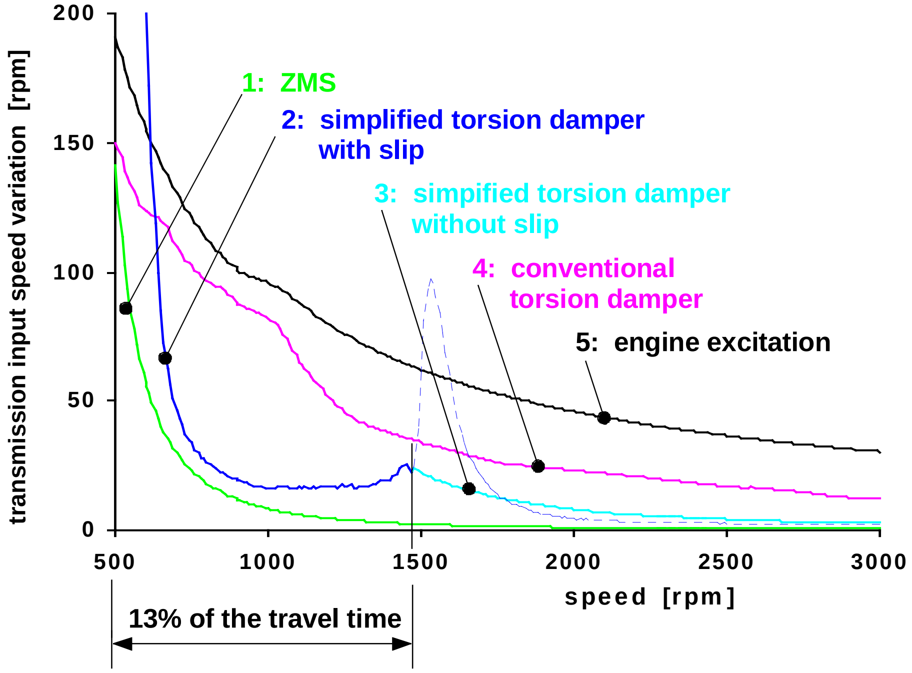


Figure 15: Optimizing the entire system with a slip clutch

It can be seen in Figure 15 that a clearly better vibration isolation can be achieved with the optimized slip system and simplified, low rate torsion damper (15/2 and 15/3) than with a conventional torsion damper (15/4) alone. In any case, the isolation quality of the DMFW (15/1) is not achieved at representative slip values.

A vehicle with such a slip system was sent for customer testing, which yielded the following interesting results: Only 13% of the driving time was within the ranges in which slip was necessary. With this collective, there was a wear increase of 13% and a fuel consumption increase of only 0.4%. The SAC slows this increase in wear slightly .

There is still one catch. This example is valid for a rear-wheel drive vehicle, whose resonance speeds are normally relatively low. The resonance of front-wheel drive vehicles is usually significantly higher and cannot be moved out of the main driving range with a low rate torsion damper. However, this optimized system still shows improvements when compared to the conventional system, but the wear and fuel consumption are still relatively high. Test on a compact class vehicle showed an increase in wear of 40 % and in fuel consumption of 0.8 %.

Automated Shift Transmission (ASG)

Set-up

The EKM and the production experience gained with it are used advantageously as a part for the manual transmission automation.

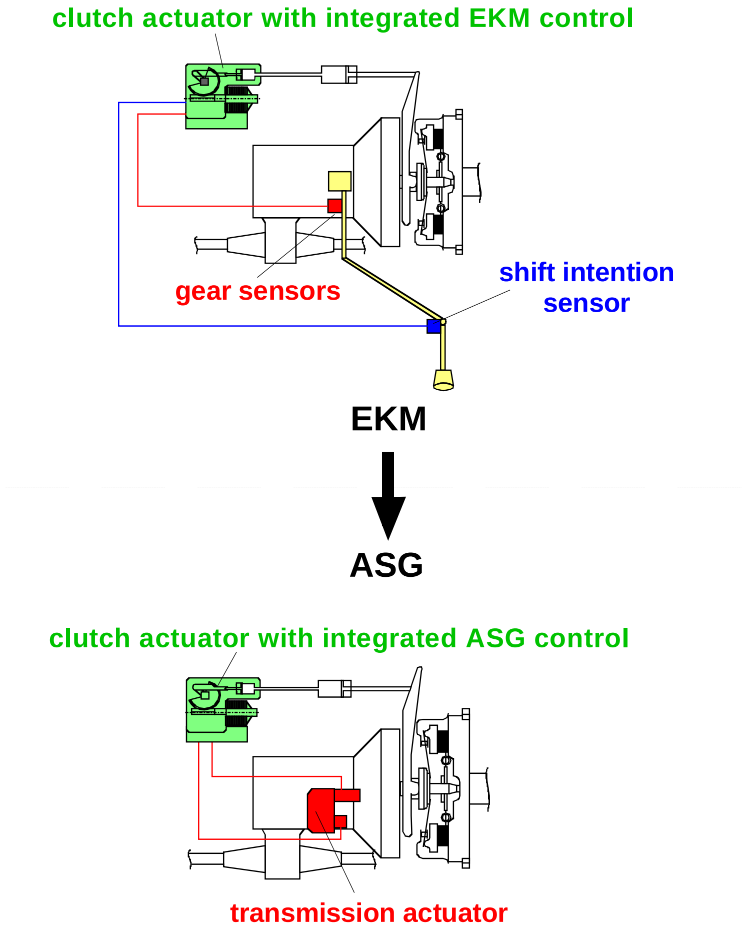


Figure 16: Add-on ASG based on EKM

The external gearshift still needed with EKM including shift intention recognition and gear recognition is no longer included. It is replaced by an electric motor-driven transmission actuator (Figure 16).

Also with this development, the attempt is made to avoid changes to the transmission, i.e., to design the system as an add-on. This requires a high degree of flexibility. In order to use as many standard components as possible, the ASG is laid out as a modular system. The adaptation of the transmission actuator to the transmission occurs via an intermediate transmission. Electric motors and controls are conceived as standard components for all users, in order to realize high production quantities. Even with the ASG, the cooperation with Bosch proves useful because they provide these standard components.

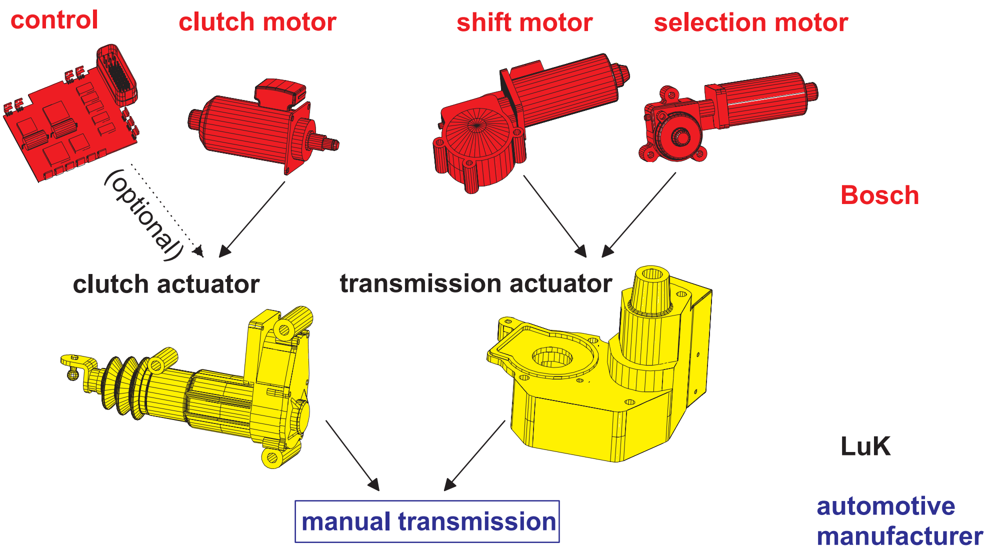


Figure 17: Modular design ASG

As mentioned in the previous chapter, the control electronics have been designed significantly smaller. For this reason, control and end stages for the ASG engines are now integrated in the clutch actuator.

The software development tasks are also being shared with Bosch. Control-operating systems, including electric motor-specific basic control and the shift-time calculation including driver model come from this worthy partner. The control of the clutch, the gear motors, the characteristic for the combustion engine and the overall coordination of the shifting process is made by LuK.

Each customer naturally has a specific philosophy when and how its transmission should change gears. The LuK concept offers wide-ranging application possibilities with which the gearshift time and process can be influenced.

Basics of the Gear-changing Process

The basic problem with an automated manual transmission is the tractive force interruption. In Figure 18 this is the valley between the two shifted gears. The torque must first be reduced on the engine and on the clutch. Then the gear is disengaged, it is synchronized, the new gear actuated and finally the torque is built up.

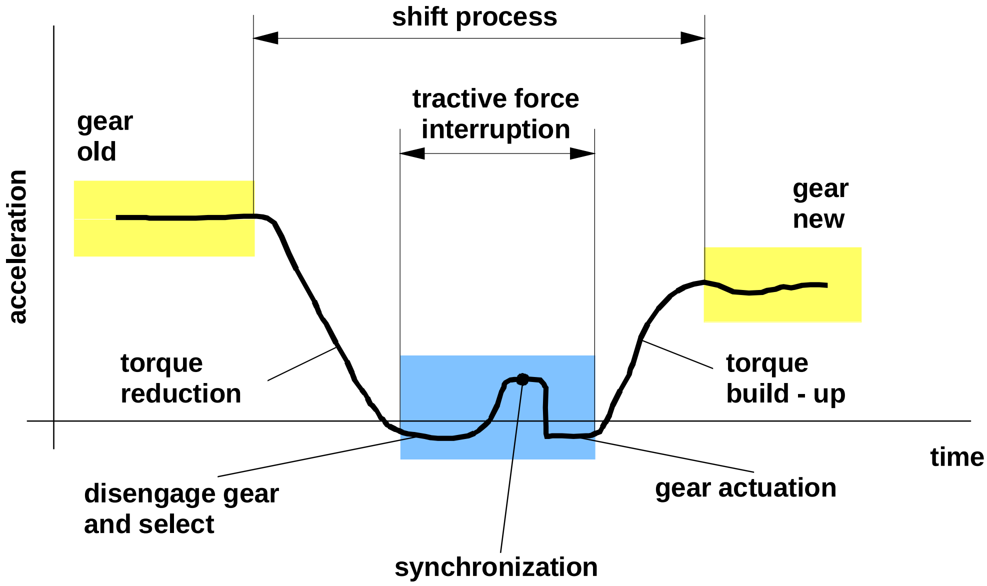


Figure 18: ASG shifting process: Phases

These phases of the shifting process can be divided into two stages with respect to the requirements of the shifting element:

- Processes that effect the vehicle acceleration
- Processes that represent pure dead times

With the processes that affect vehicle acceleration, it is apparent that the actuator element “electric motor” must be throttled, because changing the vehicle acceleration too quickly is unpleasant. Only an optimization leads to well-tuned interaction of engine, clutch and transmission engagement. The load can be greatly reduced during synchronization, for example, by intermediate gas.

In the dead times, the maximum speed of the actuators is required. With this, one must be careful that in disengaging the gear and the subsequent fast phase, the impact on the synchronization is too hard.

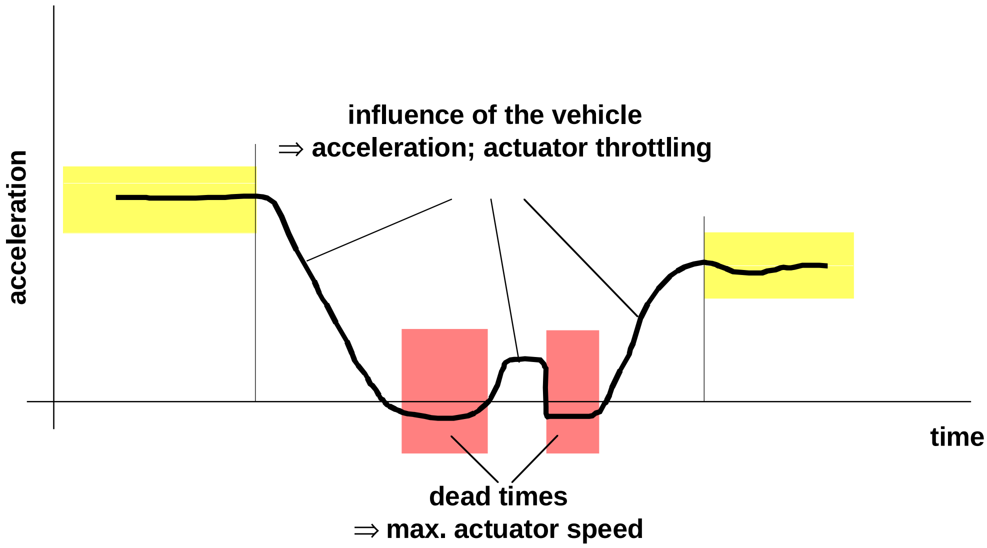


Figure 19: Optimizing tractive force interruption

In response to this contradiction, LuK has developed an integrated shifting elasticity, which responds when the shifting force threshold is exceeded and then exhibits only a low force increase (see Figure20).

This integrated shifting elasticity offers the following advantages:

- The free flight phases, which are pure dead times, are shortened if the integrated shifting elasticity was prestressed (when disengaging from the previous gear, respectively the stop position of the synchronization).
- Defined shifting force ensures constant shifting comfort.
- Transmission and actuators are protected by the elastic force limitation.

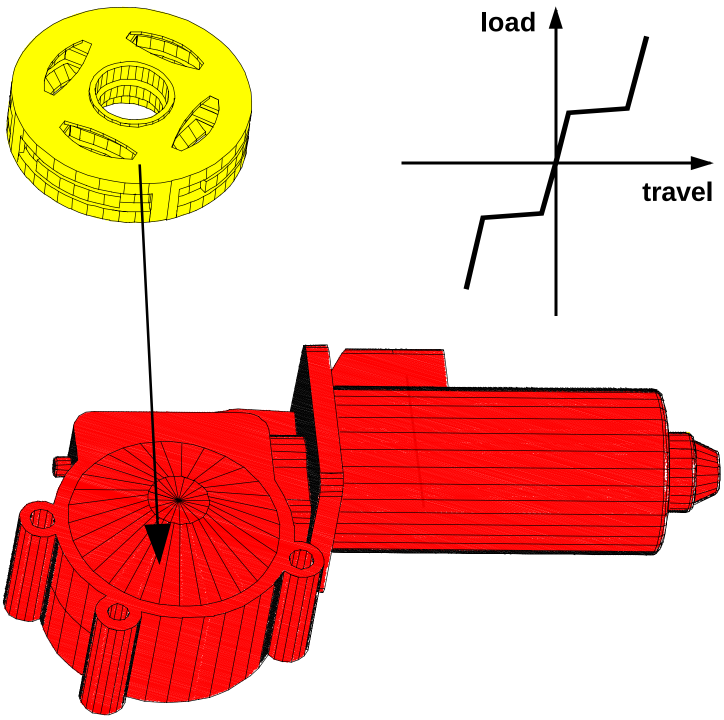
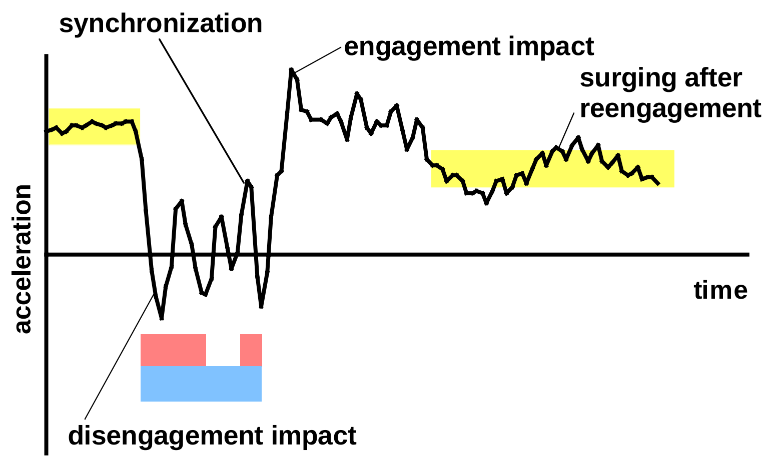


Figure 20: Shifting elasticity for transmission shifting

Measurement

The effects of different shifting process designs are illustrated with the help of measurements. Figure 21a shows the vehicle acceleration at a fast shift with full actuation speed. The rapid torque decrease causes a disengagement impact. The decay of ist oszilluations overlaps with the synchronization impact. The rapid reengagement causes an acceleration peak, followed by slip with maximum clutch torque and a resulting high vehicle acceleration. Finally, surging accurs after the end of the slippage. In comparison the vehicle acceleration during a comfortable shift is shown in Figure 21b. Torque decrease and increase as well as the synchronization period take longer in this case, while the dead times are kept at a minimum as in the shift depicted in Figure 21a.

a) uncomfortable shifting



b) comfortable shifting

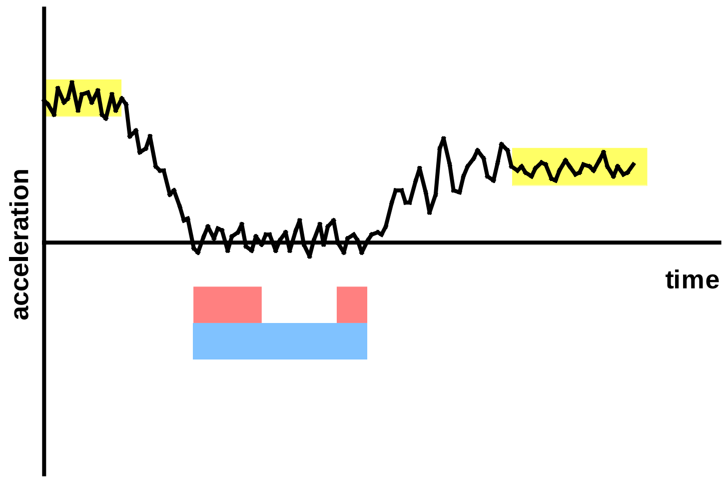


Figure 21: Shifting process measurement

The phases of the shifting process as defined in Figures 18 and 19 are marked in Figure 21 as well.

Future Outlook

LuK is developing the electric motor-driven ASG add-on as part of several customer projects. It is expected to be introduced on the market at the turn of the millennium. It has been installed in several dozen vehicles, which are currently undergoing customer testing.

At the same time, the next generation ASG is already in development. Here concepts are being investigated that differ significantly from manual transmission. The attributes of the ASG add-on are included in this new transmission at sharply reduced costs per unit

Summary

The Electronic Clutch Management (EKM), which was introduced four years ago, is now in production. This offers the customer an economical, efficient and functionally reliable solution.

Size and weight reduction, the operation of more powerful engines, functional expansion and cost reduction are in the foreground as important goals for the further development of the EKM systems. One approach to a solution involves new, more compact clutch actuators, a simplified clutch (active clutch), the elimination of a sensor and operation with constant slip in the lower speed range.

The automated shift transmission is based on the automated clutch actuation of the EKM-System. The external shifting, including gear recognition and recognition of intent to shift is replaced by another electric motor-driven transmission actuator. An important goal here as well is to produce an add-on system and to avoid changes to the transmission and release system. Theoretical investigation of the shifting process, characterized by tractive force interruption, as well as measurements confirm the assumption that fast and comfortable shifting can be achieved as well with the motor-driven ASG as with the hydraulic system.

With today’s customer expectations and the increasing number of technical options, we are convinced that EKM and ASG will have a broad market success. This especially applies to small cars and for developing countries where automatic transmission are not readily available.

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The Torque Converter as a System

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Dipl.-Ing. **Uwe Wagner**

Introduction

Since its introduction in the 1940s, the torque converter has been utilized as a proven coupling element between the engine and automatic transmission. Its advantages result from the principle of hydrodynamic power transfer from engine to transmission. The available torque amplification (torque ratio) improves acceleration and performance. The ever-present and, in principle, necessary slip isolates the drivetrain from the engine torsional vibrations, prevents Tip-in reactions and provides comfortable transmission shifts. Lock-up clutches with dampers are used to bypass the torque converter and minimize the losses due to slip in the torque converter.

In contrast to automatic transmissions which have had their functionality continuously improved, torque converter design and concept have not been changed significantly throughout the preceding decades. Even the introduction of lock-up clutches in the 1970s hardly changed the principles of torque converter concept, layout and design. This is surprising because of the well-known potential provided by the torque converter system. With a well-tuned lock-up control strategy and appropriate hardware components, tremendous improvements in performance and significant fuel savings can be realized.

In the last Symposium in 1994, LuK presented an innovative torque converter system called the LuK-TorCon-System. This system still uses the advantages of the torque converter but minimizes the disadvantages by introducing a high-performance lock-up clutch and an aggressive lock-up control strategy. Depending on the application, the LuK-TorCon-System provides significant improvements in performance and fuel economy. Furthermore, the system enables us to gain the same performance characteristics using a conventional 4-step automatic transmission instead of a more expensive 5-step automatic transmission. However, the demand for further improvements in performance and fuel economy dictates additional optimization steps. In this context, the selection and tuning of the lock-up control strategy is as important as the design and tuning of the three hardware components: hydrodynamic circuit, lock-up clutch and damper. In fact, only with this holistic approach is it possible to attain the entire advantages provided by the torque converter system.

In spite of these circumstances, over the last few years, LuK has developed torque converter systems, which provide superior characteristics compared to conventional systems. With reduced envelope, lower weight and equal costs, the LuK torque converter systems deliver more functionality, greater flexibility and enables to reduce fuel economy and emissions and increase vehicle performance.

LuK has increased the torque converter development capacity and capabilities in Bühl, and in Wooster, USA, where we have established a second Torque Converter Development Center. At LuK, Inc., USA, we started producing both medium and heavy duty torque converters for Allison’s World Transmission Series in 1997. Figure 1 shows the MD Torque Converter, that also utilizes the advantages of a lock-up clutch.

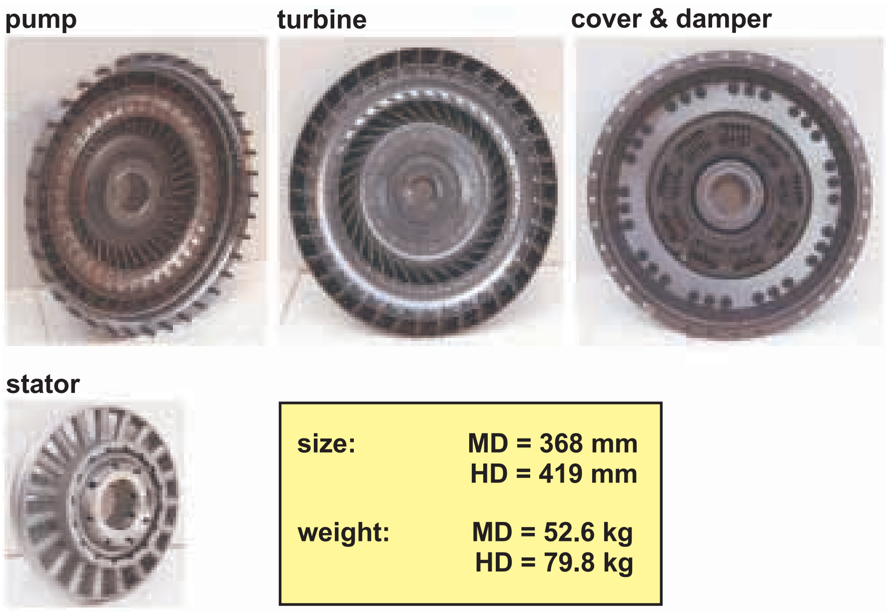


Figure 1: Components of the MD torque converter series

History and Global Development of Automatic Transmissions and their Influence on Modern Torque Converter Concepts

The American car manufacturer, Oldsmobile, presented already the first fully automatic transmission for passenger vehicles in 1940 (see figure 2).

1940

Oldsmobile introduces the 1st fully automatic transmission

Fokus

- **comfort**
- **less driving effort**

Figure 2: The beginning of automatic transmissions

At this time, the main focus was on increasing the driver's comfort. Any disadvantages in performance and fuel economy were not yet important. The state-of-the-art transmission was a 2-speed-transmission with a total gear ratio of 1.8 (as shown in figure 3). To compensate for the relatively small total gear ratio, torque converters with a torque ratio of 3 or even 4.5 were used, providing high comfort but resulting in a system with poor fuel economy. Overall, the complexity and functionality of the torque converter used in the 40s and 50s were much higher than those used today. Torque converter models with adjustable stator blades, two or more turbines and stators, and integrated transmission functions were used but lock-up clutches had not yet been introduced.

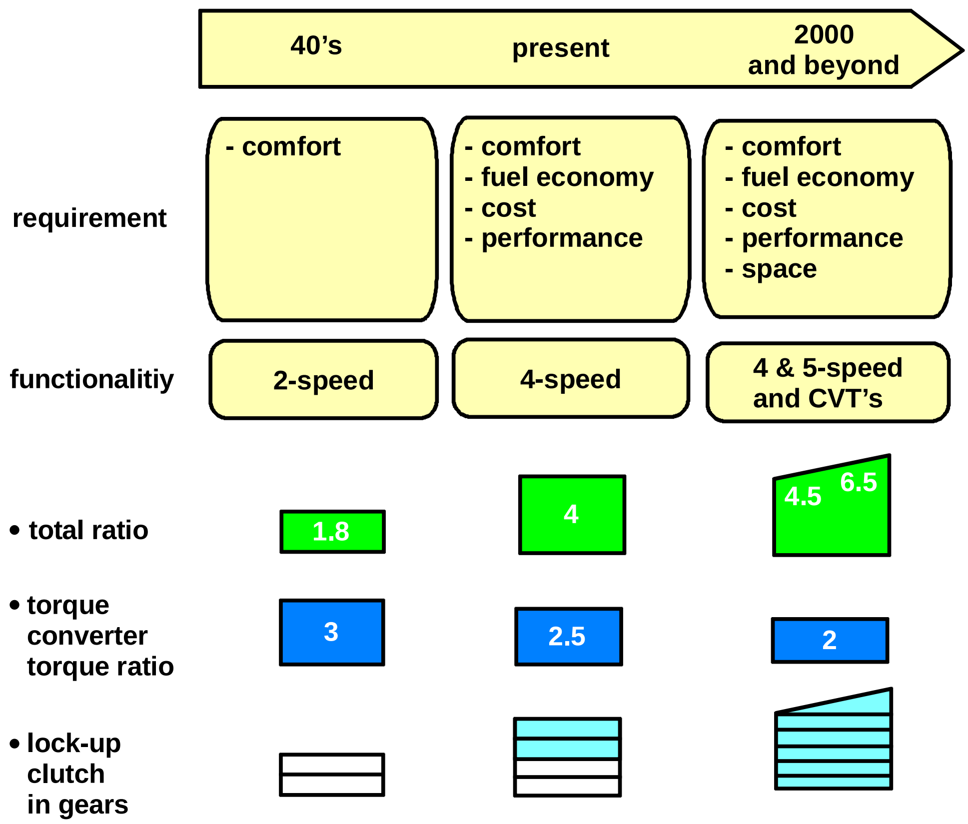


Figure 3: Global development of automatic transmissions

Stiffer torque converters had to be introduced in order to reduce losses. The large torque ratio was no longer possible and had to be compensated for by increasing the total gear ratio with more mechanical gears. In the 1970s, the 3-speed-automatic transmission (total ratio of approximately 2.5) was the most common model. Due to the oil crisis, lock-up clutches were introduced to eliminate torque converter slip losses, at least only in top gear and at high vehicle speeds. The requirements for cooling capability of the lock-up clutch and vibration isolation of the damper were accordingly insignificant.

The 4-speed automatic transmission is the state-of-the-art in the 1990s and provides a total gear ratio between 4 and 4.5. Today, 75% of these transmissions are electronically controlled, and in 2000, almost 100% will have this feature. Compared to the 1970s, the lock-up clutch operates in wider ranges. To reduce fuel consumption, the torque converter is normally locked in third and fourth gear, at engine speeds between 1100 rpm and 1700 rpm. The lock-up clutch operates partially with continuous slip, and is, in many cases, modulated, for example during gearshifts or after tip-ins.

Having reviewed the history and global development of automatic transmissions, we predict the following:

- The functionality (numbers of gears and total gear ratio) of automatic transmission will continue to increase.
- The duty cycle of the open torque converter will decrease further.
- The duty cycle of the lock-up clutch will increase further, either with slip or completely closed.

In addition to these global development trends, it is most likely that fuel costs will increase further and environmental awareness will enlarge. Both developments will require strenuous efforts to reduce fuel consumption. Likewise, the cost of torque converters will have to decline in the future. Space for the torque converter, especially in CVT- and front wheel drive applications, will be less than is currently available.

Figure 4 shows a summary of the global development trends and the resulting design targets for future torque converter systems.

global trends

- **increased functionality of automatic transmissions**
- **decreased duty cycle of torque converter**
- **increased duty cycle of lock-up clutch**

design targets

- **further improved fuel economy**
- **further improved performance/ acceleration**
- **less space and weight**
- **less cost**
- **good driveability and comfort**

Figure 4: Global development trends and design targets for future torque converter systems

The Control Strategy of the Lock-Up Clutch: The Linkage for Target Achievement

Concept, characteristics and design of the three system components of a torque converter - the lock-up clutch, the damper and the hydraulic circuit (torus) - are the result of the required control strategy for the lock-up clutch. The control strategy is the central link of the hardware elements; it determines the requirements for different operation ranges and, thus, the design of the individual components (figure 5).

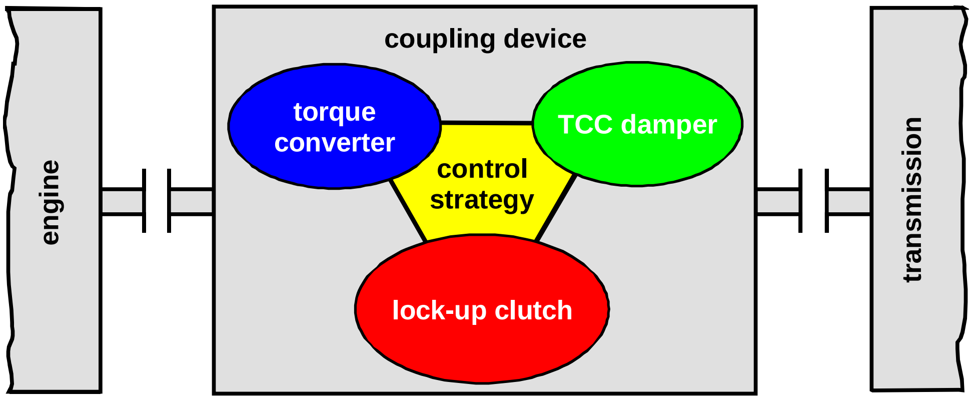


Figure 5: Total system of torque converter, lock-up clutch, damper and lock-up strategy

The appropriate lock-up control strategy is defined individually for each vehicle application by considering the vehicle functionality and the targets for fuel economy, performance, driveability and driver comfort. The total system characteristics determine the lock-up strategy and, hence, the requirements for the component designs.

One of the appropriate characteristics for the best lockup strategy is the total gear ratio of the transmission. Based on the state-of-the-art technology and the consideration of the future development of the automatic transmissions and CVT's, the control strategies can be grouped according to figure 6.

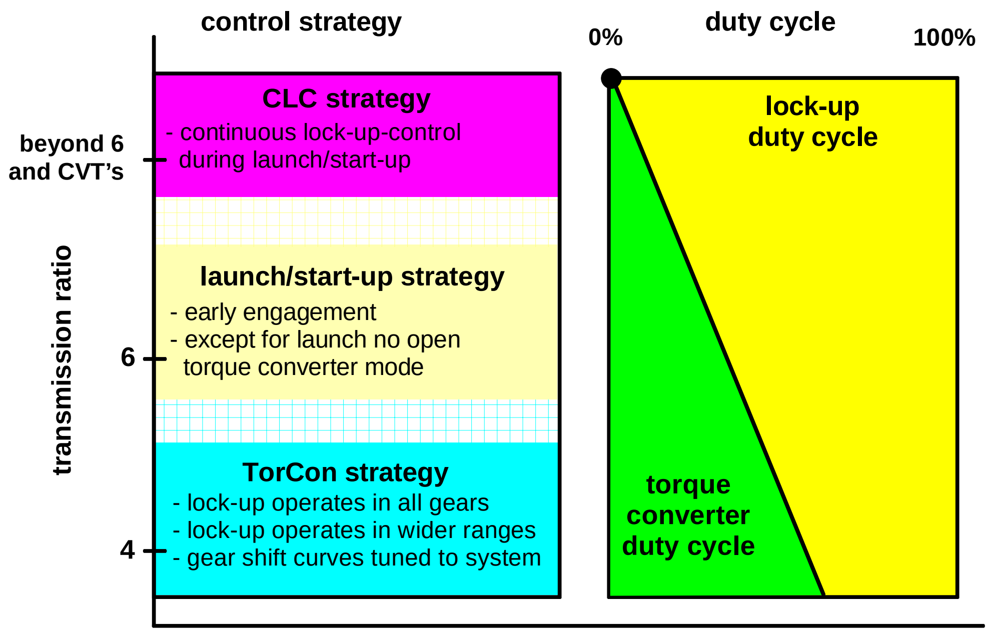


Figure 6: Lock-up control strategies

The LuK-TorCon-Strategy is appropriate for transmissions with a total ratio of 4 to 4.5. To reduce fuel consumption and to improve performance, the lock-up clutch operates

- in all gears and
- over large speed ranges (down to 900 rpm)

In order to ensure sufficient noise reduction in this low rpm-range, at high loading condition, or to avoid engine lugging, a large amount of slip is necessary during lock-up. With the use of a high performance damper, the noise level can be reduced further or the amount of slip can be lowered. The loss due to the high amount of slip can generate excessive heat and temperature build-up at the friction surface of the lock-up clutch. The high temperature causes ATF-oil and friction material degradation. The result is falling friction coefficient with increasing slip speed, which leads to shudder and possible transmission failure. As a counter measure, the lock-up clutch must have a high cooling capacity. The LuK high performance lock-up clutches provide this design feature.

Because the Torque Converter is still partially open with the TorCon Strategy, the appropriate torque converter characteristic must still be selected in order to optimize the system. Loose torque converters with a high stall torque ratio and a high stall speed create less idle losses and provide better acceleration when compared with stiff torque converters.

The engine warm-up time is also reduced and, thus, the emission level is lower. However, with a loose torque converter there are excessive losses generated in some operation ranges. As a counter measure, the lock-up of a loose torque converter must take place very soon. This was anticipated in the LuK-TorCon-System and the hardware is designed to permit this.

Figure 7 shows a comparison between a loose and a stiff torque converter when equipped with a conventional lock-up strategy and with the TorCon Strategy individually.

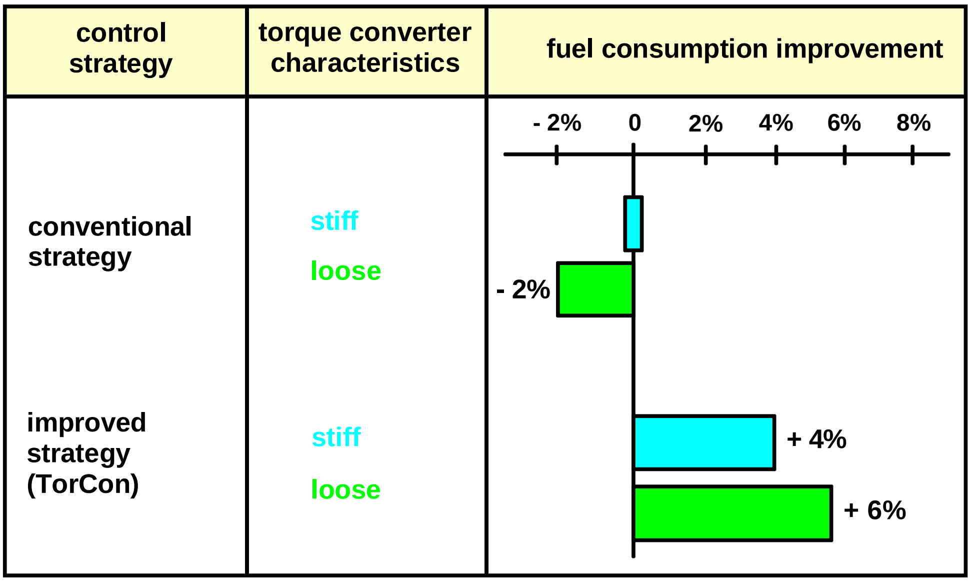


Figure 7: Influence of the torque converter characteristic and the control strategy on fuel consumption (simulation). Example for a 4-step transmission.

With respect to fuel economy, one can say:

- When using a conventional lock-up control strategy, the stiff torque converter is more advantageous than the loose one.
- When using the TorCon Strategy, the loose one is better.
- The influence of the control strategy on fuel economy is higher than the influence of the torque converter characteristic.
- When combining a loose torque converter with the TorCon Strategy, it is in some applications possible to achieve an improvement in fuel economy by 5 to 10% over the combination of a stiff torque converter with a conventional lock-up control strategy.

Similar conclusions can be made, if evaluating the vehicle performance.

With increasing transmission total gear ratio (>5), the selection of the torque converter characteristic (loose/stiff, stall torque ratio and peak efficiency, etc.) becomes less relevant for

- vehicle performance and
- fuel economy.

The reason for this is that there is always an operating condition with a closed torque converter for transmission types with large total ratios and number of transmission speeds that is better with respect to fuel economy or tractive force than an open torque converter. Opening the torque converter makes neither technical sense, nor is required for comfort. The torque converter serves mainly as a start-up element for a comfortable launch.

With this in mind, the conceptual layout of the torque converter has to focus more on the required installation envelope and cost optimization rather than on the traditional criteria, like peak efficiency, stall torque ratio and loose/stiff characteristic. With respect to the torque converter characteristic the function K-factor versus speed ratio and the position of the coupling point are getting most important.

LuK has developed a super squashed torque converter, which is axially 45% smaller when compared with conventional round torque converter designs. This was done without any performance tradeoff by employing the appropriate combination of the lockup clutch and control strategy. Figure 8 compares fuel economy and acceleration between the conventional round torus with the LuK super squashed torus in a CVT- application as an example.

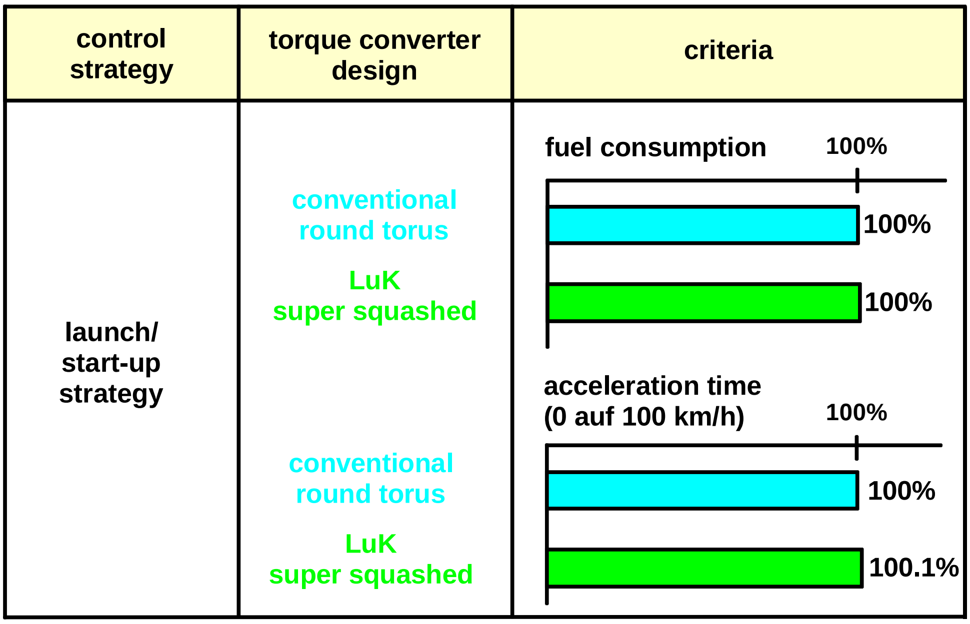


Figure 8: Influence of torque converter design concept on performance and fuel economy. Example for a CVT application.

For CVT-applications, it is even possible to think of a control strategy, which operates continuously without defined shifting points of the lock-up clutch. With a CLC (Continuos lock-up control) strategy, the clutch is activated by releasing the brake when the vehicle is not moving. The clutch is partially applied with a minimum torque capacity of about 10 Nm. The clutch apply increases continuously from vehicle launch until the transmission ratio changes (figure 6). An open torque converter mode during driving is not planned. The advantage of such a system is clearly the improved comfort (no shifting of the lock-up during driving). This strategy requires a larger cooling capacity and a modified cooling system to compensate for the losses in the clutch. This project is now in the initial development stage at LuK.

Based on the presented control strategies, the simulation results of the fuel economy and performance and the development trends, one can summarize quantitatively the development targets for the torque converter system as illustrated in figure 9.

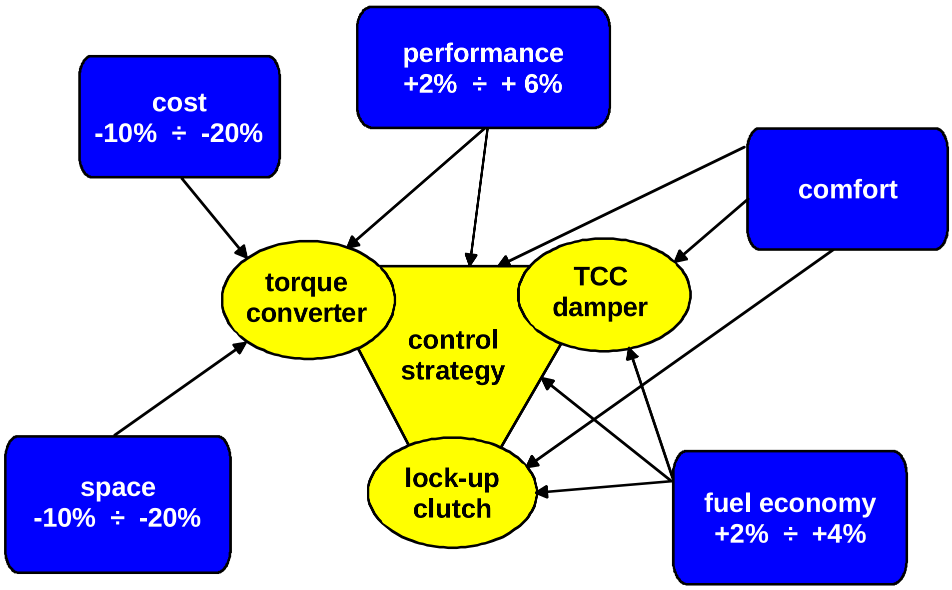
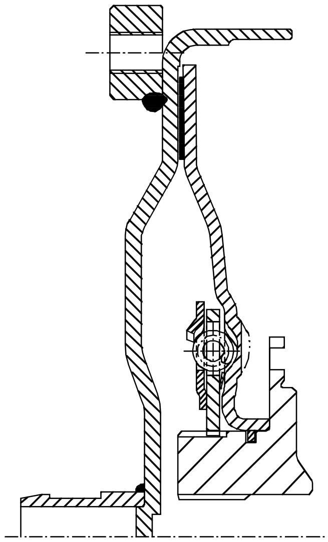


Figure 9: Design targets for the total system consisting of: Torque converter, lock-up clutch, damper and control strategy

Hardware Components

The High Performance Lock-Up Clutch

The common basis of the previously mentioned lock-up strategies is the increased requirement on the clutch functionality. In addition to the clutch torque capacity, both the cooling capability and the clutch controllability need to be taken into consideration.



design targets:

- **sufficient clutch capacity**
- **sufficient cooling capability**
- **good controllability**
  - smooth engagements
  - neutral behavior in drive and coast
  - constant and short response time
  - torque increase with increased slip speed (positive friction coefficient gradient)

Figure 10: Requirements of the high performance lock-up clutch

Torque Capacity

The permissible facing unit pressure of the facing material predominantly determines torque capacity of the lock-up clutch. The differential pressure across the piston plate and the active surface area determines the facing unit pressure. The active surface area is dependent on the mechanical and thermal deformation and the manufacturing tolerances of both the piston plate and the cover. Therefore, the nominal facing area is less important. A more detailed description of this mechanism has been given from LuK already in the 4th LuK Symposium 1994 [1] and in other publications, like [2], showing the substantial experience which already exists at LuK. Depending on the specific application and the operating conditions LuK can offer various lock-up concepts; flat single plates (Figure 10), conical lock-up concepts (Figure 26), twin-plate lock-up concepts (Figure 25) as well as multi-plate lock-up concepts.

Controllability

The lock-up engagement can be divided into three phases:

- Shift phase from open torque converter mode to lock-up mode
- Contact phase (piston plate touches cover)
- Torque build-up phase

It should be possible to engage the clutch with a minimum amount of volume flow. This can be accomplished with a low resistance to applying the piston plate and with a small gap between piston plate and cover, so that a small volume flow already creates sufficient, differential pressure to overcome the apply resistance and closes the gap. A large volume flow would cause a sudden torque build-up during the engagement, since the kinetic energy of the volume flow would be converted to pressure energy. The hydraulic resistance between piston plate and cover and piston plate and turbine hub is also important. It is favorable to keep it at a minimum. With the help of numerical analysis it was possible to optimize the shape of the cover, piston plate and turbine hub for the described purpose.

For the contact phase it is important that the grooves don't cause a self-energizing effect. Experimental development lead to an optimized groove design, which will cause a neutral engagement in drive as well as in coast. In order to achieve good torque controllability a fine tuned torque capacity and a positive friction coefficient gradient versus slip speed are necessary.

Cooling Capacity

Sufficient cooling capacity of the lock-up clutch is very important. The power losses that are generated at the friction surface can cause high peak temperatures, which can destroy the oil additives and, in interaction with the mechanical shear load, split the molecule chains. This causes a negative friction gradient, which can lead to shudder and finally can cause failure of the transmission.

The thermal loading of the high performance lock-up clutch is mainly dependent on the lock-up strategy. The strategy will determine the maximum power loss and the magnitude of the total energy, to be absorbed at the friction surface. The following is valid:

- The lower the rpm at which the lock-up engages (lugging limits), the higher is the needed slip to avoid noise and, therefore, the higher the power losses.
- The lower the gear, the more critical are tip-in reactions. In Figure 11 a typical tip-in event is plotted. Immediately after the tip-in the lock-up clutch is slipping, which is meant to eliminate the usual and annoying surging vibration. The plotted event appears to be comfortable. However, this leads to high power losses.

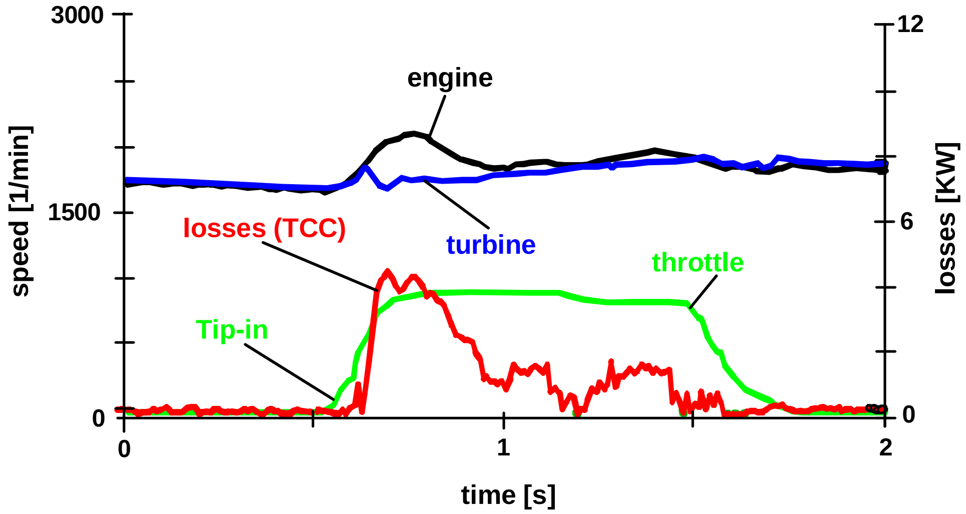


Figure 11: Thermal loading of the lock-up clutch during tip-in

The lower the gear and the lower the lugging limits, the more comfortable, and with this longer, the engagement of the clutch needs to be. The longer the engagement, the higher is the total energy absorbed. Figure 12 shows a comfortable engagement. The maximum power losses are 7 kW during this non-noticeable engagement.

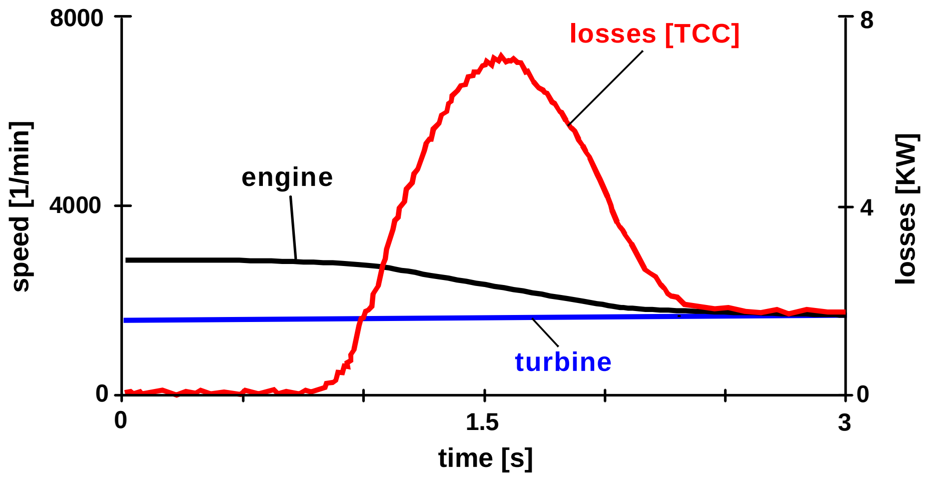


Figure 12: Thermal loading of the lock-up clutch during a “non-noticeable“ engagement

What are the differences between LuK's high performance lock-up clutches and conventional lock-up clutches, and why is the allowable power loss higher? The answer is:

- Well tuned parts.
- Usage of conical shaped piston plates with high stiffness in the friction surface area (Figure 25) or a twin-plate design (Figure 24) for even higher power losses.
- Optimized geometry of the cooling groove in the friction material.

To optimize and fine-tune the components the following interactions are important:

1. The development of the temperature on the friction surface is dependent on the local specific power loss (the product of the local facing unit pressure on the friction surface and the slip) and the development of the power loss function over time (see figure 13). Especially for engagements, the criterion for optimization is the specific power loss on the friction surface and not the local facing unit pressure.
2. When the facing unit pressure on the friction surface is calculated, not only the elastic deformation of cover and piston plate due to apply pressure should be taken into consideration. The thermal deformation also must be considered. This alone can raise the facing unit pressure by 20%.
3. Converters with 2 pass systems should have the friction material on the piston plate, while 3 pass systems should have the friction material on the cover.
4. The development of the interface temperatures during lock-up engagements is very different when compared to continuous slip conditions.

In order to analyze the temperature development LuK uses a special software package which incorporates the mechanical and thermal deformation of the piston plate, the material properties of the friction material and the heat transfer coefficient between steel/oil and steel/air which locally can be very different.

The results from a computer simulation performed with this software package are compared with measured data of a rotating test part in figure 13. The figure shows the temperature rise on the friction surface during an engagement. The maximum surface temperature is reached towards the end of the engagement in the middle of the friction surface even though neither the facing unit pressure nor the specific power loss is at maximum at this location and time.

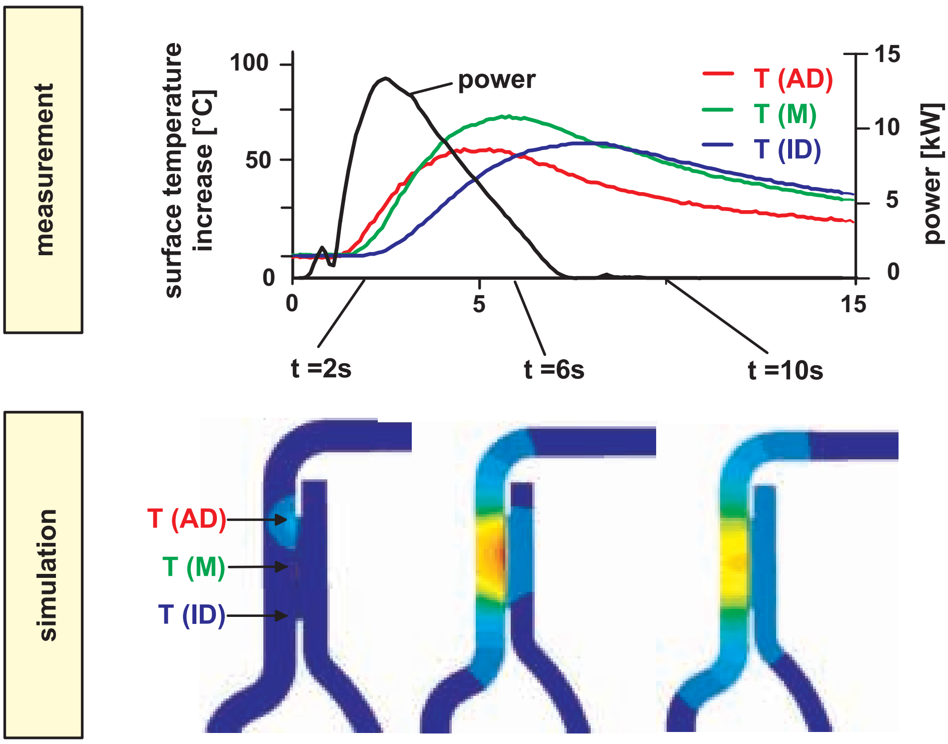


Figure 13: Temperature development at the friction surface during an engagement

Due to these results new development directions and design guidelines could be established, which differ significantly from those that only consider facing unit pressure distributions.

The second important design part of the LuK high performance lock-up clutch is the cooling groove pattern, which shape allows an ideal optimization of volume flow and hydraulic resistance. This shape provides an increased heat transfer coefficient by a factor of 4 when compared to conventional grooves. In addition the LuK groove design exhibits a neutral behavior with respect to friction coefficient and the torque build-up during drive and coast condition. Figure 14 shows the allowable specific power loss of facings with LuK groove designs. It becomes apparent that in continuous slip applications the allowable power loss can be increased by 120%. During shorter slip events the increase in allowable power loss is less.

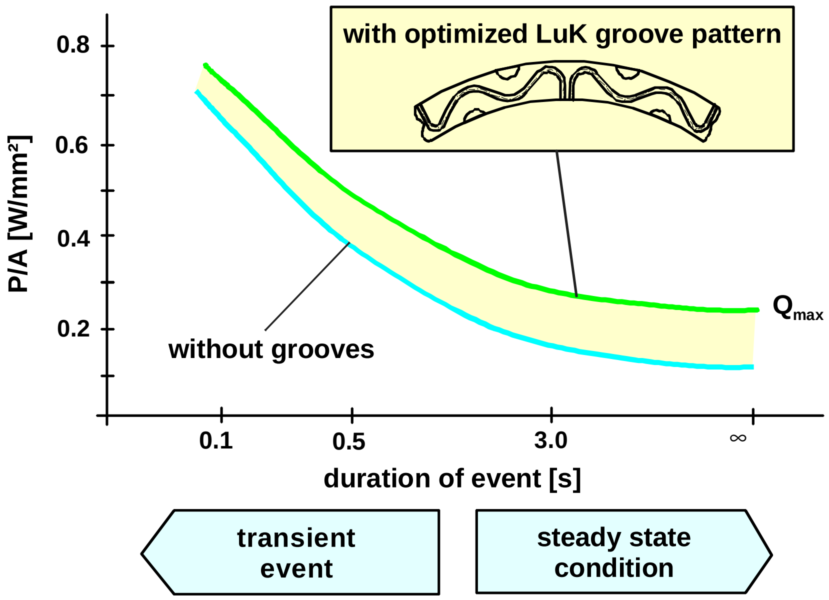


Figure 14: Permissible specific power loss at the friction surface

Figure 15 shows the absolute permissible values of the power losses, if the specific power losses from figure 14 are transformed to a real part. It should be noted, that the pictured lock-up concepts exhibit a strong design-dependent influence with respect to their cooling capacity. The different deformations of the parts and different realistic manufacturing tolerances of each design concept can explain this. The lock-up concept with the conical design of the piston plate exhibits the largest specific cooling capacity.

With the described lock-up concepts the initially mentioned control strategy problems can be solved. The risk of oil degradation does not occur.

Similar care must be taken when selecting the proper facing material for the lock-up clutch. The following criteria are very important:

- Increasing friction coefficient over increasing slip speed
- Good bonding and grooving capabilities
- High mechanical capacity
- Low wear rate
- High thermal capacity
- High level of friction coefficient

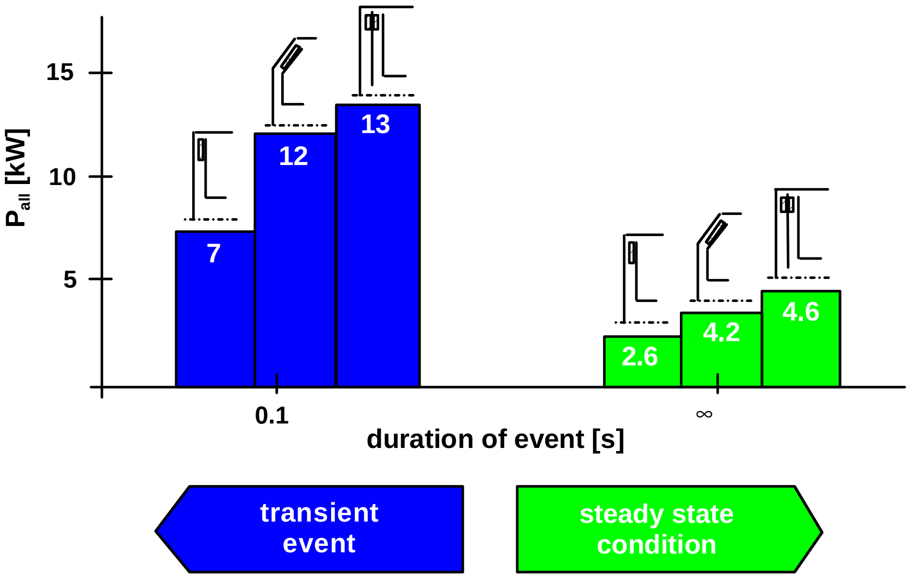


Figure 15: Permissible absolute power losses of high performance lock-up clutches with optimized groove pattern and maximum volume flow

The Damper

LuK produces approximately 3 million torque converter clutches per year, world wide, of various configurations. LuK has a Two-Mode Damper in production since 1996. This innovative damper concept offers many advantages for certain engines and drivetrains.

If a four-inertia model is used to describe the vibration modes of a drivetrain (engine, turbine, transmission and vehicle, like described in figure 15), the third mode is the transmission inertia swinging relative to the vehicle and turbine inertia. The largest vibration angle occurs in the transmission input shaft. The vibration angle in the damper is small in comparison. This mode can cause unacceptable noise, like in this case the boom noise, if the natural frequency of this vibration mode lies in the critical engine speed for the vehicle from 900 rpm to 2000 rpm. A reduction of the damper rate offers no improvement because of the small vibration angle. A long travel damper is therefore no solution for this problem.

A reduction of the transmission input shaft stiffness has a positive effect on this vibration problem, but is usually not possible because of the reduced input shaft strength.

LuK has developed a damper to solve this problem. The turbine is rotationally fixed to the torque converter clutch piston plate and the damper is in series with the transmission input shaft (figure 16).

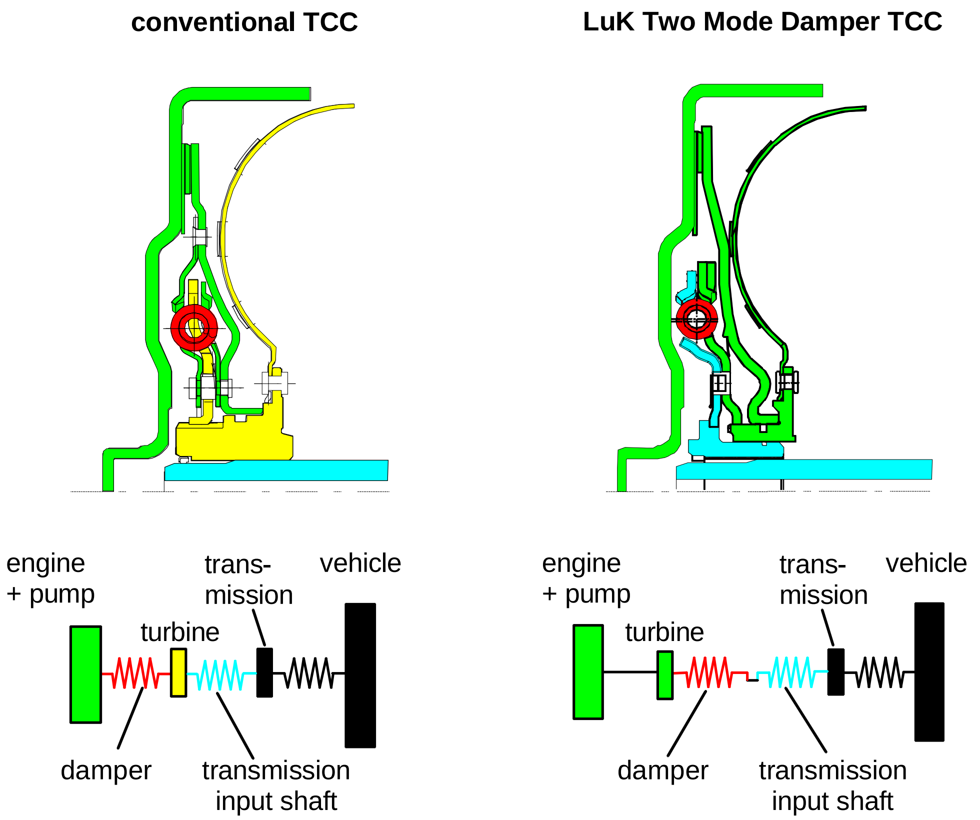


Figure 16: Principle of the LuK Two-Mode Damper compared to a conventional lock-up damper concept

The LuK Two-Mode Damper functions as follows:

- The damper and the transmission input shaft are attached in series which results in a very soft rotational stiffness of the total system (this was the original goal).
- The turbine is rotationally attached to the torque converter housing when the clutch is applied. This adds turbine inertia to the converter housing and engine inertia, thereby reducing the vibrations transmitted from the engine to the drivetrain.
- Combining the engine, torque converter and turbine inertia eliminates one degree of freedom from the drivetrain.

The third vibration mode can always cause noise problems if the natural frequency of this vibration mode lies in the critical engine speed for the vehicle from 900 rpm to 2000 rpm. Simulations with typical drivetrains show that this can be critical in front wheel drive vehicles with 8- and 10-cylinder engines and rear wheel drive vehicles with 6- and 8-cylinder engines. Front wheel drive vehicles with 8- and 10-cylinder engines are unusual. Rear wheel drive vehicles with 6- and 8-cylinder engines are almost the rule. The Two-Mode Damper offers clear advantages for these vehicles.

By eliminating the otherwise critical third natural frequency, the torque converter clutch can be closed at lower engine speeds without any loss of comfort. The fuel consumption of vehicle types considered (figure 17) could be reduced up to 6%, using a LuK Two-Mode Damper.

The drivetrain natural frequencies for various rear wheel drive vehicles are represented in figure 17. These vehicles are all equipped with the same torque converter and the same automatic transmission. Every point in the diagram indicates a drivetrain resonance point and thus a potential vibration or noise problem. The resonance frequencies that occur with a conventional damper are compared with those that occur with the Two-Mode Damper. One notices that only one resonance frequency occurs with the Two-Mode Damper in the critical area, and this resonance is uncritical with low damper friction. Also important is the conventional damper with 15 Nm/° has a much lower damper rate than the Two-Mode Damper with 45 Nm/°. This means that a smaller spring volume can be used with the Two-Mode Damper.

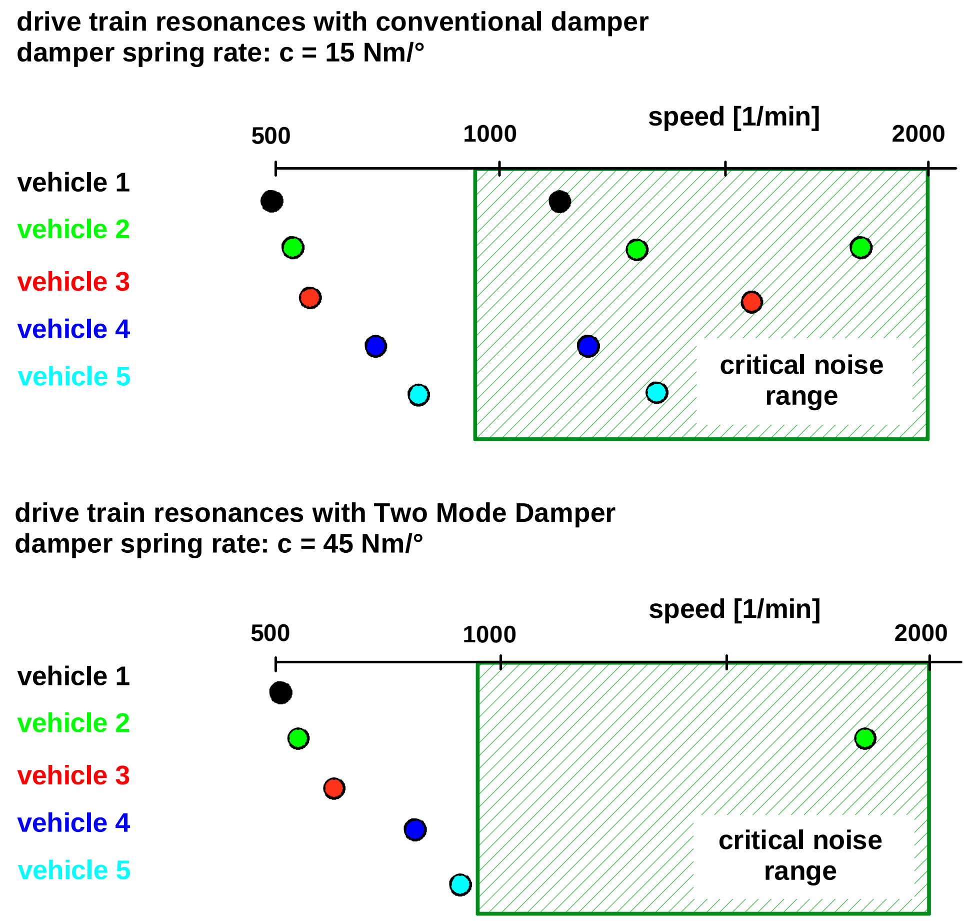
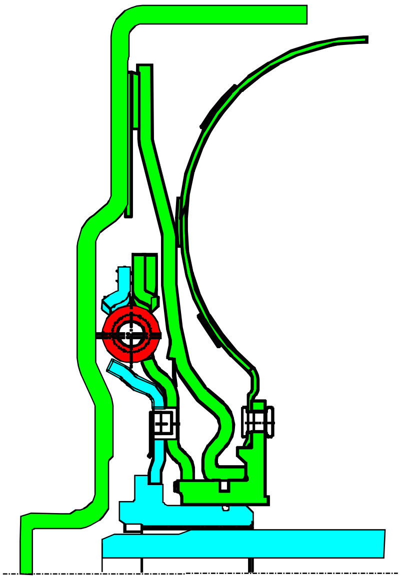


Figure 17: Drivetrain resonance with a conventional damper and the LuK Two-Mode Damper

Another important point is that the torque flow is through the Two-Mode Damper also when operating the converter in open mode. This gives additional vibration isolation in the open mode and provides an advantage for some unique vehicle types, for example, in vehicles with direct fuel injected engines or with cylinder shut-off which can have boom problems in the open mode.

The LuK Two-Mode is also appropriate for CVT transmissions because of the inertia distribution in those applications.

The advantages of the Two-Mode Damper are summarized in figure 18.



advantages

- **better fuel economy due to reduced lugging limits**
- **improved NVH**
- **also active in open torque converter mode**

Figure 18: Advantages of the Two-Mode Damper

The Squashed Torque Converter Torus

As mentioned in the beginning of this report, the torque converter for the future should be smaller, lighter and less expensive (figure 9).

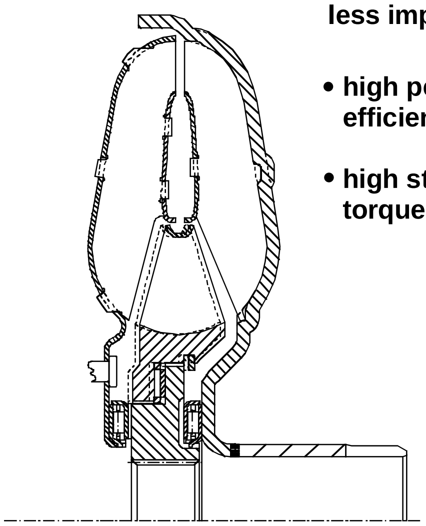
A geometrical downsizing of the hydrodynamic torque converter torus without simultaneously changing the blading offers little flexibility with regard to the torque converter characteristic. Moreover, such converters cavitate or have an unacceptable K-Factor over speed ratio characteristic curve.

Consequently, the best procedure is optimizing the pump, turbine and stator blade to the new geometrical ratios of an extreme squashed torque converter. Besides the avoidance of cavitation the attention has to be directed to the K-Factor characteristic versus slip speed.

- The ability of the torque converter to transmit a certain torque at a given speed level (torque capacity = 1/K) has to be sufficient in order not to increase the torque converter size (diameter).
- The reduction of idle losses requires a high K-Factor at stall. However the clutch engagement is more comfortable and can take place earlier if the coupling capacity (1/K at coupling point) is as high as possible. This also helps to reduce the thermal load of the lock up clutch during an engagement. Together, both criteria require the ratio of torque capacity at stall and coupling point to be high (flat K-Factor curve)

This development goal is supported by the fact, that the fuel consumption or performance (see chapter 2) no longer dictates the design of the innovative torque converter along with the lock-up control strategy. Important criteria for the classic torque converter design, such as peak efficiency or high torque ratio, are therefore of lesser importance. The development can be much more focused. Figure 19 summarizes requirements for the future torque converter.

design targets



less important

- **high peak efficiency**
- **high stall torque ratio**

more important

- **extremely squashed torus**
- **no cavitation**
- **sufficient low k-factor**
- **high coupling point**
- **flat k-factor-curve**
- **sufficient heat capacity**

Figure 19: Requirement for the innovative torque converter

LuK has developed a torque converter whose torus is up to 45% axially smaller compared to a conventional round torus.

The torque capacity over speed ratio is a similar curve gained with a conventional, round torus. Without the use of suitable tools, such as powerful software to numerically calculate fluid flow in the circuit and the blade geometry, as well as Rapid Prototyping for fast and cost effective development of geometrically complex prototypes, this development would not have been possible.

The flow phenomena in the torque converter are critical especially at stall condition and at lower speed ratios. The energy builds up in the pump, as well as the energy dissipation in the turbine and also the redirection of the fluid in the stator (change of impulse) are maximum at this operating point. Simply stated the load acting on each fluid particle is the highest at this operating point. This principle is valid for each torque converter type. These relationships get worse tremendously if the fluid flow is unnecessary accelerated in quantity or direction caused by the inertia of the fluid particles. Unless geometrical improvements are made, those problems will occur in a squashed torque converter.

Figure 20 shows the fluid flow field inside the torque converter of an initial and of an optimized design. The initial torque converter design has a tear drop shape for the meridian cross section. This tear drop shape keeps the cross sectional flow area constant in order to avoid undesired accelerations or decelerations of the fluid. In spite of this, the velocity field shows a large separation area starting at the leading edge of the turbine and along the entire inner shell (core ring). This separation area considerably narrows the fluid flow channel and leads to a restriction of the circulating volume flow and therefore, reduces the torque capacity. Furthermore, there is an exchange zone where the energy from the 'healthy' flow is transferred to the circulating flow in the separation area. This distinct loss region can be clearly shown with the help of the turbulence energy (figure 20). This energy is almost fully dissipated in the separation area. In the optimized torus design, adjusting the design of the core and the blades reduced the separation area and the resulting energy losses. This change contributes considerably to the advantageous torque capacity characteristic.

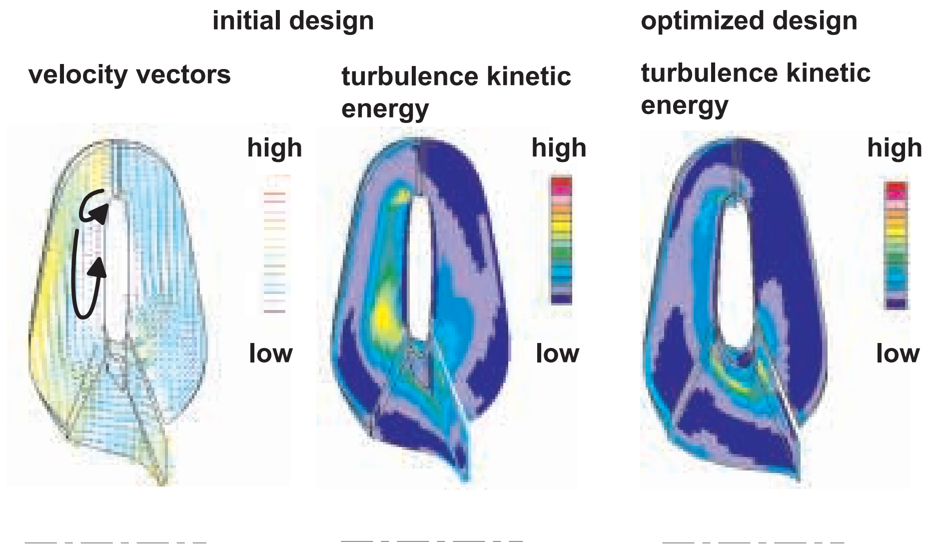


Figure 20: Optimized turbine; flow condition at stall

The critical and most difficult challenge in designing extremely squashed torque converters is avoiding fluid cavitation. Cavitation can be explained as the build-up and subsequent collapsing of vapor bubbles in the fluid flow.

If the absolute static pressure in the flow drops under the vapor pressure, vapor pockets grow and are dragged along in the flow. These cavities contract the cross sectional flow area and throttle the circulating volume flow. The torque capacity and the efficiency drop drastically. In areas of higher static pressure these vapor pockets implode. This collapsing of vapor pockets happens very fast, so the fluid particles impact on the channel surface with a very high velocity (jet-impact), which can result in mechanical destruction of the channel surfaces.

In torque converters, the region of lowest pressure is the suction side of the pump, consequently, the area between the pump and the stator. The profile losses in the stator have to be reduced to keep the static pressure at the stator outlet as high as possible. Figure 21 shows the fluid flow conditions in the stator passage for the initial design. One can identify a distinct separation region at the suction side of the stator blade in the cylindrical cross section. The flow field behind the trailing edge indicates strong turbulence. Both conditions lead to a drastic pressure drop in the stator passage and in the region behind the trailing edge of the stator blade (figure 21 and figure 22). Because of cavitation at the pump suction side, the initial design could not be run at high speed and at speed ratios less than 0.5.

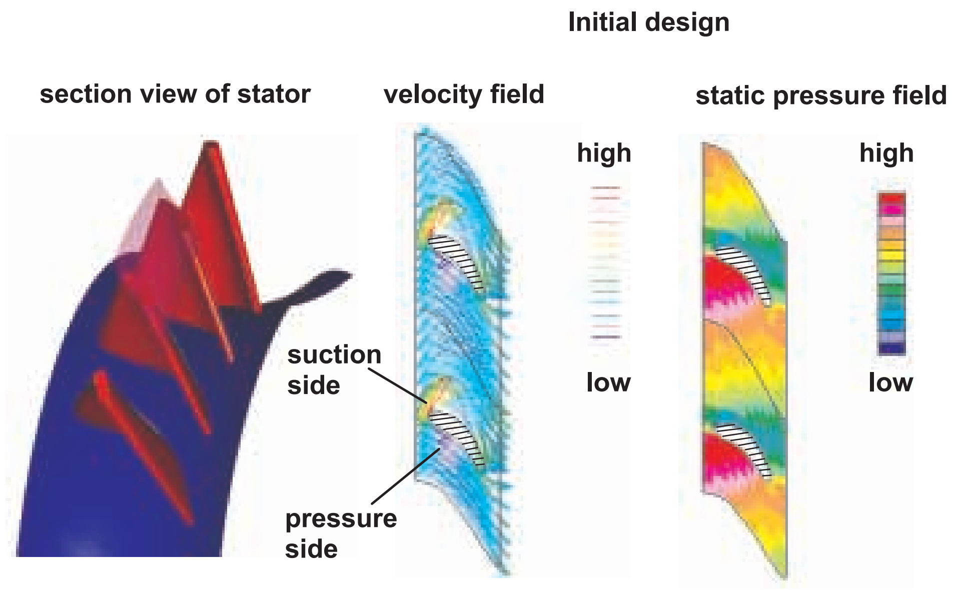


Figure 21: Flow field and static pressure field of a non-optimized stator blade; flow condition at stall

Figure 22 shows the velocity distribution and the static pressure field of the optimized stator design. The separation region at the suction side of the stator blade is much smaller than in the initial design. The flow field behind the stator is almost turbulence free. Both conditions result in a reduction of the pressure drop in the stator and an increase of the relative pressure in the pump inlet region.

As a measure for the quality of the energy transfer the total pressure field of the initial and optimized design are compared in figure 23. The comparison of the loss factors of both profiles indicates more than a 60% reduction of the profile losses in the optimized stator.

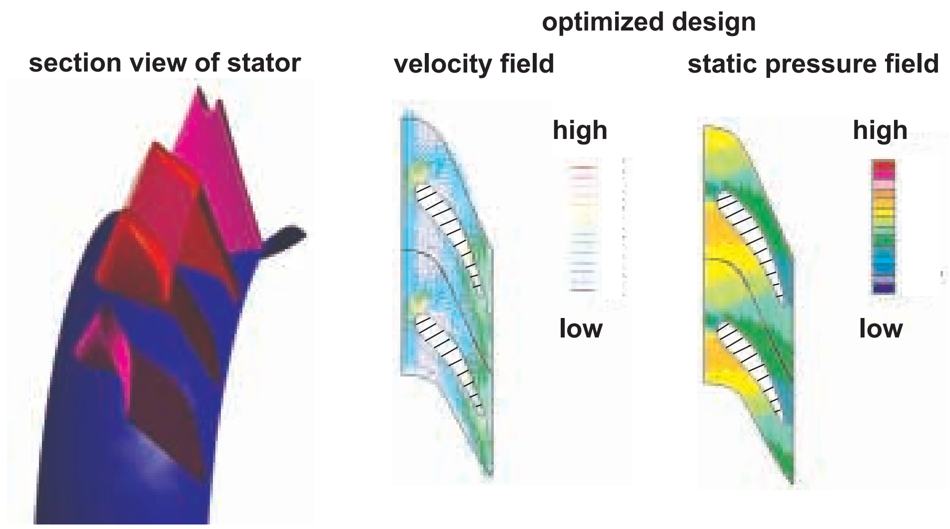


Figure 22: Flow field and static pressure field of an optimized stator blade; flow condition at stall

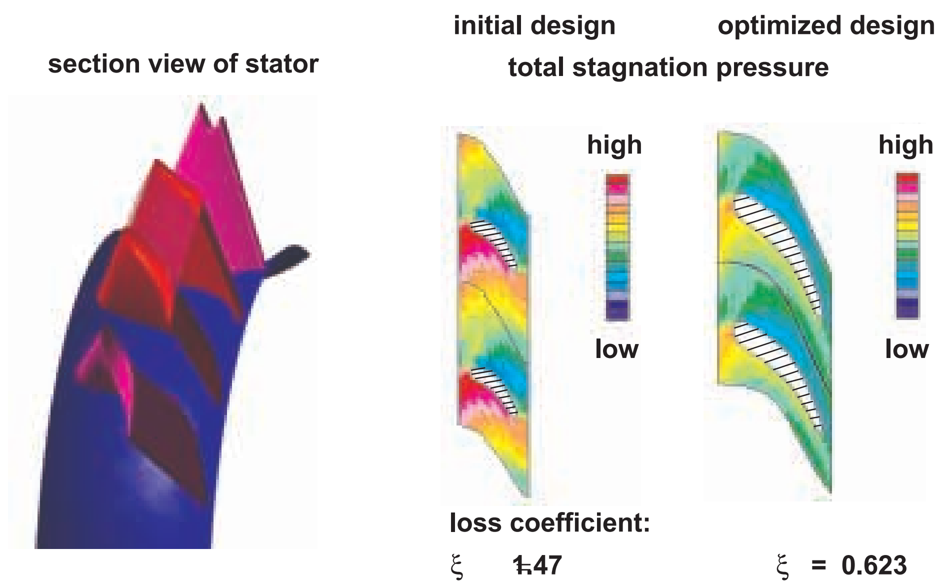


Figure 23: Comparison of profile losses; initial and optimized stator design; flow condition at stall

The optimized design of the extremely squashed torque converter shows no indication of cavitation. The quality of energy transfer in this case can be compared to a conventional, round torus torque converter (figure 24). Disadvantages regarding functionality of the squashed torque converter in combination with lock-up control strategy are not expected.

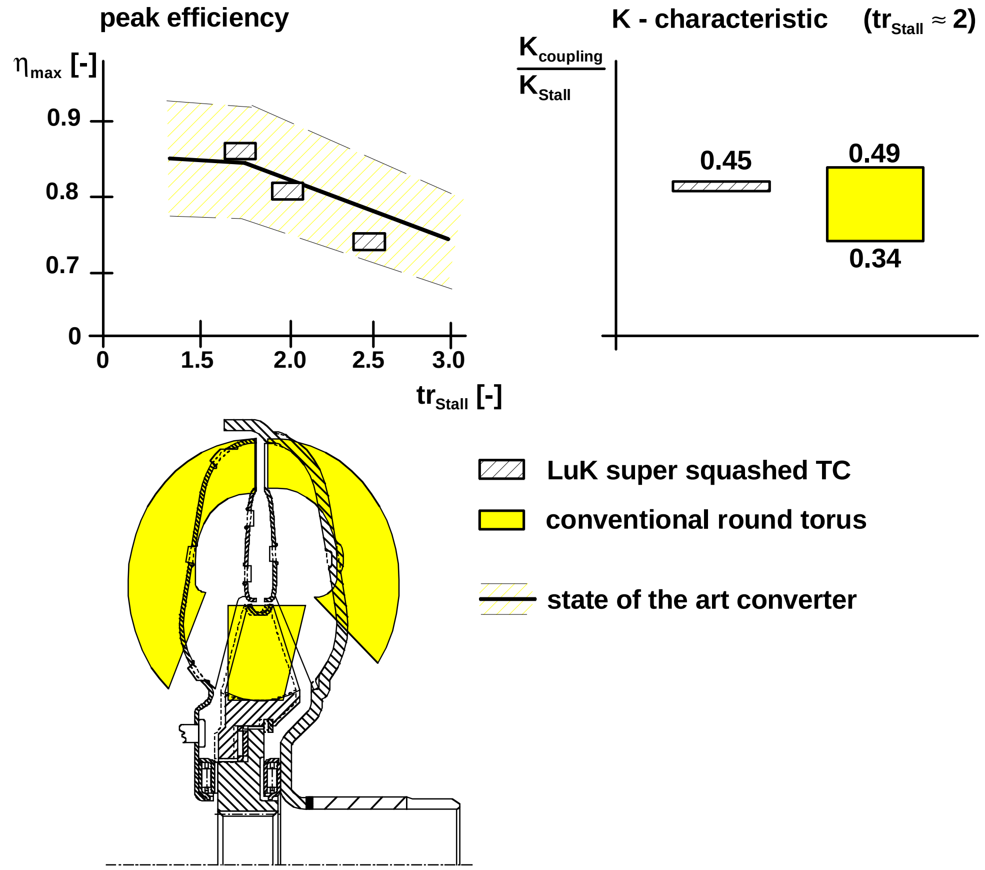


Figure 24: Comparison of quality of energy transfer, conventional round torque converter and LuK super squashed torque converter

The weight and envelope advantage of the extremely squashed torque converter is between 20 and 26% when compared to the conventional round torque converter. This advantage depends on the configuration and selection of the torque converter clutch concept (figure 25 and 26). The space gained can be used for the transmission design, or the vehicle drive train can be shortened.

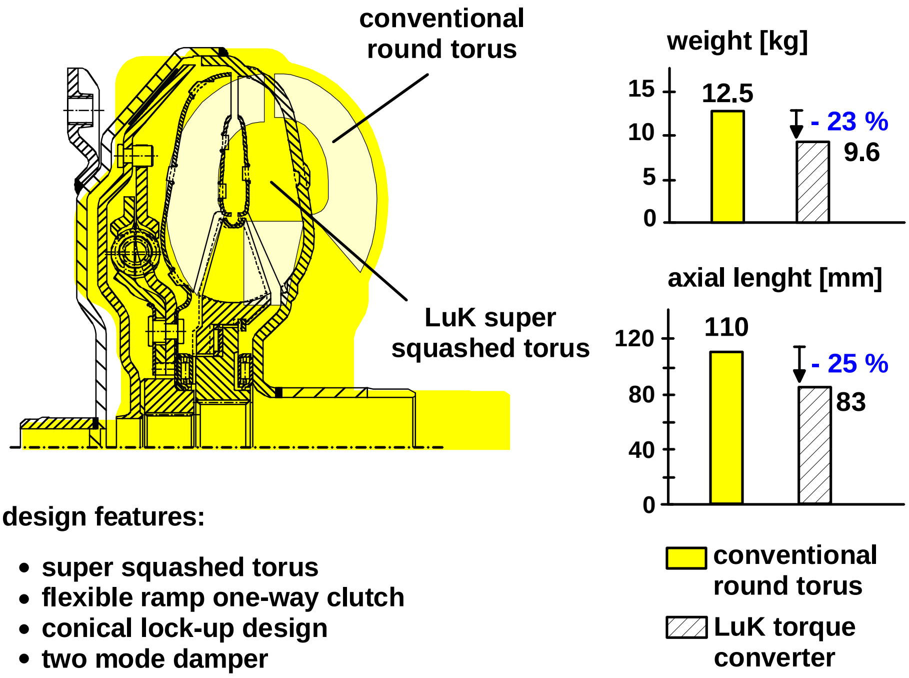


Figure 25: Comparison of required installation envelope for 260 mm size: Conventional, round torus and LuK super squashed torus with Two-Mode Damper and conical lock-up clutch.

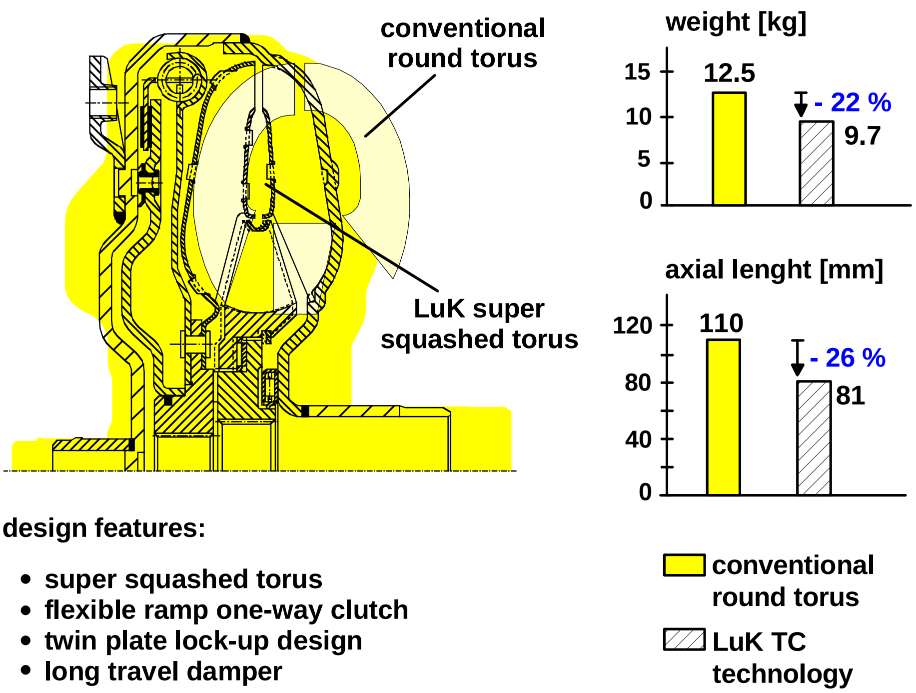


Figure 26: Comparison of required installation envelope for 260 mm size: Conventional, round torus and LuK super squashed torus with long travel damper and twin plate lock-up clutch.

LuK has developed many torque converter design features in order to reduce manufacturing costs:

- Low material use due to extreme squashed torus
- Use of axial one way clutch
- Phenolic stator with integrated features
- Stamped or pierced slots for blade fixturing
- Reduced number of blades
- Use of innovative manufacturing processes
- Use of powder metal hubs
- Reduction of machining operations

Summary

Modern drive train development, the desire to increase performance and driver's comfort, the demand to improve fuel economy and decrease emissions, and the continuously declining space require a new approach to torque converter concept, layout and design.

The torque converter, with its three elements (lock-up clutch, damper and hydrodynamic circuit) has to be viewed as a part of the total system (figure 27). In this context, the control strategy for the lock-up clutch takes on an important role as the link between these elements. Only with a carefully tuned control strategy which takes into account the demands of each vehicle, and only with hardware components which meet the new requirements, can the entire potential of the torque converter system be fully used.

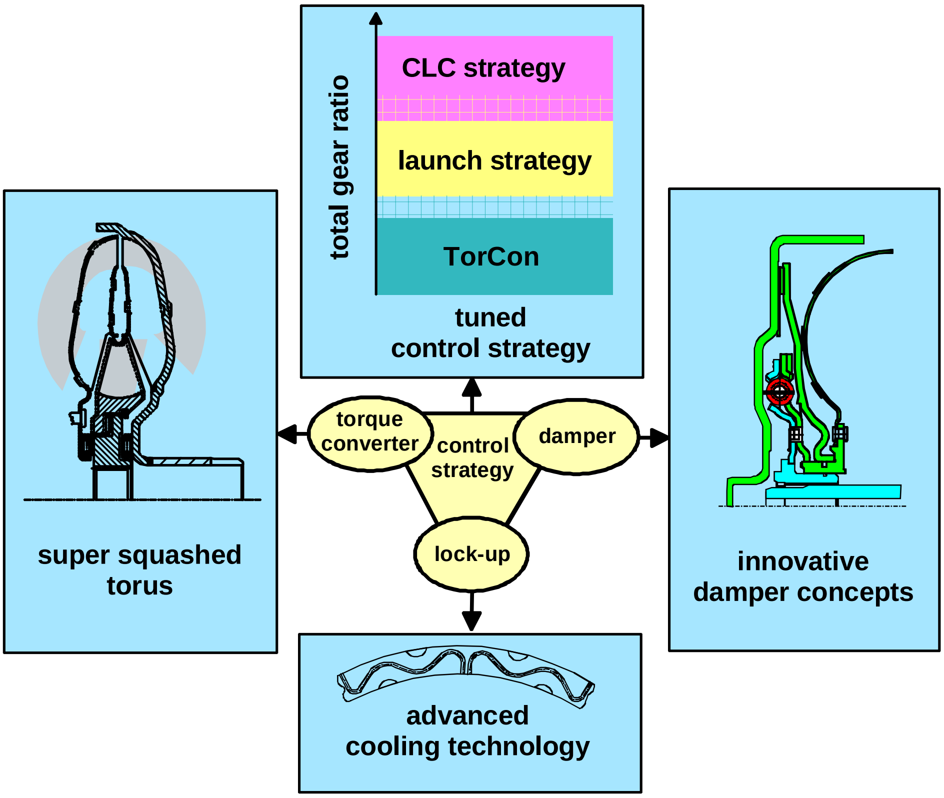


Figure 27: The Torque converter as a system – Summary

Using the full potential will allow for fuel economy savings and increase in performance of up to 10 %, depending on the application. The torque converter concepts developed by LuK make this possible. In addition to decreasing material costs, they provide tremendous savings in weight and space.

Literature

[1] R. Fischer, D. Otto, " Wandlerüberbrückungssysteme," 4th International LuK Symposium 1994 (in German), pp. 133.

[2] R. Fischer, "Das TorCon-System – Ein neues Wandlerüberbrückungs-Konzept als Beitrag zur Ökonomie und Fahrfreude," (in German) VDI Report No. 1175 pp. 301.

[3] D. Piper, " Automatic Transmissions – An American Perspective," VDI Report No. 1175, pp. 25.

CVT Development at LuK

Dr.-Ing. **Hartmut Faust**

Dr.-Ing. **André Linnenbrügger**

Introduction

At the last LuK Colloquium in 1994, as part of a comparison of various transmission systems [1], a prototype of a continuously variable transmission (CVT) was introduced. The major advantages of CVT are:

- Increased driving comfort due to smooth transmission ratio changes,
- Low fuel consumption due to large spread of gear ratios, and
- Excellent dynamics of movement.

Despite these advantages, which are particularly evident in larger engines, CVT is marketed for engines only up to approximately 150 Nm torque [2], although a 200 Nm [3] version recently became available in Japan.

In the interim, LuK has used the belt/chain principle to develop CVT components for torques up to and exceeding 300 Nm that are ready for mass production. This presentation will showcase some of the special features engineered into these components.

Functions of the Hydraulic Control System

Figure 1 compares the function of hydraulic control systems for 5-speed step automatics and CVT.

The hydraulic system must serve a start-up device and a reverse unit in both kinds of transmissions. In a 5-speed automatic, it is also responsible for shifting gears, which requires a total of approximately six solenoids and approximately 20 valves, depending on the design. In a CVT transmission, the only additional function is the adjustment of the clamping force between the pulley flanges and the chain and the adjustment of the CVT ratio. This type of design requires only three solenoids and nine valves.

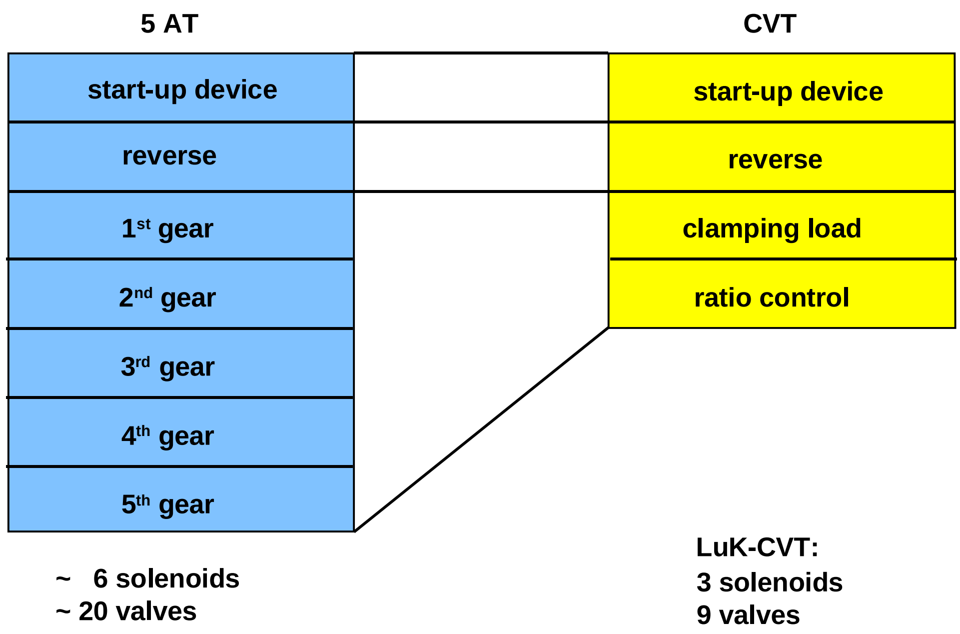


Figure 1: Comparison of hydraulic control systems for 5-speed step automatics and CVT

Using the general concepts outlined during this presentation, a compact CVT hydraulic control can be realized. A dual-stage torque sensor arranged directly in the power flux of the variable speed mechanism, where it can provide excellent functional safety, ensures very accurate, highly dynamic clamping control.

LuK is supervising both the development and the production of the hydraulic control system.

Clamping System Requirements

A whole range of factors affecting the entire drive train from the engine to the wheels influences the required clamping force (Figure 2).

- The engine torque, which is dependent on the driver and on electronic engine controls, is one factor.
- Converter torque multiplication and clutch or lock-up clutch controls cause additional torque to act on the system.
- Gear ratio is a factor, because it makes a difference whether torque is transmitted by the primary pulley in underdrive (UD) with a small effective radius or in overdrive (OD) with a large effective radius.
- The lowest occuring friction coefficient determines the required clamping force.
- The influence of the wheels on the torque to be transmitted by the variable speed mechanism is critical because torque can change suddenly and unpredictably, such as when a wheel leaves the road surface, or during the transition from an icy patch to normal road. An appropriately controlled clutch can make it possible to limit these influences.

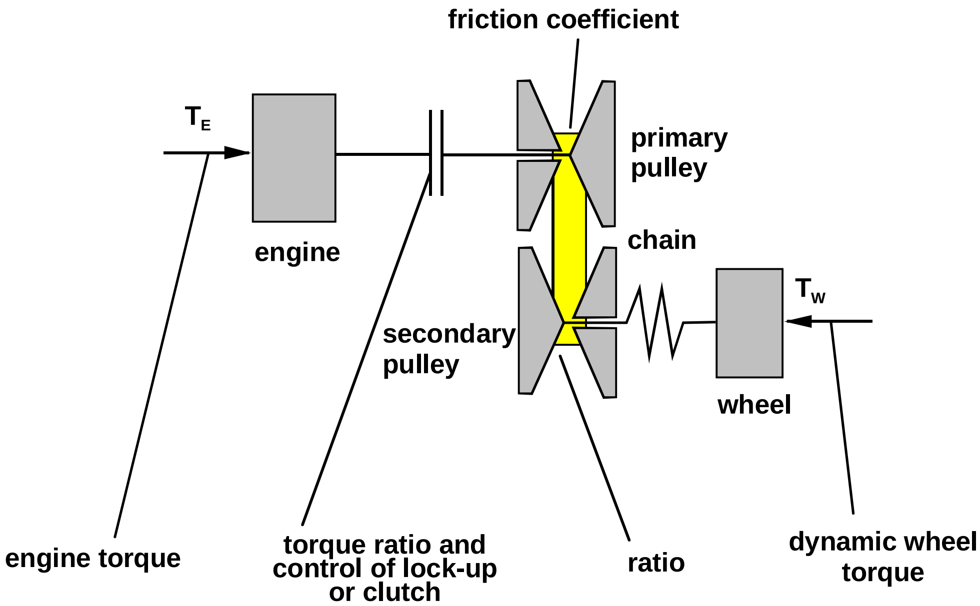


Figure 2: Influences on the required clamping force in the variable speed mechanism

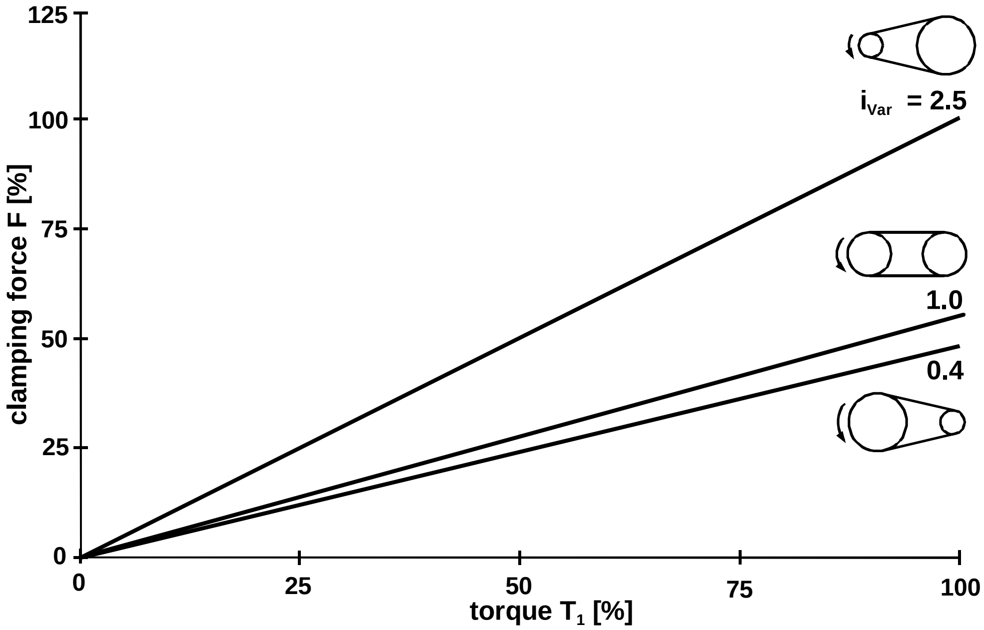


Figure 3a: Required minimum clamping force for various transmissions over torque

For a given coefficient of friction, the required minimum clamping force increases in a linear fashion as torque increases. This correlation is depicted for various transmissions in Figure 3a.

A full-load safety factor of approximately 25 % above the slip limit allows for fluctuations in the friction coefficient, as shown for the underdrive transmission in Figure 3b. This is intended to prevent slipping. Greater safety margins lead to unnecessarily high forces and lower efficiency. In conventional systems, the full safety factor is present even when torque is low [2], because the strength of unpredictable jolts is not dependent on the adjacent torque. Thus, systems incapable of quickly changing the clamping force with the aid of automatic pumping action require a constant reserve. Consequently, at 1/4 of the nominal torque, the excess clamping force increases to 100 % in relation to the adjacent torque. One can see the negative effects on the variable speed mechanism's efficiency and the effect of the unnecessarily high resulting system pressure on pump losses. This also results in unfavorable fuel consumption under partial loads.

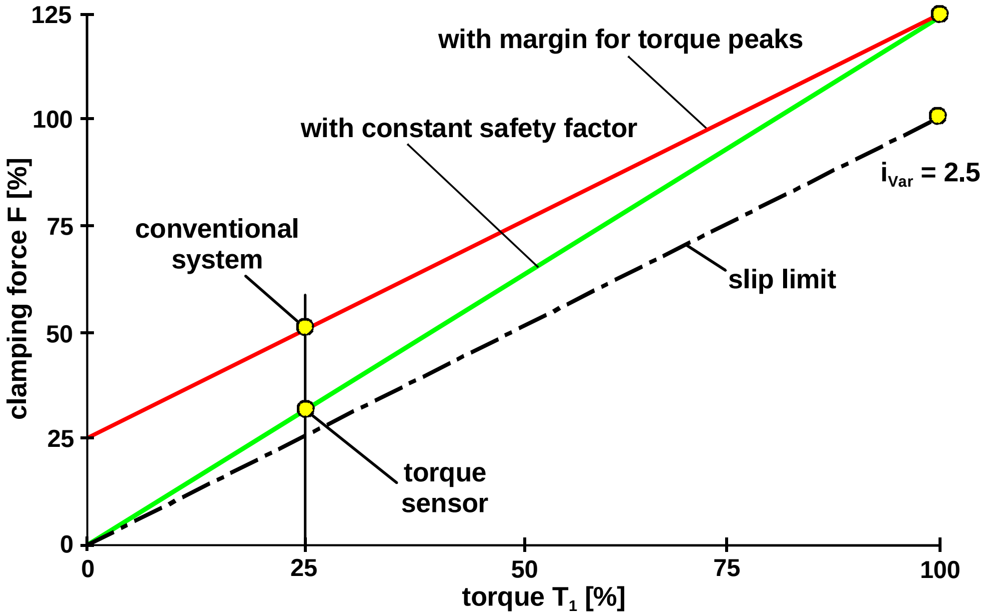


Figure 3b: Progression of clamping force over torque in conventional systems with clamping force offset and with hydro-mechanical torque sensors with a constant safety factor

The LuK system, with its hydro-mechanical torque sensor, uses a constant safety factor, i.e., the absolute excess clamping force is reduced as torque decreases. Almost instantaneously, it records the torque directly at the variable speed mechanism, and it can also provide short-term pumping action with no assistance from other control devices in the event of drive-side or output-side impacts. This is described in more detail below. By avoiding excess clamping force, the transmission in this type of system also works very efficiently during partial load operation, which is extremely important for fuel consumption.

An analog depiction of the minimum required clamping force for various torques over CVT ratio is provided in Figure 4a. The result is a hyperbola-like curve of required clamping force over transmission ratio.

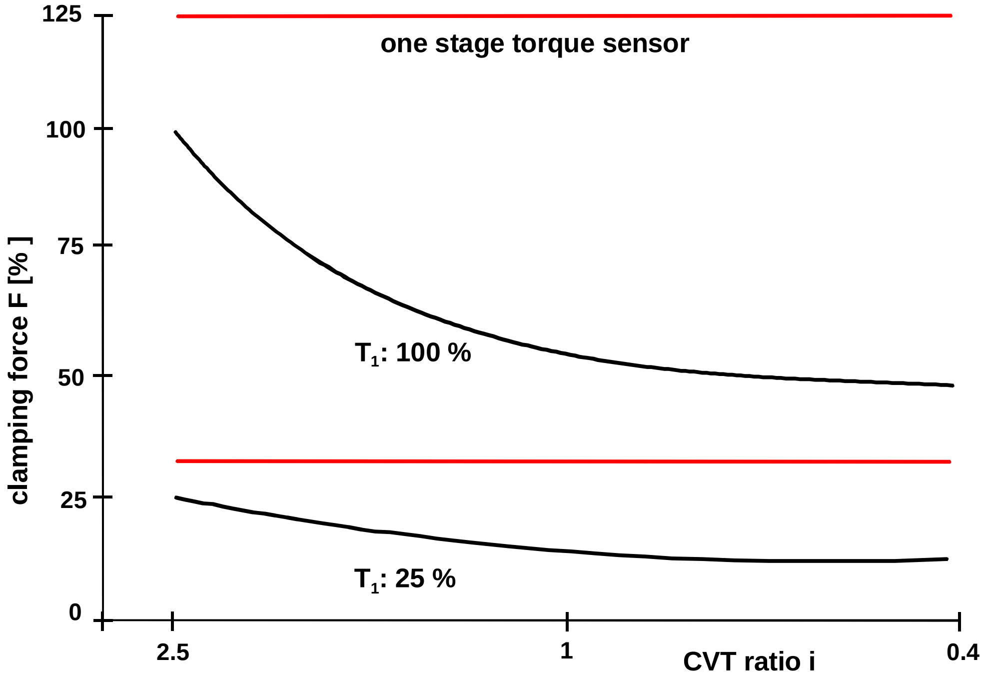


Figure 4a: Possible clamping force for various torques over CVT ratio with a single-stage torque sensor

A constant characteristic clamping curve is undesirable because the greatest portion of driving time is spent in overdrive. In Figure 4b, total driving time is divided into time shares showing the results of longer trips on country roads and highways in a CVT vehicle in the 300 Nm segment.

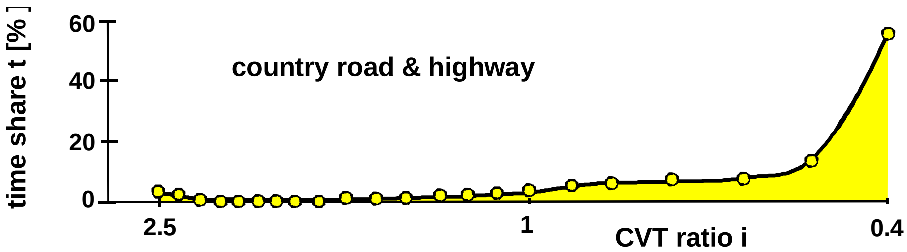


Figure 4 b: Time shares measured on country roads and highways in the CVT vehicle

Two-stage switching of the torque sensor's characteristic curve, as shown in Figure 4c, seems to be well suited to the distribution of driving time shares. Time shares spent in underdrive with occasional periods of unnecessarily high clamping force are very much underrepresented. Simulations performed at LuK have shown that fuel consumption at the theoretical optimum clamping force is only approximately 0.3 % lower than the fuel consumption achieved by this simpler two-stage characteristic curve.

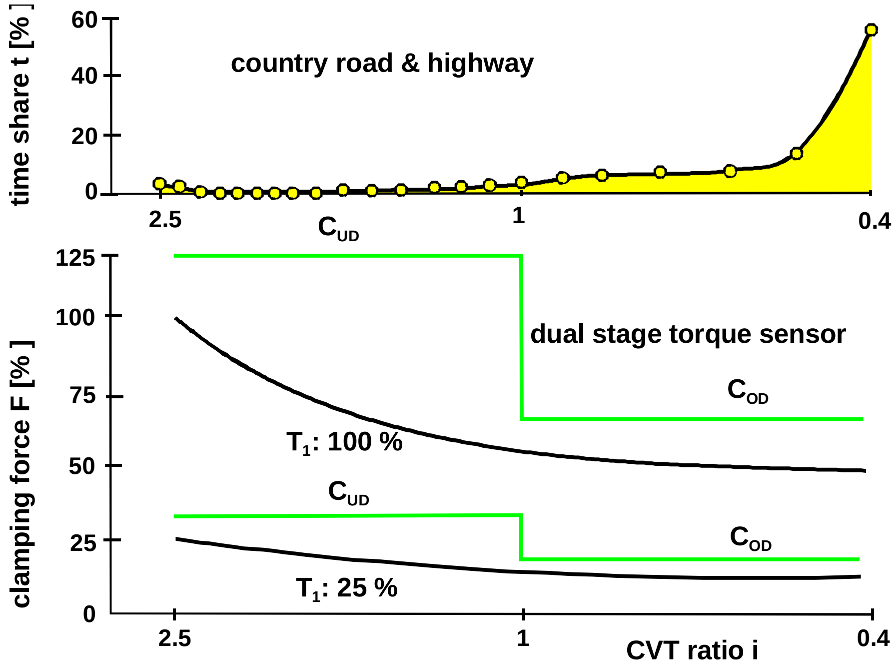


Figure 4c: Measured time shares (above) and clamping force with dual-stage torque sensor (below)

Dual-Stage Hydro-Mechanical Torque Sensor

The previous section discussed the considerations leading to the development of the dual-stage torque sensor. The basic principle behind the familiar single-stage hydro-mechanical torque sensor is depicted in Figure 5. Torque is induced using a ramp plate, through which the power flows over balls to an axially moveable sensor piston supported by oil pressure. Oil coming from the pump flows out through a discharge bore whose flow resistance changes with the movement of the sensor piston until equilibrium is established between the axial force of the ball ramp and the compression force. In this manner, the torque sensor adjusts the pressure routed directly into the clamping cylinder in exact proportion to the adjacent torque.

The moveable sensor plate closes the discharge bore if there is a sudden change in torque. If torque continues to rise, the sensor plate then actively forces the oil out of the torque sensor chamber into the pulleys to increase the clamping force. In other words, the torque sensor acts like a pump for a short time. This “back-up pump”, which works only when needed and does not require any drive power, can provide a short-term flow of more than 30 l/min in the event of a sudden change in torque.

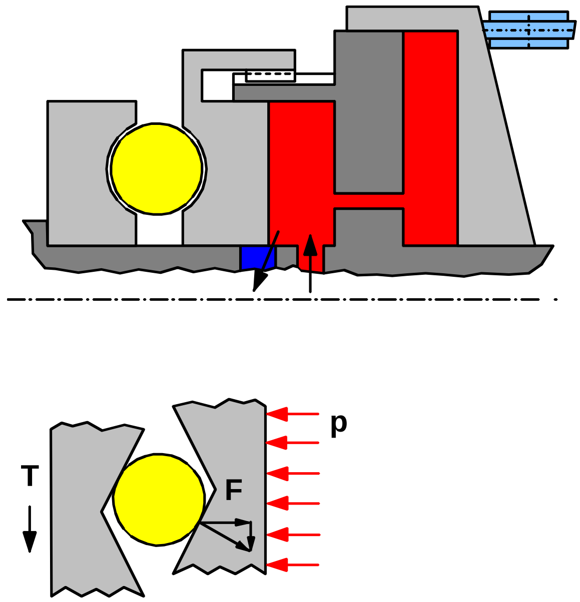


Figure 5: Principle behind the single-stage hydro-mechanical torque sensor

To produce a two-stage characteristic curve, the pressure area of the sensor piston is divided into two parts (Figure 6). In underdrive, where clamping force must be higher to transmit the torque due to the small effective radius of the chain, pressure acts on only one part of the surface. To equalize the ramp force, supplied by torque, the pressure in the torque sensor must be high and consequently also in the clamping cylinder.

In overdrive, beyond the switch point, pressure acts on both parts of the surface. This is why the clamping force is lower at a given torque.

Changing the gear ratio creates axial displacement of the moveable pulley flange of the primary pulley. This then switches the characteristic curve directly by enabling or disabling the second partial surface. In underdrive, the second partial surface is ventilated by the right switch bore at atmospheric pressure as shown in Figure 6. However, in overdrive, this bore is closed by the moveable pulley flange, and the left switch bore provides a connection to the hydraulic fluid.

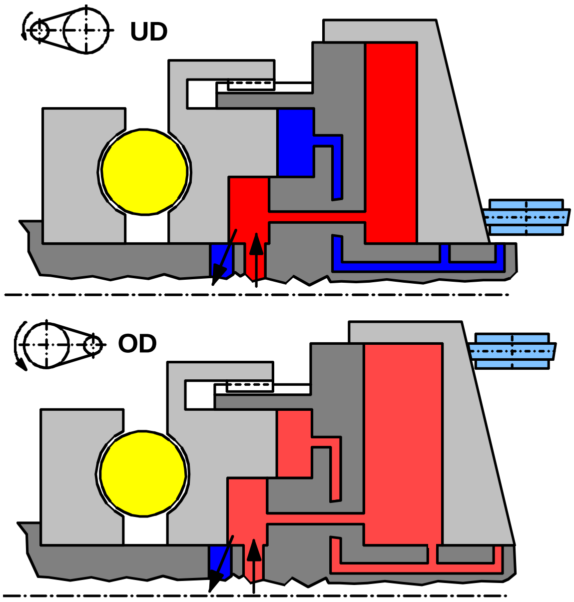


Figure 6: Principle behind the dual-stage torque sensor with high clamping force in underdrive (above) and lower clamping force in overdrive (below)

Requirements of the Adjusting System

Now the oil flow required for rapid adjustment of the variable speed mechanism will be considered. Figure 7 illustrates the necessary flows for a rapid adjustment to underdrive. Rapid adjustment is required during sharp braking maneuvers where underdrive transmission is desired immediately after the vehicle comes to a stop, after kickdown actuations, or on driver demanded gearshifts.

The flows listed are required for the given system reaction time as a function of initial driving speed, and thus, the braking time. If a conventional variable speed mechanism system is installed, the flow must exceed 13 l/min at the critical speed (for this case) of approximately 25 km/h. Contrastingly, the required flow for an adjustment cylinder operating on the LuK double piston principle is reduced to approximately one-third of the conventional system's requirements. This is possible because the pump must activate only part of the cylinder, the so-called adjustment cylinder, when it adjusts the variable speed mechanism.

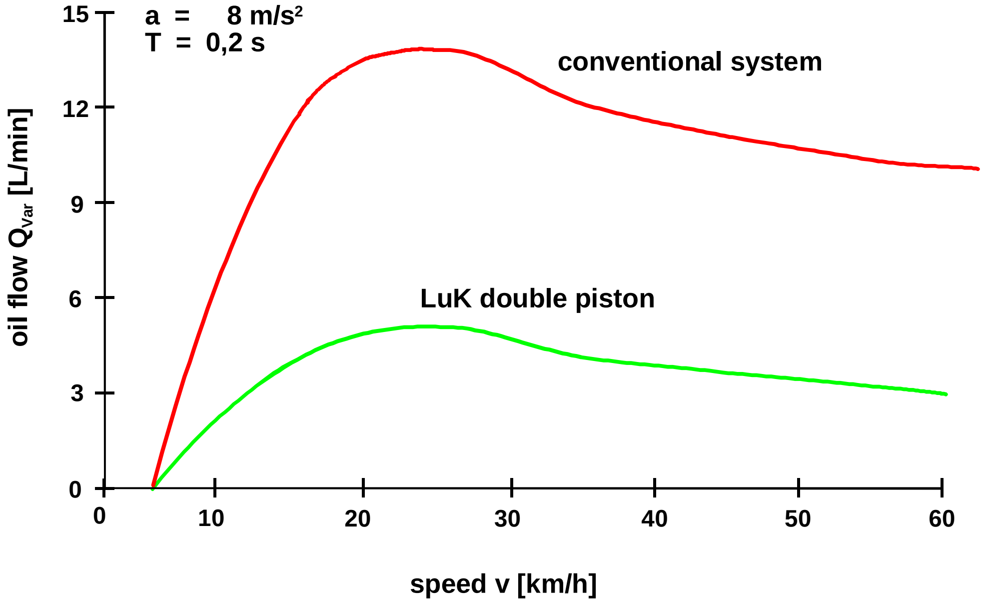


Figure 7: Required oil flow into the pulley cylinder for rapid underdrive adjustment in conventional systems and in the LuK double piston system

LuK Double Piston System with Dual-Stage Torque Sensor

Conventional systems have one pressure cylinder on the drive-side pulley and one on the output-side pulley (Figure 8 left). The oil flows from the pump to a control unit that directs the pressure to be induced in the cylinders. These cylinders combine clamping and transmission adjustment functions into one component. The primary cylinder surface is often designed to be significantly larger than the secondary surface.

The primary reason for this is the inability of many CVT hydraulic systems to set primary cylinder pressure higher than secondary cylinder pressure.

For a rapid adjustment into underdrive mode, the pump must satisfy the high flow requirements of the entire secondary cylinder surface. At the same time, hydraulic fluid is released from the primary pulley into the oil sump, which results in an energy loss. This occures similarly for overdrive adjustments. Therefore, a pump with a large transport volume is necessary to fulfill the system's dynamic requirements, with the corresponding negative effect on the pump's energy requirements.

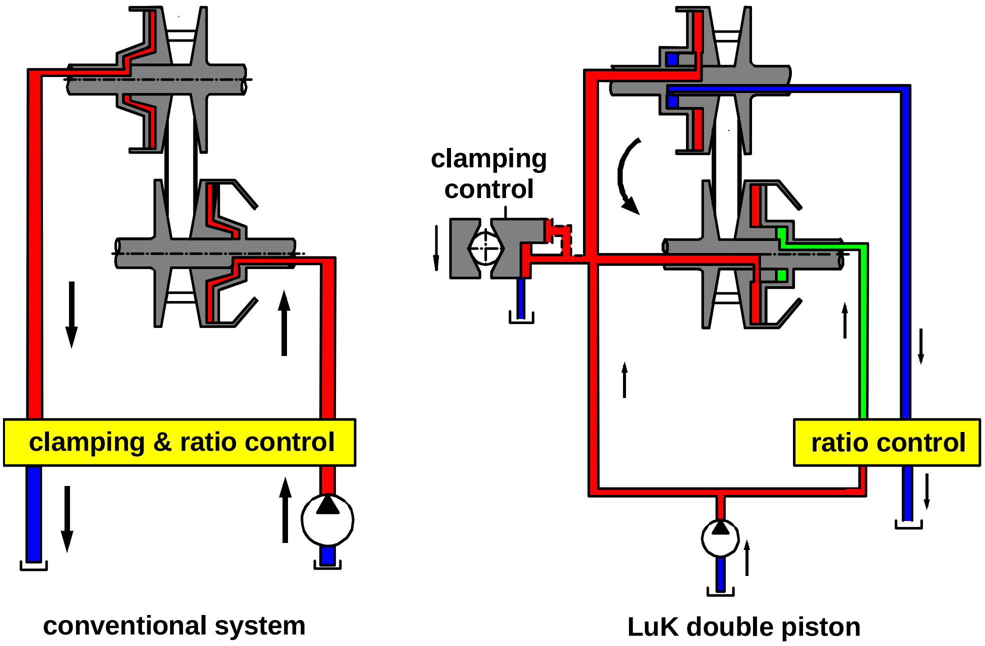


Figure 8: Principle behind CVT hydraulic systems and comparison of volume flows during a rapid adjustment to underdrive performed by a conventional system (left) and by the LuK double piston model (right)

Unlike the conventional system, the LuK double piston model divides the cylinder areas into:

- partial surfaces (red) that ensure clamping, and
- smaller, separate partial surfaces (blue or green) that are responsible for making adjustments.

As we have already discussed, the dual-stage torque sensor ensures clamping. In two stages, the sensor adjusts the pressure in the clamping cylinders in strict proportion to torque as a function of the CVT ratio.

Adjusting the pulleys requires only a small volume of oil to service the comparatively small surfaces of the adjustment cylinders. When the variable speed mechanism is adjusted, the clamping oil, which is under high pressure, is transported directly from one pulley to the other without requiring any additional expenditure of energy. This means the pump for the LuK double piston principle is significantly smaller than pumps for conventional CVT systems, which improves overall transmission efficiency and subsequent fuel consumption.

In Figure 9, the basic idea of this concept, relating to the expenditure of energy during ratio adjustment, can be clearly seen. As far as the conventional system is concerned, pressurized oil is released from one pulley into the oil sump. In order to facilitate filling of the other pulley, oil from the sump, at atmospheric pressure, has to be raised to the level of pressure demanded for operation, requiring an appropriately high expenditure of energy. This is different to the LuK Double Piston principle: here, the oil is transported directly from one pulley to the other, maintaining the high pressure level, without additional energy expenditure.

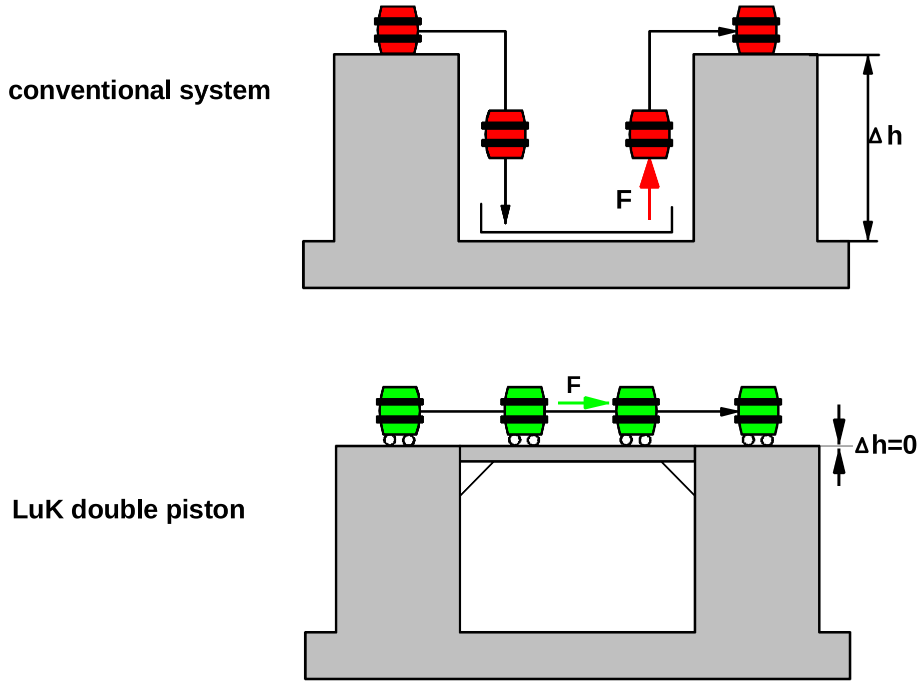


Figure 9: Comparison of the energy expenditure for adjusting the pulleys for the conventional system (above) and the LuK double piston concept (below), which is more favourable from an energy point of view

The upper portion of Figure 10 depicts the application of the double piston and dual-stage torque sensor concepts with the primary pulley for underdrive; overdrive is depicted in the lower portion. The hydraulic fluid (red) flows from the pump into the torque sensor chamber and off to the left over the discharge edge with lower pressure. The clamping force set by the torque sensor to create equilibrium of forces directly affects the clamping cylinder of the primary and (not pictured) secondary pulley simultaneously. The second chamber of the torque sensor (blue) is radially further outward. It is not under pressure because it is ventilated by the right switch bore. The path of the hydraulic fluid to the left switch bore is blocked by the altered position of the sliding pulley element.

The path of the adjustment oil to the radially outer primary adjustment cylinder is indicated in green. The total axial force is equal to the compressive force of the clamping cylinder plus the additional compressive force of the adjustment cylinder. Pressure acts upon the adjustment cylinder so that it can not only make adjustments, but also secure the ratio by axially loading the primary and secondary pulleys in relationship to the force requirements.

In this manner, the entire available surface area for the pulley's size is used to generate axial force.

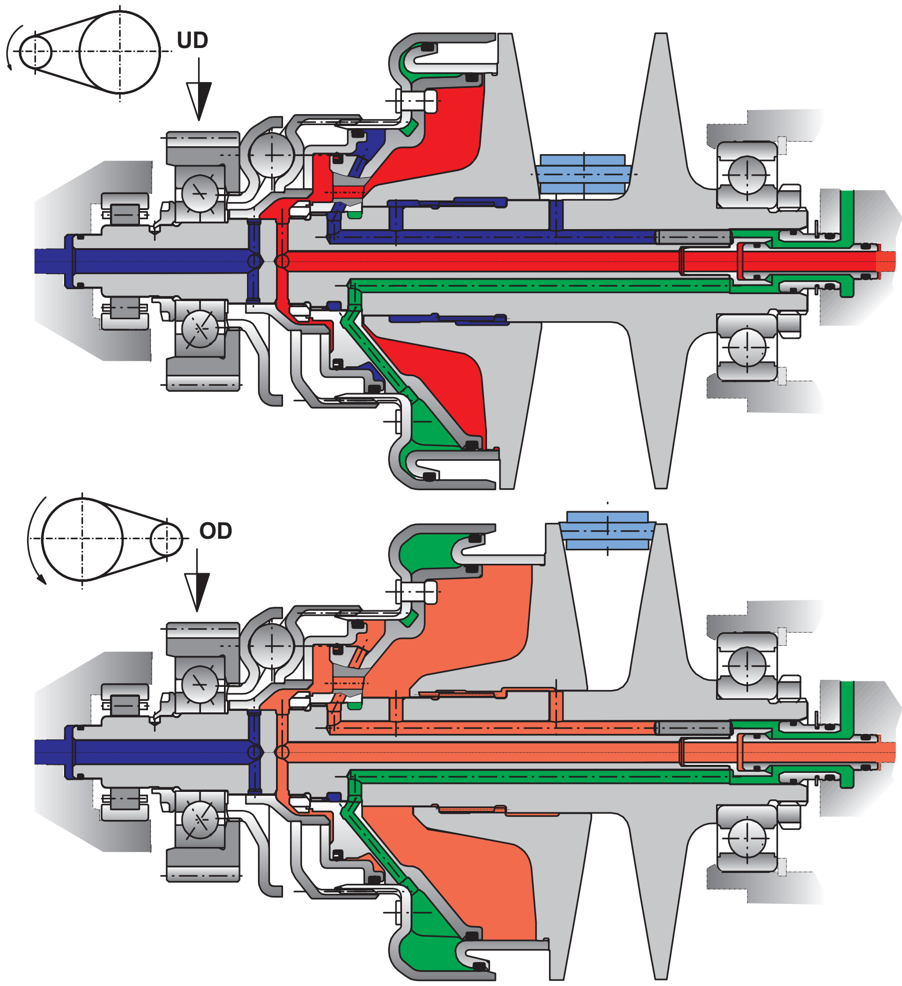


Figure 10: Design of primary pulley with double piston and dual-stage hydro-mechanical torque sensor

In the overdrive position, with correspondingly low clamping force requirements, the sliding pulley element moves towards the fixed pulley element (bottom). This closes the right ventilation switch bore and opens the left bore leading to the clamping cylinder, allowing the hydraulic fluid to flow through this bore system into the second torque sensor chamber. Because pressure is being applied to a larger surface area, the torque sensor decreases the clamping force accordingly, despite the fact that the adjacent torque remains constant. The adjustment cylinder continues to operate as above. The torque sensor's ramp plates are manufactured from sheet steel. The teeth for the plates are also formed during the manufacturing process.

Figure 11 illustrates the development of torques, clamping force and wheel revolutions for a vehicle equipped with the system described while it moves onto a partially frozen section of road and during its subsequent transition to normal road.

The lower portion of the figure illustrates that the right drive wheel rotates at a speed corresponding to the gradually increasing vehicle speed, while the left drive wheel spins with rapidly increasing slip. After the vehicle has reached the normal road surface, the slipping wheel is slowed in large gradients. At this point, the right wheel accelerates more strongly in tandem with the vehicle. Following the slip phase, both wheels accelerate in sync with the vehicle.

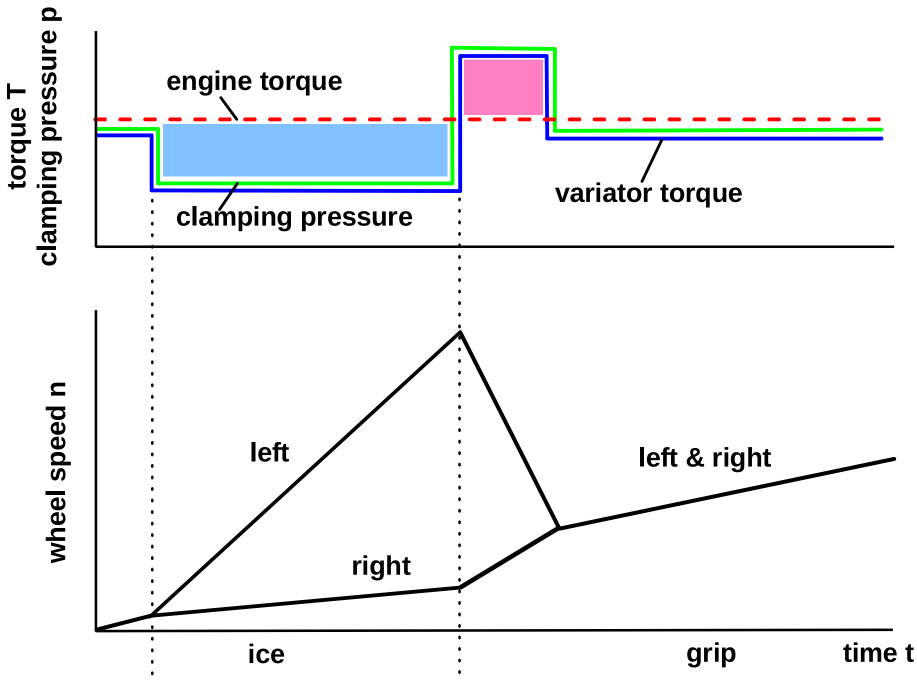


Figure 11: Torque and clamping force curve while driving onto partially frozen road, followed by a transition to normal roads

The top portion of the figure illustrates the development of torque and the clamping force set by the torque sensor. The internal engine torque remains almost constant during the approach, but while the vehicle is on the ice, most of the torque is used to accelerate the rotating parts. The variable speed mechanism torque is correspondingly low during this phase.

During the slip phase on the normal (non-slippery) road, the variable speed mechanism torque increases drastically to a value far above the engine torque. This is caused by the large negative acceleration of the rotating parts along the entire drive train, including the engine. The engine torque and variable speed mechanism torque are not substantially identical until the subsequent normal acceleration phase.

Because the torque sensor records the exact torque induced in the variable speed mechanism, the clamping force set by the torque sensor can behave in a static or dynamic manner analogous to the variable speed mechanism torque. The short-term pumping capacity of the torque sensor and its ability to make rapid adjustments ensures highly dynamic clamping force adjustment in accordance with the adjacent torque.

A further advantage of the torque sensor is its ability to make extremely precise adjacent torque-clamping force conversions using only geometrical data. This allows the clamping force to be set precisely at the minimum point even under partial load operation, where the torque values given by the engine control system possess a greater degree of uncertainty. This also has a beneficial effect on fuel consumption.

The CVT Chain as Contact Element

Using a rocker pin chain made by P.I.V. Antrieb Werner Reimers as a starting point, LuK made further improvements to the CVT chain for automotive applications. The development process focused on improving its strength to achieve the high required power density and on improving its acoustic properties.

Figure 12 illustrates the CVT chain for applications producing torque up to 300 Nm. It is constructed of various links, which form the strands, the rocker pins, and retaining elements.

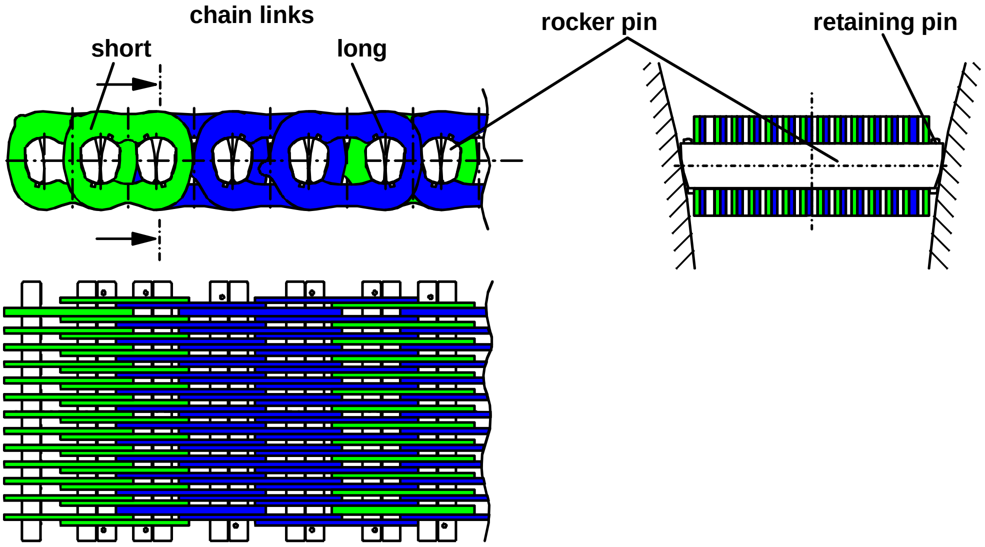


Figure 12: Design and components of the LuK-P.I.V. CVT chain

The CVT chain has the following characteristics:

- It has **low fuel consumption** and **excellent power transmission performance**. This is made possible by the rocker pin design of the CVT chain, which allows short rotations around the pulley flanges and a **high spread of gear ratios**.
- The CVT chain allows transmission of **high torque levels**. Thicker links on the outside edges of the strands equalize load distribution.
- Because the rocker pins are able to "**seesaw**", the chain experiences **low internal friction losses**, ensuring **high transmission efficiency**.
- The CVT chain is **resistant to axle offset** due to the rocker pins' crowned faces. In combination with cambered pulley flanges, these elements **reduce additional axle offset** created whenever ratio is changed. Furthermore, the CVT chain is **resistant to pulley deformation** under load, **angular errors** and **relative rotations** between the fixed and moveable pulley flanges. **Ball-guidance** for the axially adjustable pulley flanges are therefore **unnecessary**.
- The CVT chain produces **low axial forces acting on the primary pulley.** This allows work to be done at low hydraulic pressure with the given cylinder surfaces, an additional plus for **transmission efficiency**.

- The strand is designed with a basic three link asembly module, allowing the base pitch to be small. In a short chain link, a link clip is all that separates neighboring rocker pins from each other. A second, different link length allows **favorable acoustic behavior** to be achieved by calculating the optimal pitch sequence of long and short links.

Durability Calculation and Design of the CVT Chain

Determining the operational life of a CVT chain is divided into four subprocesses, each of which is depicted schematically in Figure 13.

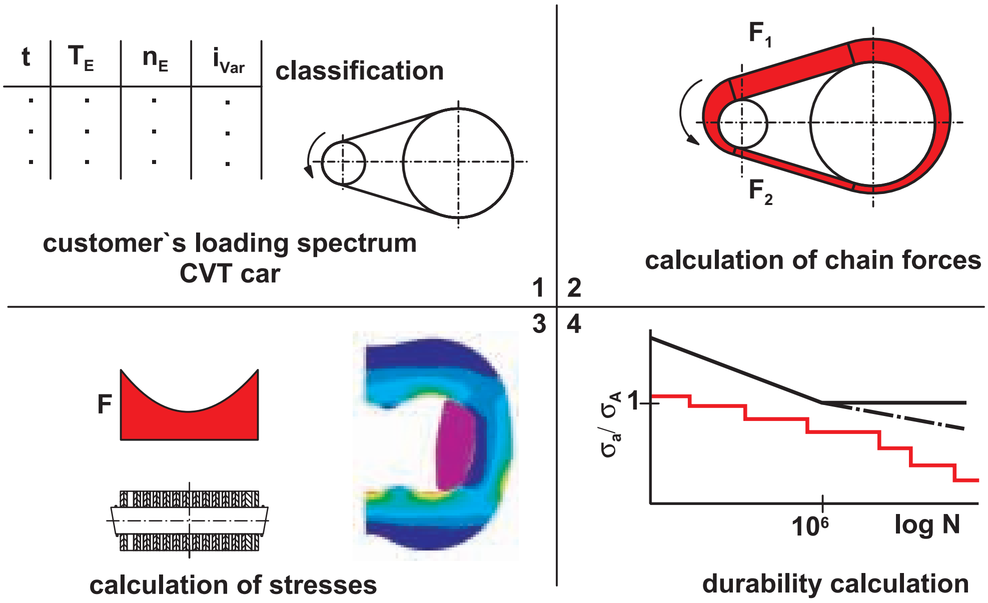


Figure 13: Load analysis and calculation of operational life of the CVT chain using car manufacturer's loading spectrum as a basis

(1) Classification of the customer's loading spectrum

Ideally, the customer provides a loading spectrum that is representative of the required performance specifications. In addition to the engine characteristics and the test track profile, gross weight, drive train design, and the vehicle's chassis and tires also play a role in the load placed on the CVT.

The available curves for engine torque, engine speed, and CVT ratio are prepared for damage calculations by gathering classes of these parameters. In this manner, we can gather several hundred different sets of driving conditions with their respective percentage of the collective running time.

(2) Calculation of chain forces

Tight side and slack side forces acting on the CVT chain for each operational point in (1) are calculated. In addition to the peripheral forces resulting from torque transmission, axial forces for both pulleys are also included in the calculation, which is based on Dittrich's theory [5, 6].

(3) Calculation of component stress

Using the chain tensile force figures as a starting point, appropriate calculations are made to determine the stress distribution for each point of the spectrum. This applies both to the distribution of forces within the strands and to the distribution of tension in the links and rocker pins.

(4) Durability calculation

Using damage accumulation hypotheses, a calculation of operational life is performed based on experimentally gathered Wöhler material or component curves. The resulting damage total provides information on the feasibility of the prospective chain design.

CVT Chain Load Distribution

Figure 14 illustrates the load distribution for the chain strand during underdrive ratio at full engine torque.

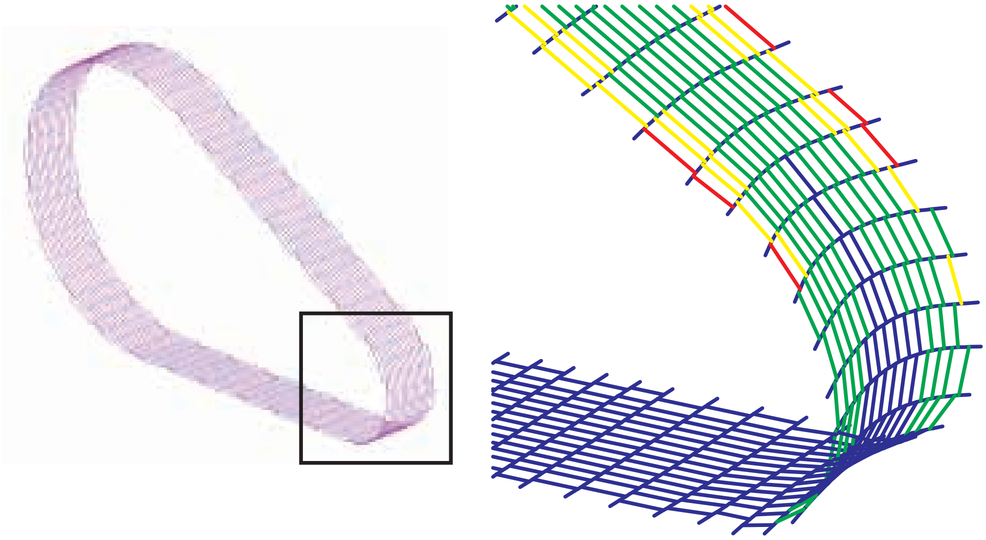


Figure 14: Unequal distribution of link forces on the CVT chain during underdrive ratio at maximum engine torque, including the effects of rocker pin deformation

Clamping forces, tensile forces, and induced frictional forces deform the rocker pins, shown in an enlarged view on the right side of Figure 14. Deformation places stress on the rocker pins and leads to an unequal distribution of link forces, shown here by the differently colored areas.

Introducing links of varying thickness was a definite improvement over known designs. The tensioning of the components was optimized using finite element method (FEM). Without altering the CVT chain's main dimensions, its life was more than doubled by all the improvements made using the given spectrum.

Optimization of Acoustics

Special attention was given to optimizing the acoustics of the CVT chain.

By using links of different lengths and carefully sequencing them along the CVT chain, we suppressed the disturbing monotone note to a large degree. The mixture ratio and the pitch sequence were mathematically optimized for the corresponding application.

Figure 15 uses plastic-head measurements in the vehicle's interior to illustrate the success of such computer-assisted optimization processes. Acceleration trials at speeds between approximately 30 and 80 km/h were performed with a special measurement parameter set for transmission control that maintained a constant engine speed. This procedure allowed the various sources of noise (listed below) to be clearly separated from each other:

- Horizontal Lines: Drive (engine, assembly, transmission input),
- Diagonal Lines: Output (wheels, axles, transmission output),
- Curved Lines: Chain actuation of the variable speed mechanism.

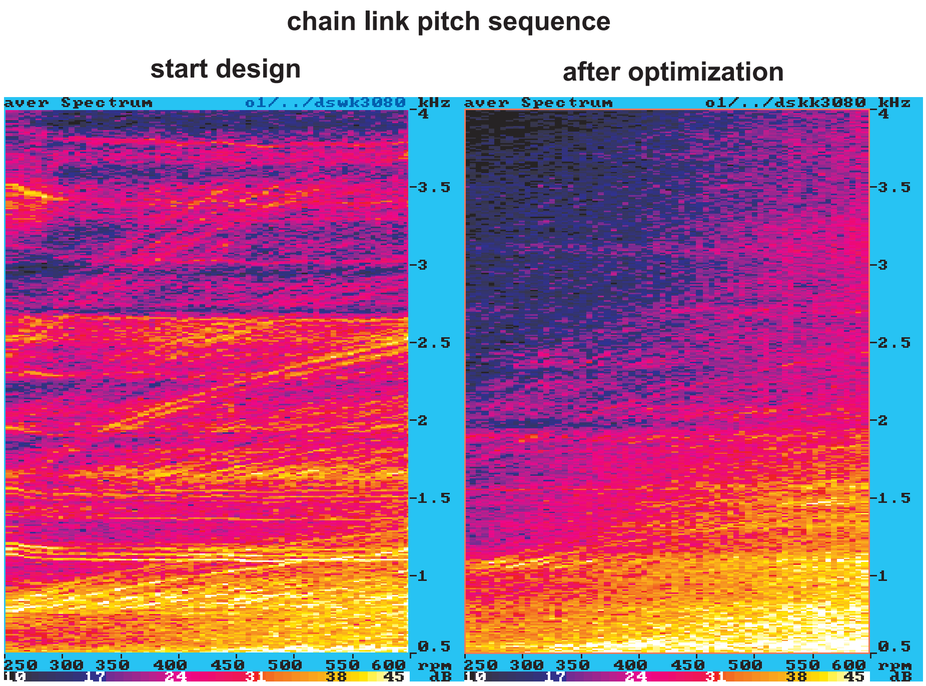


Figure 15: CVT chain acoustics: optimization via calculated simulations and advance planning of the pitch sequence of short and long links

Peaks in the form of curved lines are visible on the left side of Figure 15. These are the points at which the CVT chain engages the pulleys, points considered disturbing here.

The CVT chain on the right side of Figure 15, after improvements to its mixture ratio and pitch sequence, has almost no disturbing monotone note at all. This allowed us to achieve favorable acoustic behavior.

Other important influences on the acoustics of the complete CVT system are the pulley arrangement, housing design, and all other paths of structural noise transfer. These factors are additional candidates for acoustic optimization.

Summary

LuK has developed CVT components for torque up to 300 Nm and beyond. Their most important special features are:

- Precise control of the minimum clamping force required for proper torque transmission and high efficiency;
  this is accomplished by means of a dual-stage hydro-mechanical torque sensor that measures adjacent torque at the variable speed mechanism with great accuracy and ensures highly dynamic clamping. It is also capable of providing additional short-term pumping action when necessary.
- Excellent adjustment dynamics at low pump capacity, accomplished by direct transport of the clamping oil, which is under high pressure, from one pulley to the other when adjustments are made;
  made possible by the LuK double piston concept with separate cylinders for clamping and adjustment.
- Greater spread of gear ratios, sturdy design, high efficiency and optimized acoustics in the power transmission element;
  made possible by the CVT chain, developed with the aid of computer simulations, for torques up to 300 Nm and beyond.

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