PREFACE

After a period of four years, the 5th LuK Symposium marks a turning point in our endeavours to introduce both customers and specialists to the knowledge and experience of our development.

Besides the clutch sector, for the first time we have turned our attention to the automatic transmission sector, thus following the new areas of activity for LuK.

Together with the motor vehicle manufacturers, we wish to give fresh impetus to our branch of industry so that we can get it moving through new ideas during the present "stagnation" period.

Ernst H. Kohlhage

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Advanced Development of Dual Mass Flywheel (DMFW) Design - Noise Control for Today's Automobiles

Dr.-Ing. Albert Albers

Introduction

The clutch system in a vehicle performs two main functions:

- Power interruption and modulation during start up and when shifting
- Reduction of rotational vibrations in the drive train induced by engine irregularities

During the LuK Clutch Symposium, LuK will introduce some new developments which successfully fulfill these functions for our customers.

The following presentation will illustrate a cross-section of development efforts aimed at reducing engine-induced rotational vibrations in the drive train.

Rotational vibrations affect durability of the drive train components and create

- Gear rattle
- Body boom
- Tip-in/back-out vibrations

These factors produce considerable noise and a loss in driving comfort.

The main cause of these rotational vibrations is variation in torque. This variation results from the discrete piston combustion cycle of the engine as a function of the ignition frequency.

The vehicle drive train is a vibrating system. Figure 1 shows a simple model designed to simulate fundamental vibration behavior. The engine, transmission and vehicle are represented as rotating inertias connected by springs. The spring C_3 represents the stiffness of the drive train, while spring C_2, located between engine and transmission, represents the spring characteristic of the torsion damper.

Such a system has two vibrations modes. The first mode, with a natural frequency of between 2 and 10 Hz, is known as the tip-in/back-out reaction. This is generally excited by a driver-induced load change.

The second mode, where the transmission inertia vibrates against engine and vehicle, has a natural frequency of 40 - 80 Hz with conventional torsion dampers. This is a typical cause of gear rattle.

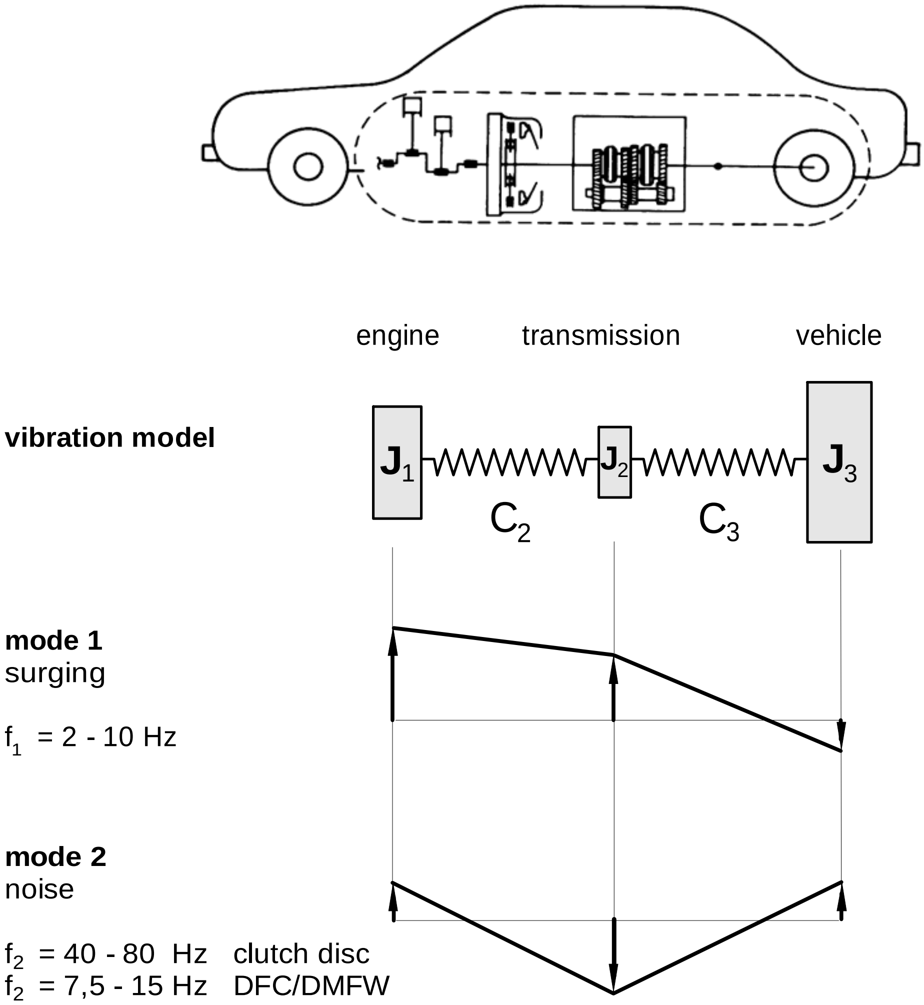
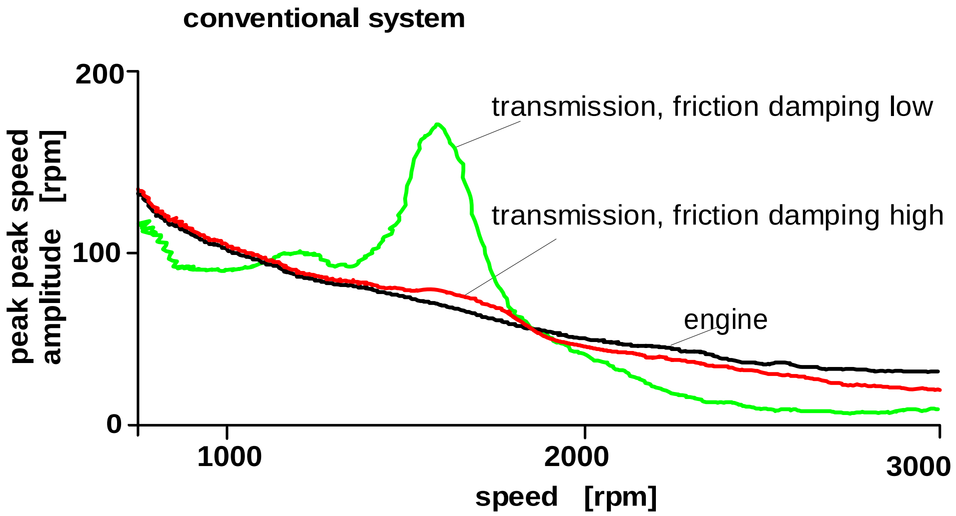


Figure 1: Vehicle drive train with vibration modes

Consequently, the tuning of a conventional automotive torsion damper - a clutch disc with its corresponding spring characteristic - always involves compromise. The upper graph of Figure 2 shows typical speed fluctuations in a vehicle with a clutch disc. In this case, the friction-damped resonance is located at around 1700 rpm. Further damping of this resonance leads to a worsening of the hypercritical isolation of rotational vibrations (at speeds higher than the resonance).



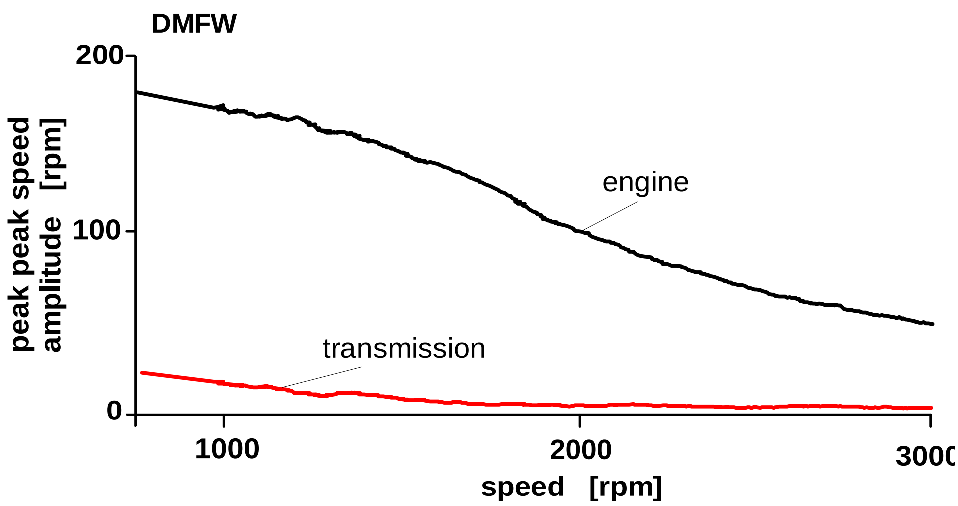


Figure 2: Torsional vibration isolation with conventional clutch disc and dual mass flywheel (DMFW)

The goal of torsion damper development is to keep the torsional vibrations induced by the engine as far as possible from the rest of the drive train.

A conventional system only satisfies this requirement at high engine speeds, because the attainable torsion damper spring rates lead to natural frequencies which are always within the normal driving range.

This unsatisfactory situation led to the development of a new torsion damper concept - the dual mass flywheel (DMFW). This design shifts part of the flywheel inertia to the transmission input shaft and drastically lowers the torsion damper spring rate by introducing new spring designs (Figure 3), thus reducing the resonance speed to very low engine speeds. Figure 2, lower graph, shows the hypercritical isolation of rotational engine vibrations (starting from idle speed).

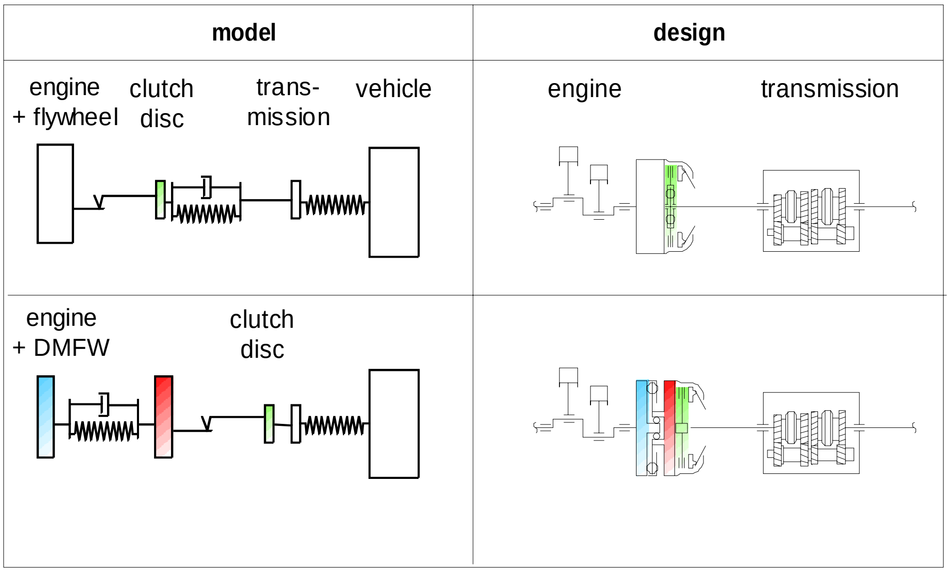


Figure 3: Principle of the dual mass flywheel

Improvements in driving comfort achieved by the dual mass flywheel, together with low-cost designs resulting from goal-oriented, value-analized development, has led to the increased popularity of this system. Currently the LuK dual mass flywheel is used by ten car manufacturers in approximately 80 different models, thus covering a wide range of engines, as shown in Table 1.

number of cylinders	engine type	
	gas	diesel
4	8 (5)	3 (5)
5	3 (-)	4 (7)
6	14 (4)	4 (5)
8	2 (5)	
12	1 (1)	

( ) = in development

Table 1: Dual mass flywheel used in production and development projects

Figure 4 shows a current dual mass flywheel with all its fundamental components. The primary side of the DMFW (shown in blue) consists of formed sheet metal parts which make the spring channel, and a cast hub. The secondary side of the DMFW (shown in red) consists of a cast disc, into which the torque is transmitted from the flange. The secondary side is mounted in the primary side over a ball bearing. The heart of the system is the arc spring, whose special properties will be described in the following section.

The arc spring damper - characteristic and function

The dual mass flywheel consists of the following main function groups:

- **primary and secondary inertias**
- **the torsion damper spring rate**
- **the damping characteristic**

The influence of the moment of inertia has been thoroughly discussed in /1/, /2/ and so will not be discussed in detail here.

The spring rate and the damping characteristic are crucial in determining the operating performance of a DMFW.

What requirements does the ideal torsion damper have to fulfill?

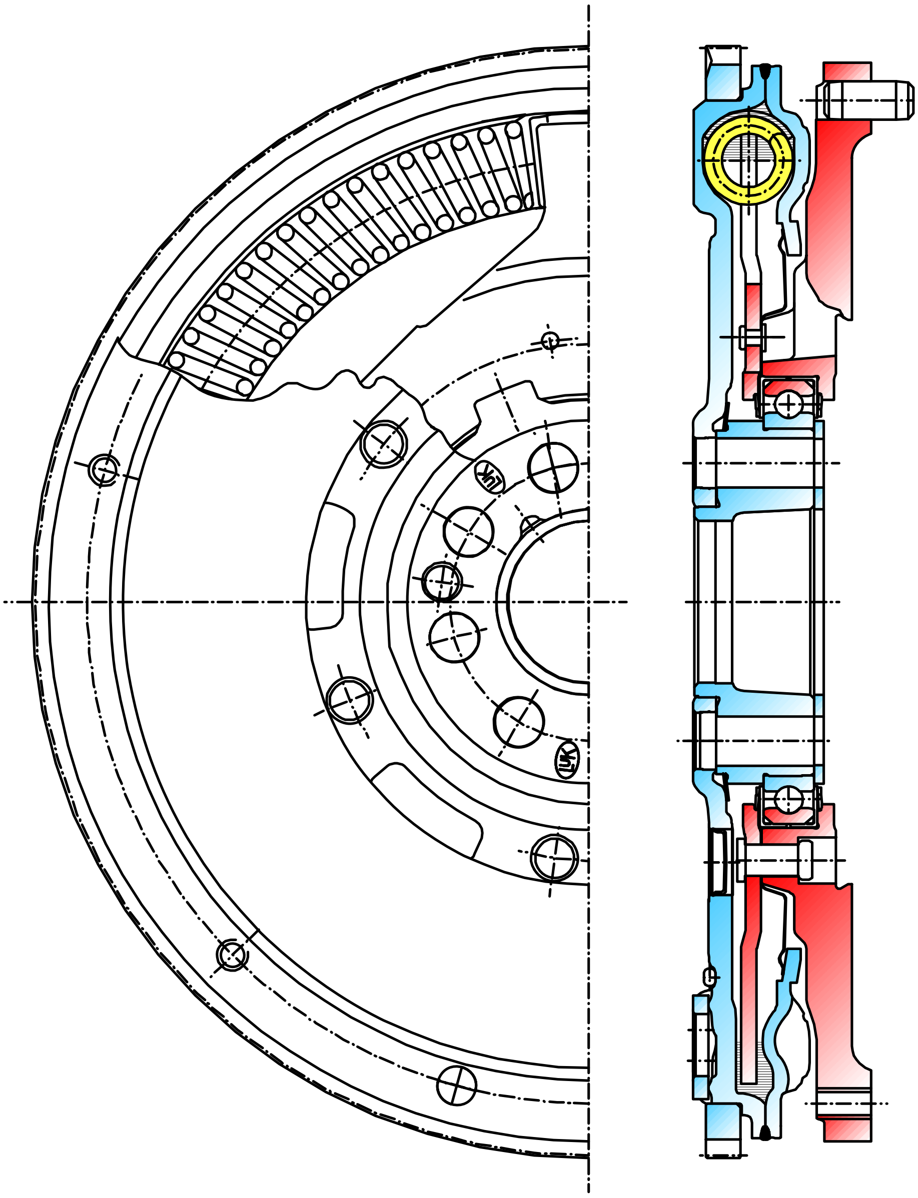


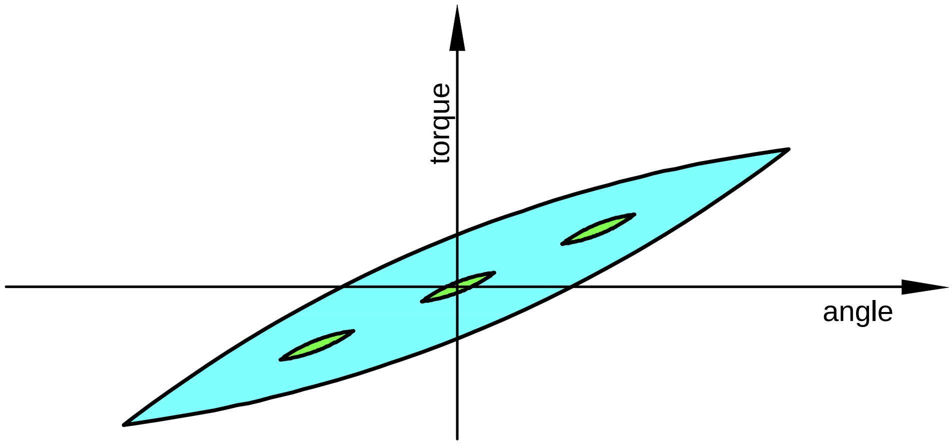
Figure 4: Dual mass flywheel

It has to control three basic operating modes:

- **transmission rattle during idle, drive and coast**
- **resonance break-through during engine start and stop**
- **surging associated with torque changes**

Significant characteristics for these operating modes are the frequency and the vibration amplitudes.

Figure 5 shows how they are interrelated.



operating mode	problem	frequency	angle	TD-requirement spring rate	TD-requirement damping
idle, drive, coast	noise	high	low	low	low
load cycle	surging	low	high	low	high
resonance break-through	noise durability	low	high	low	high

Figure 5: Torsion damper requirements

Transmission rattle occurs during higher excitation frequencies (20 - 400 Hz). Vibration angles due to irregular engine torque are very small in this case.

Even for a diesel engine with its characteristically extreme torsional irregularity, the vibration angle is seldom larger than ±2 degrees. In order to achieve the best possible hypercritical isolation for this operating mode, the torsion damper should have a low spring rate together with a low damping characteristic.

The second operating point is the resonance break-through. During engine start and stop, speed always increases from zero or is reduced to zero. This means that the system always passes through the resonance range of the drive train. When a drive train with a dual mass flywheel is designed, the aim is to achieve hypercritical isolation in the normal operating range, i.e. engine speeds above 700 rpm. This means that the development goal is to achieve maximum reduction of the resonance speed.

The resonance break-through is characterized by low frequency vibrations together with a large vibration angle, because the vibration angle of the engine increases in association with decreasing speed. In this case, the torsion damper design requires a low spring rate with a high damping characteristic in order to avoid resonance magnification while passing through the resonance range.

Load cycling is characterized by low frequency vibrations at large vibration angles. In this case the damper requirements call for the lowest possible spring rate and a high damping characteristic. Sudden excitations of the drive train result in large wind-up angles coupled with high friction damping in the torsion damper. This method dissipates the energy of the free natural vibrations in order to reduce the vibration amplitudes.

Figure 5 represents an idealized damper characteristic designed to meet these requirements, i.e. produce a low spring rate and a high damping for large vibration angles. It also shows that the damping is very low for small vibration angles.

LuK dual mass flywheels contain an arc spring as the main element in order to achieve suitable spring rates and damping characteristics. Arc spring principles are illustrated in Figure 6.

In order to best use the available space, a coil spring with a large number of coils is inserted into a semicircular channel. In the DMFW, the coils of this arc spring are supported by support races mounted in the spring channel in the DMFW. When a load is applied to the spring, the movement of the coils along the support races produce friction, creating the damping. The contact surfaces of the arc spring are lubricated with grease.

The enlarged area in Figure 6 shows the load equilibrium on one coil i of the arc spring.

As the spring load is transmitted along its curved line of action, a normal reaction F_{i*} is created at the contact surface of each coil. In addition to this, there is a speed-dependent centrifugal force F_Z. The sum of these two loads produces the normal reaction force, which in turn produces the friction load F_{Ri} for each coil.

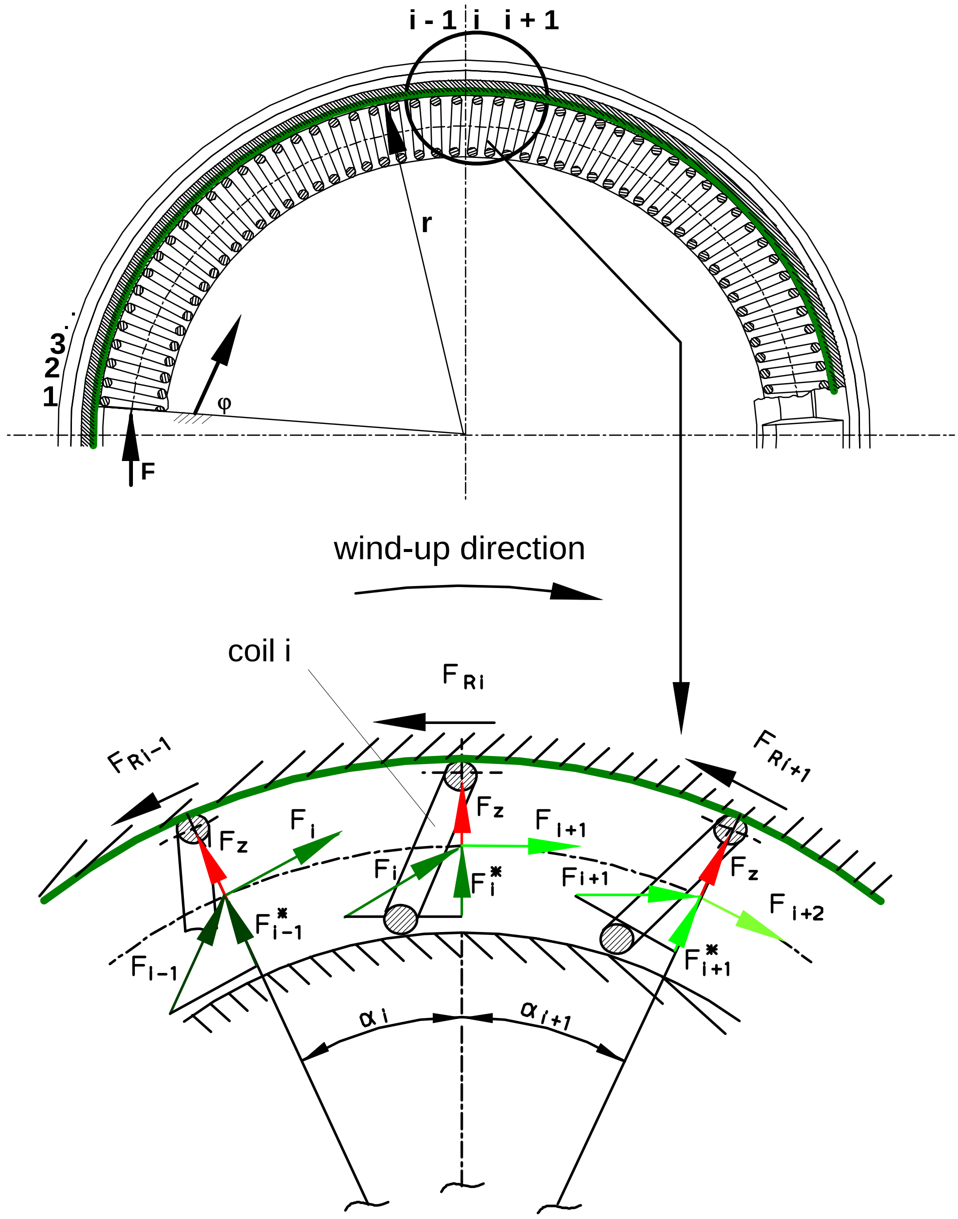


Figure 6: Arc spring

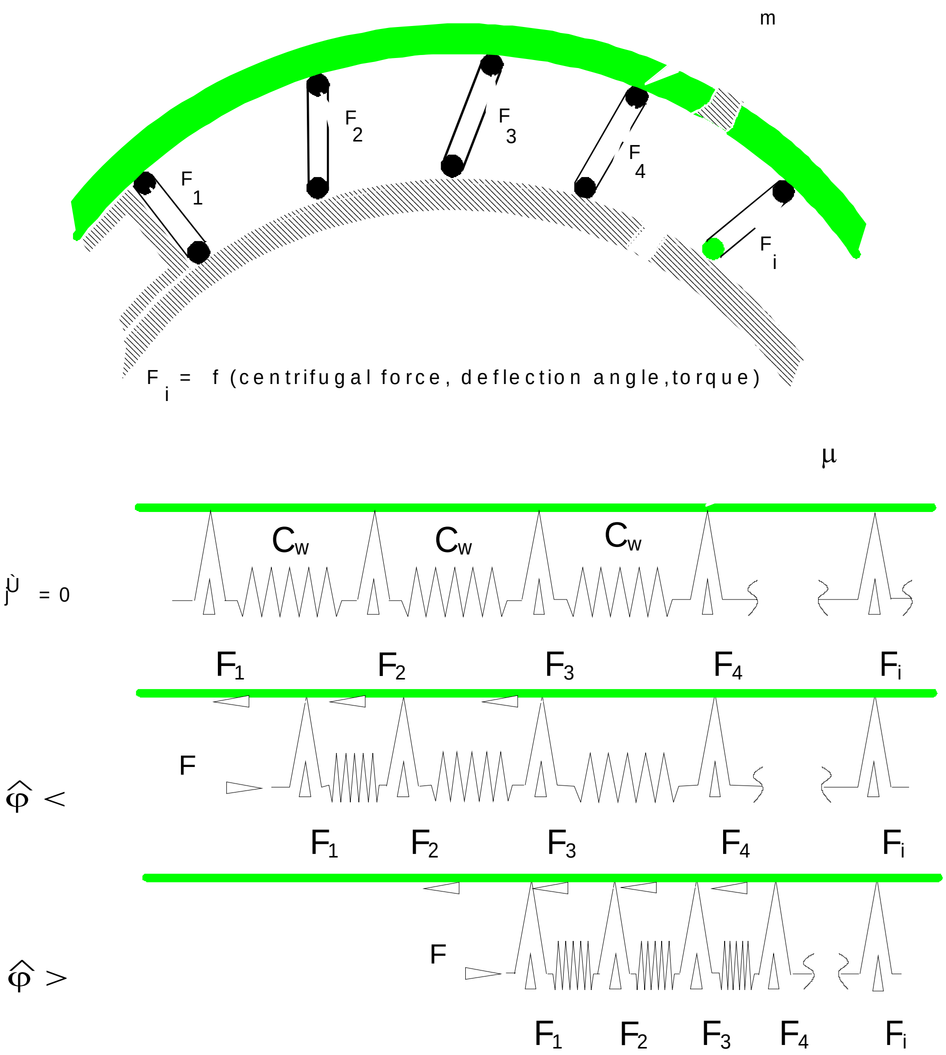


Figure 7: Arc spring damper function

Figure 7 illustrates how this system functions. It shows the individual contact points of the arc spring coils that are linked by the spring stiffness of each coil. There is a normal force F_i at each support point and the friction coefficient μ acts upon each of these support points.

Assuming the system is preloaded to a predetermined operating point and a low cyclical load is applied around this point, an equilibrium will be reached at the contact points between the external load F, the individual coil spring load F_{Fi} and the friction load F_{Ri}.

This operating condition is typical for normal driving (i.e. small vibration angles) and produces spring rates in excess of the nominal spring rate of the complete arc spring. At the same time, however, the resulting friction damping characteristic remains very low.

Figure 8 shows just such partial hysteresis loops with a low damping characteristic (in green). The spring rate, in this case, is clearly higher than the nominal spring rate of the complete arc spring.

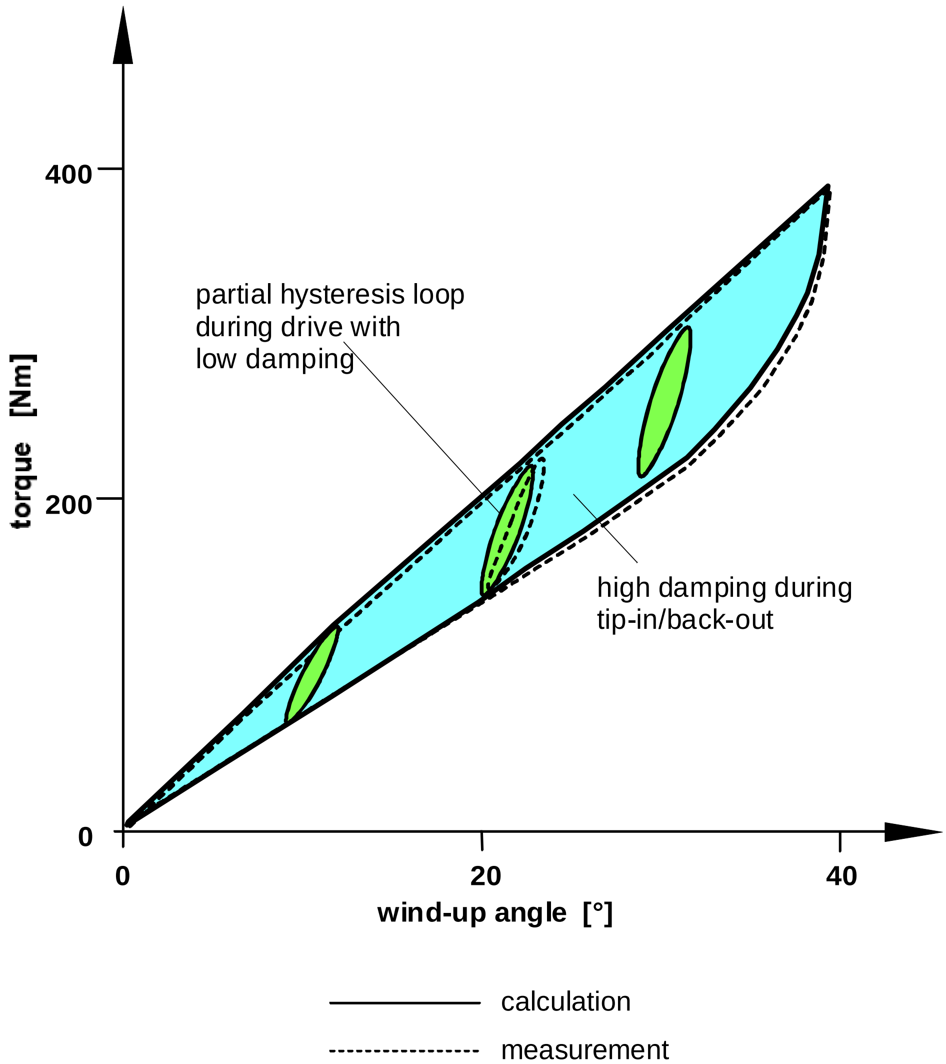


Figure 8: Characteristic curve of an arc spring damper

When large vibration angles occur in the second operating mode as is typical for tip-in/back-out or resonance break-through, all coils of the arc spring become active. This results in a reduced spring rate together with high damping (as shown by the cross-hatched area in Figure 8). Figure 8 also shows the close match between the measured curve and the curve calculated using the method shown above.

The dependence of spring rate and friction damping on engine speed and vibration angle for a special DMFW is shown in Figures 9 and 10.

Figure 9 shows that for an increasing speed and a decreasing vibration angle, the spring rate of the arc spring damper increases (because of the deactivation of the coils). The diagram also shows the engine performance curve with the vibration angle as a function of velocity for drive/coast (in red) and for start/stop (in yellow). The curve represents actual operating points for a specific 2.5 l Diesel engine.

Figure 10 shows the corresponding friction damping pattern. It can be seen that the torsion damper friction increases with increased engine speed. But unlike the spring rate, the friction damping characteristic decreases sharply with reduced vibration angle. Again the reason is that some of the arc spring coils are deactivated.

In-vehicle performance is determined by a combination of spring rate and damping.

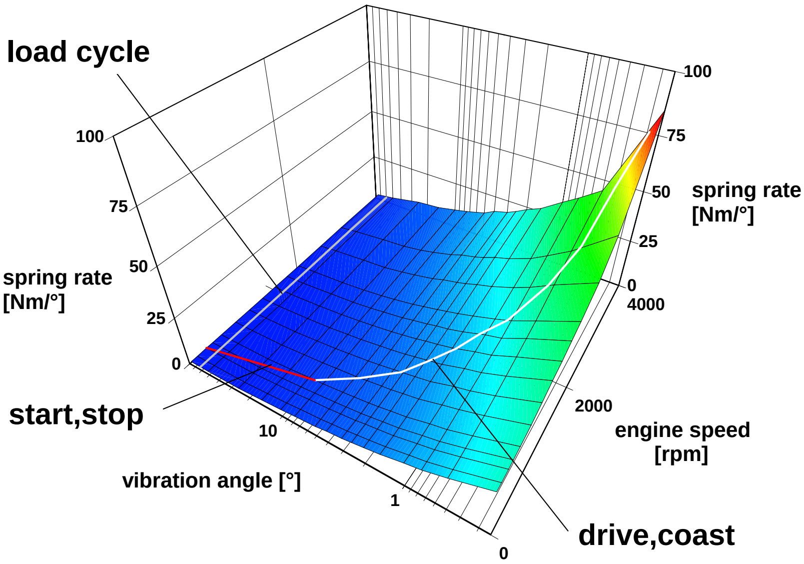


Figure 9: Spring rate of the arc spring damper as a function of engine speed and vibration angle

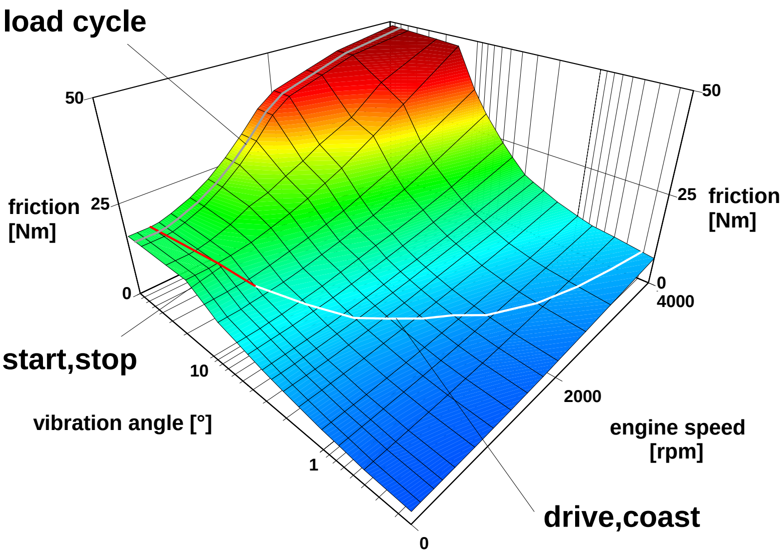


Figure 10: friction damping as a function of engine speed and vibration angle

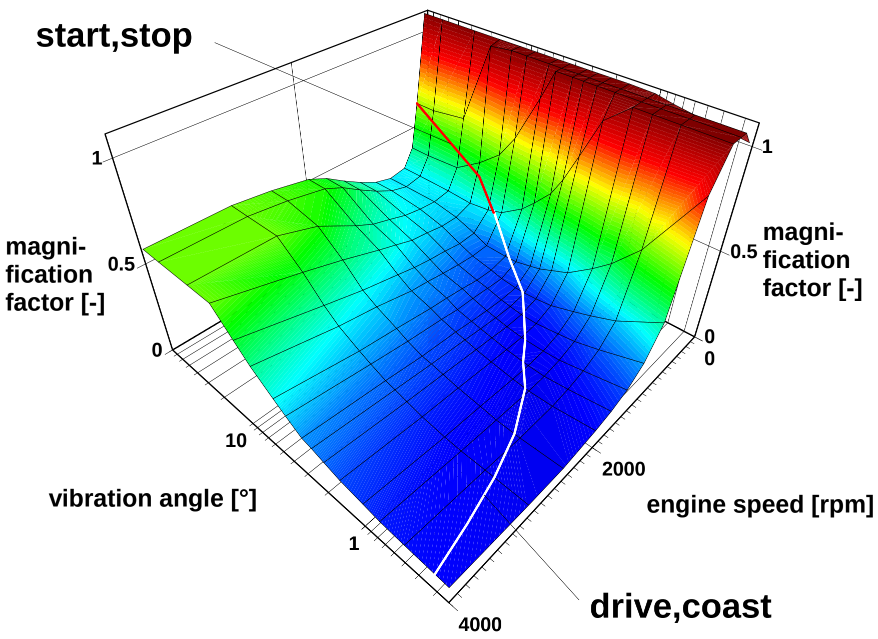
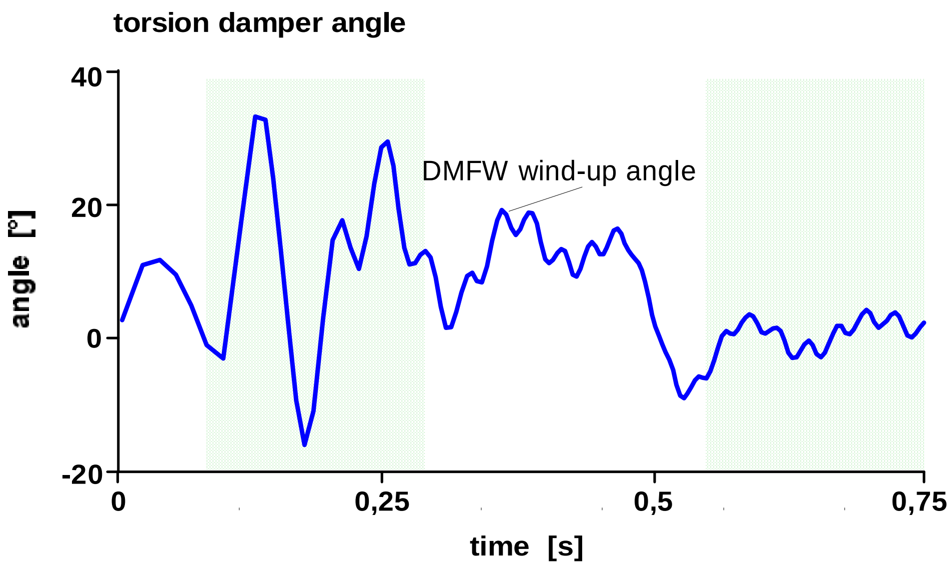


Figure 11: Magnification factor of the arc spring damper

The magnification factor of an arc spring damper characterises damper performance and its effective range. Here, magnification factor is the ratio of the speed irregularity from the DMFW-output (transmission side) to the DMFW-input (engine), depending on operating parameters. This calculation is shown in Figure 11 as a function of engine speed and vibration angle. The line, respectively the surface, with a magnification of 1 corresponds to a complete transmission of the engine irregularities into the transmission input shaft, as if a rigid connection existed between the engine and the transmission.

The graph clearly indicates that an excellent isolation effect has been achieved over wide ranges of engine speed and vibration angle and that a magnification factor of 1 almost never occurs. The engine performance curve displays the actual established values for the magnification factor at any given point. From low to high engine speeds, engine vibration amplitudes are effectively isolated from the rest of the drive train, preventing transmission and vehicle noises.



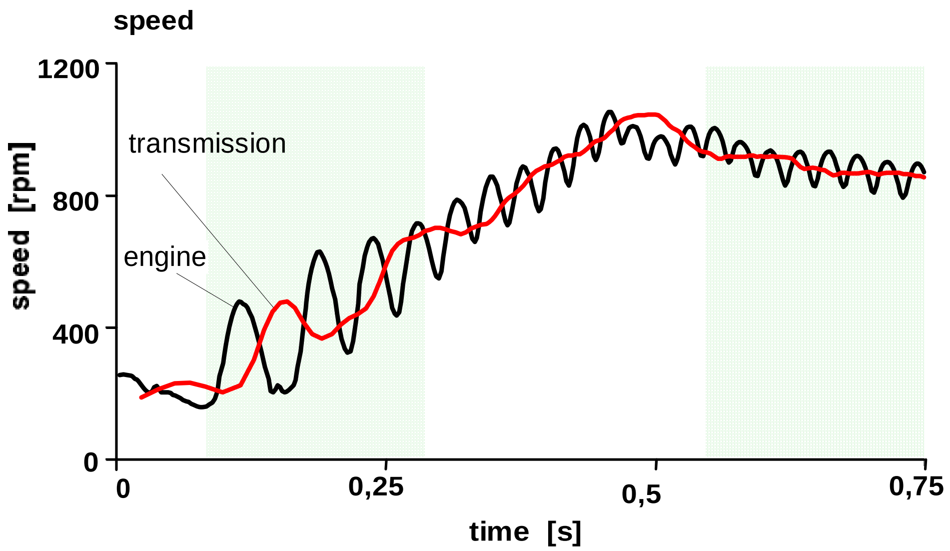


Figure 12: Arc spring damper performance during start-up

Figure 12 demonstrates these results using measurements taken during start-up. One can see that when resonance break-through occurs at relative large vibration angles, the resulting low spring rate and high damping characteristic do not induce excessive vibrations. Even in idle, when friction damping automatically decreases due to decreasing vibration angles, vibration isolation is excellent.

In summary, it is apparent that the arc spring damper easily meets all the requirements shown in Figure 5 for an ideal torsion damper, even if its characteristic curve does not match the ideal curve in Figure 5 at first sight. The seemingly contradictory requirements are met without the addition of costly design elements. At the same time the arc spring contains a self-regulating mechanism that automatically establishes an efficient combination of spring rate and damping.

Irregularities are isolated up to high engine speeds, and negative influences on tip-in/back-out performance are avoided by superimposing lower spring rates with high damping at large vibration angles.

The lubricant used in the design also effects the coefficient of friction μ, which in turn affects the arc spring damper characteristic. Figure 13 shows measurements of partial hysteresis loops for friction damping in a DMFW; the friction coefficient μ, occurring during sliding friction, was changed by selecting different lubricants.

This procedure is used during torsion damper tuning for DMFW systems. LuK, in cooperation with partners in the lubricant industry, has developed specific lubricants that permit the modification of the friction coefficient to suit the requirements of the vehicle, as illustrated in Figure 13.

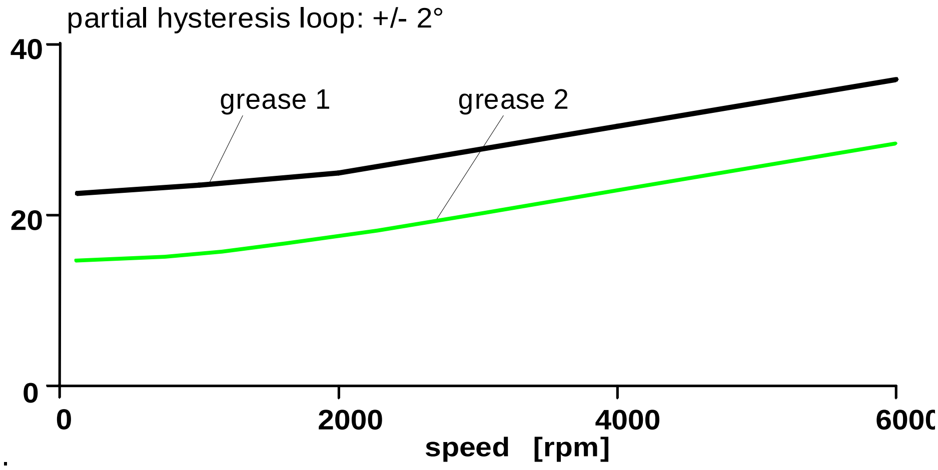


Figure 13: Influence of lubricant on damper characteristic

Damper concepts

As explained in the previous chapter, customizing the spring rate and damping characteristic has a decisive effect on the performance of the dual mass flywheel. The arc spring damper represents the ideal cost-effective solution for most vehicle drive trains.

There are exceptions, however, where vehicle tuning reveals special problems that require supplements to the arc spring system.

The three basic solutions for these special problem are as follows:

- **arc spring dampers with serial inner dampers**
- **arc spring dampers with a separate coast stage**
- **arc spring dampers in series with a torsion-damped clutch disc**

Where the isolation effect of the DMFW has to be increased for sensitive drive trains, the arc spring damper is used in series with an additional spring system - a separately functioning inner damper - as described in detail in /1/ and /2/.

Figure 14 shows such a production design. This design improves the isolation capabilities from 93% to 95 % over a DMFW that only uses arc springs (see Figure 15).

As the relationship between irregularities at the transmission input shaft and subjectively experienced transmission rattle is non-linear, this small correction in the isolation function can, in special cases, produce a definite improvement in a subjective evaluation.

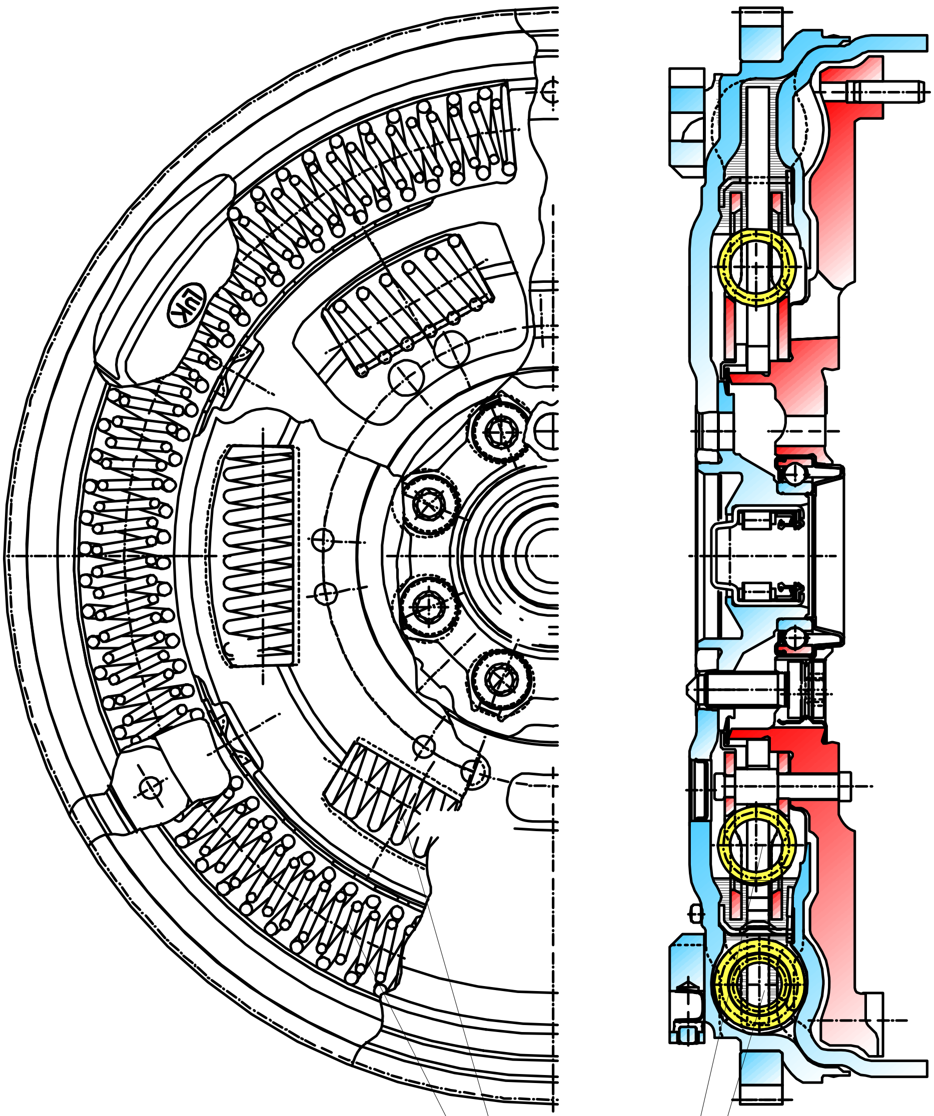


Figure 14: DMFW with independent inner damper

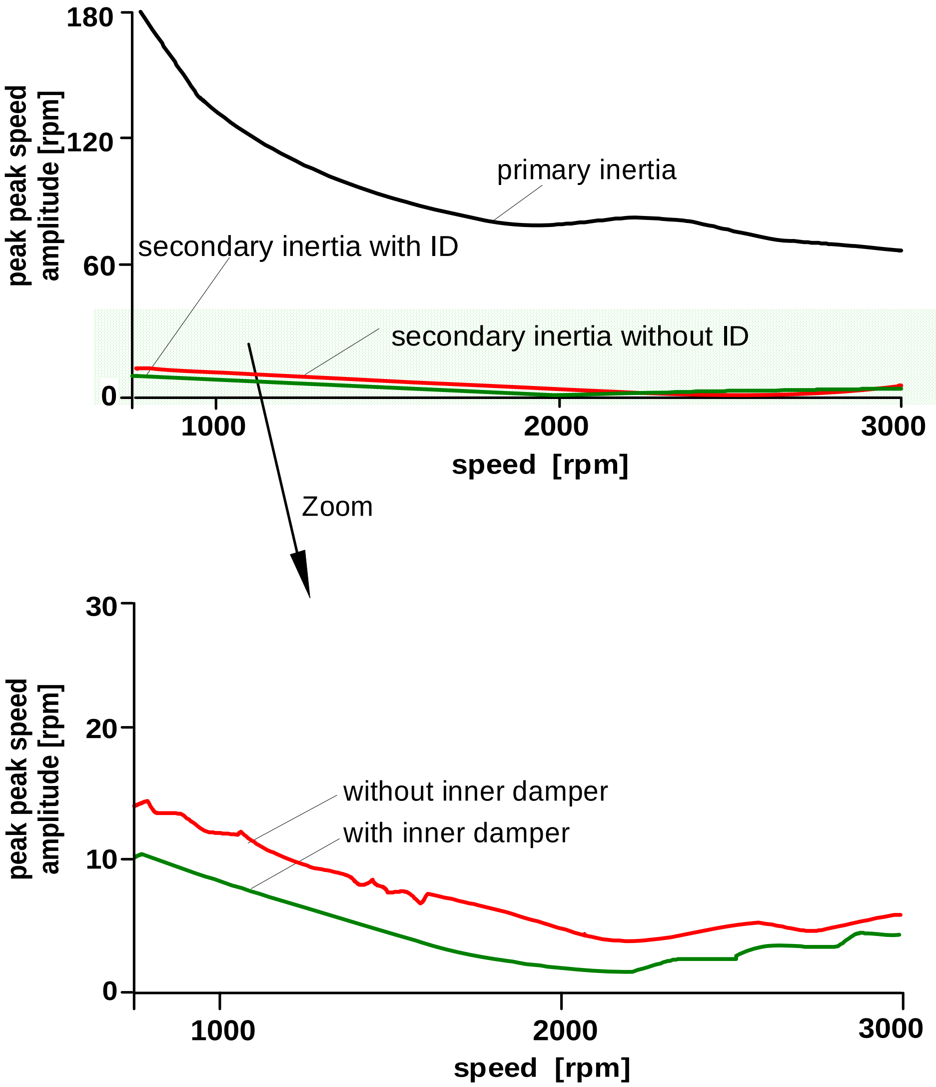


Figure 15: Engine and transmission speed irregularities using a DMFW with and without an independent inner damper (IID)

The second damper concept contains an additional coast stage. This feature is only added if the spring rate and damping characteristic of a standard arc spring do not eliminate gear rattle in the drive train during coast. This can occur occasionally in small four-cylinder vehicles. The reason is usually that the excitation is lower than the ignition frequency.

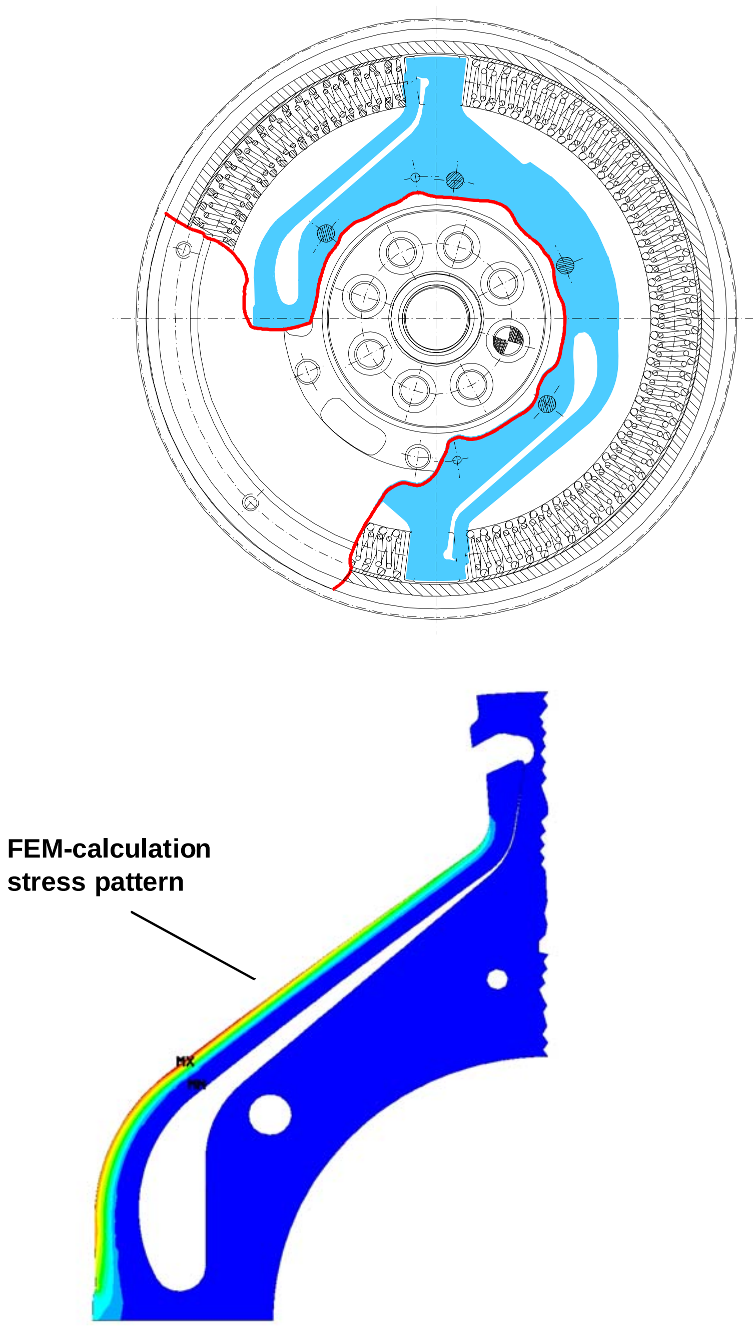


Figure 16: DMFW with flange spring

In order to effectively isolate even these low orders of excitation, an additional spring is integrated into the DMFW flange design. This modification creates a virtually frictionless solution that enables the achievement of a reduced spring rate for a typical coast torque range.

Figure 16 shows this design together with the corresponding finite element mode (FEM) calculation. Figure 17 shows a sample measurement illustrating the effect of this design.

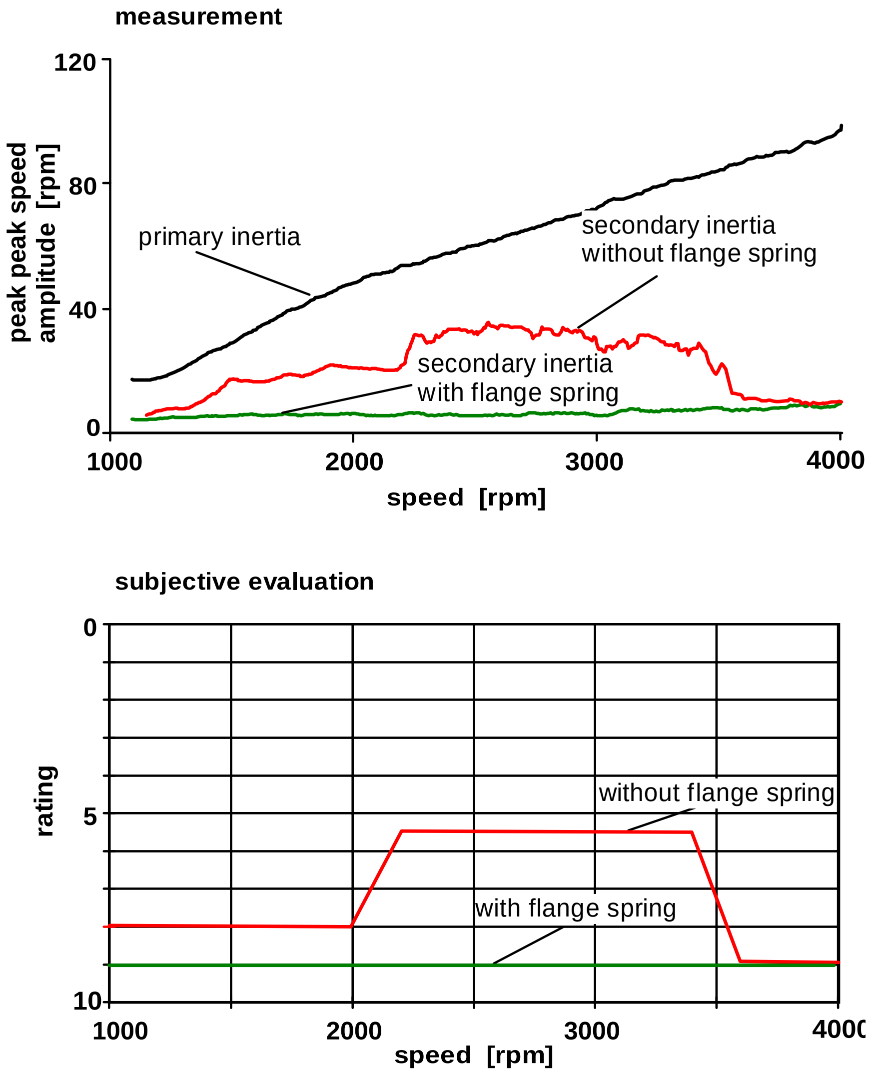
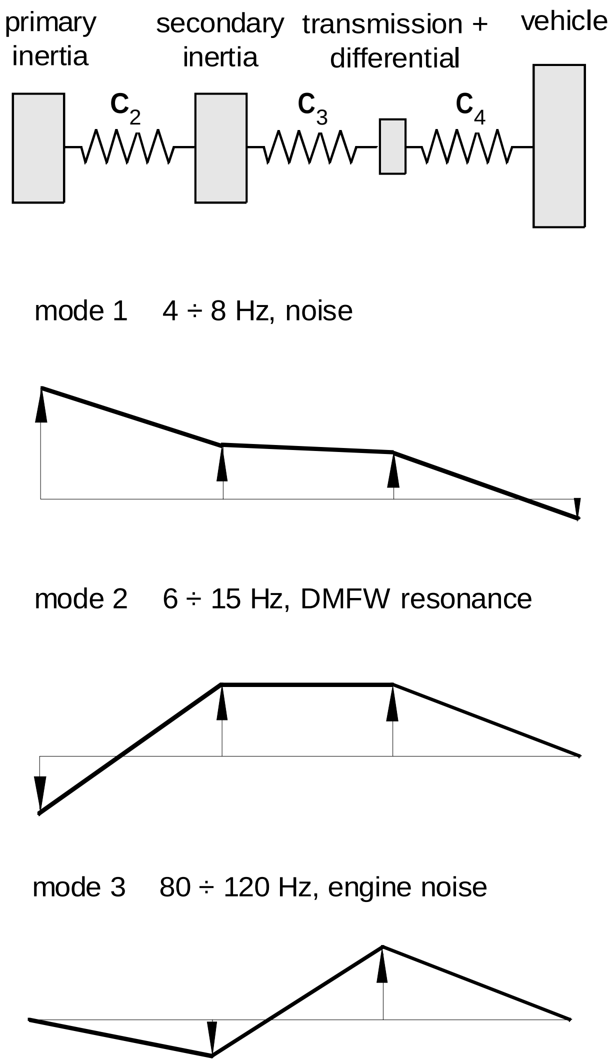


Figure 17: Effect of a flange spring in coast



influence on natural frequency		
C_2	C_3	C_4
●	○	●
●	○	◑
○	●	○

● large
◑ slight
○ without

Figure 18: Body boom due to resonance magnification on the differential - Vibration model

The third damper concept - an arc spring damper in series with a torsion-damped clutch disc - is applied when an additional vibration mode produces body boom. This is particularly the case with rear-wheel drive vehicles.

Figure 18 shows the relationship among these various factors. An additional torsional inertia was added to the vibration model. This represents the inertia of transmission, drive shaft and differential. In this model, the spring rate C_3 is mainly determined by the torsional stiffness of the transmission input shaft.

Specific combinations of parameters while driving will produce resonance. This can be detected and measured as rotational vibration at the differential (see Figure 19). This rotational vibration induces the body boom as previously stated. Figures 18 and 19 show that it is virtually impossible to influence this natural frequency by altering the spring rate in the dual mass flywheel.

By designing a torsion damper disc with a customized, relatively high spring rate it is possible to introduce the desired change in the effective stiffness C_3. This moves the natural frequency outside the driving range, which eliminates body boom effectively. LuK has also used these solutions in special applications.

In summary, the arc spring damper provides the optimum solution for almost all applications. Modular supplements to the system are only necessary in special cases in order to achieve further optimization of the torsion damper function.

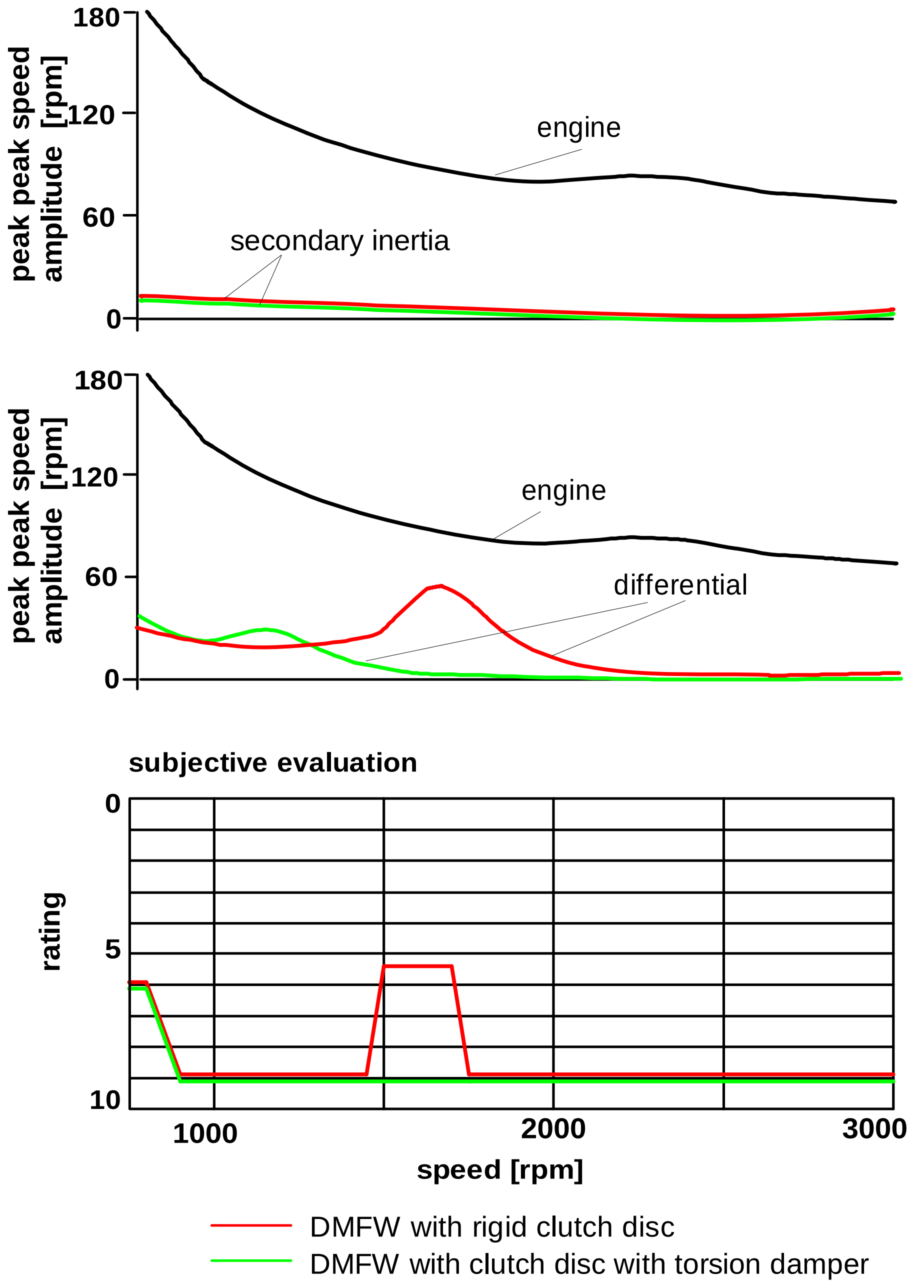


Figure 19: Body boom from resonance magnifications at the differential - Measurement and subjective evaluation

Damped Flywheel Clutch - DFC

The dual mass flywheel provides an extremely efficient system for damping the torsional vibrations in the drive train. It has established itself as an effective solution in larger vehicles.

In future, the importance of smaller vehicles with transversely mounted engines will increase. The demand for fuel efficient, low pollution engines will result in increased engine irregularities. The increase in diesel engines with fuel injection is a good example. Dual-mass flywheel systems will also be needed in small cars in order to optimize driving comfort by reducing noise.

However, increased application of DMFW systems in these vehicles requires that two basic conditions be satisfied. First, the installation space available in front wheel drive vehicles with transversely mounted engines is very limited, especially in axial direction. Second, the price range of these vehicles dictates a cost-effective solution. One must be able to justify the costs for a better torsion damper in these lower priced vehicles.

Based on this scenario, LuK is conducting a development project using targeted application of simultaneous engineering and project-oriented procedures to develop a cost-effective DMFW-system requiring a minimum of axial space /3/. This is called the DFC or compact dual mass flywheel.

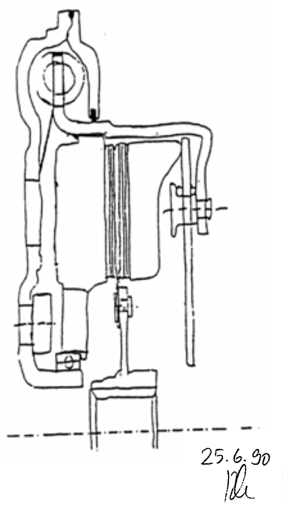


Figure 20: Concept of the Damped Flywheel Clutch

The basic idea for this evolutionary development is explained in Figure 20. The torsion damper and clutch system are integrated into one compact unit.

A space optimizing, nested torsion damper/clutch design has reduced the required axial length.

Figures 21 and 22 show two solutions designed for different vehicles. As can be seen, the clutch has been moved axially inside the damper in order to optimize space. As is typical for this new solution, the ball bearing is located on a small diameter within the mounting hole circle. The reliable arc spring design was chosen for the damper.

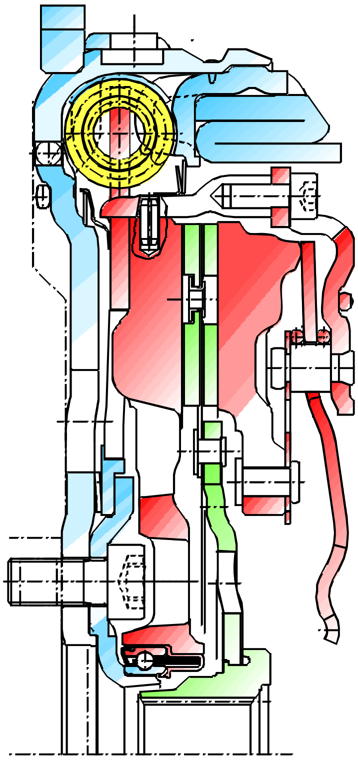


Fig. 21: DFC for 4cyl. diesel engine

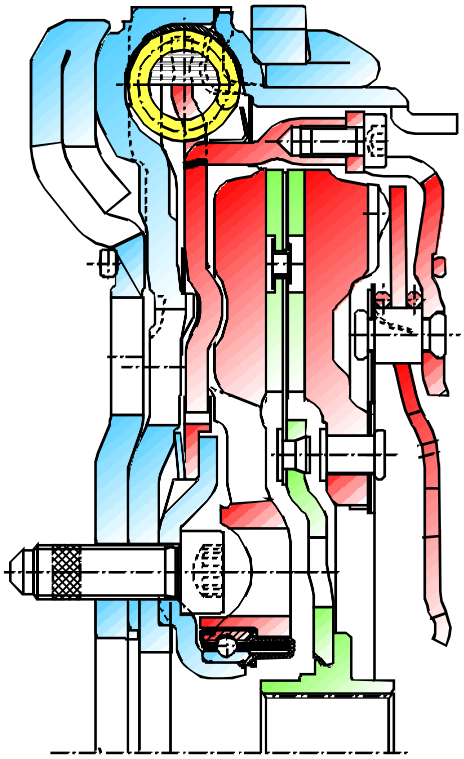


Fig. 22: DFC for 5cyl. diesel engine

The clutch can be designed either as an integrated unit or, as shown here, a bolt-together unit. The bolt-together design was developed exclusively for service requirements. The integrated design is delivered to the vehicle manufacturer complete with crankshaft bolts and installed there as a unit. Installation involves a single operation: multispindle drivers pass through holes in the diaphragm spring and the disc to tighten the crankshaft bolts. This combination of integrated component design and reduced assembly effort yields high cost reduction potential in terms of both materials management and assembly.

Both designs feature additional rings mounted on the primary side of the clutch. They are used to increase the moment of inertia of the primary side and therefore to reduce the effective torsional irregularity passed on to the crankshaft, and especially to any engine accessories. The isolation effect of the torsion damper does not require any increase in the moment of inertia on the primary side.

The additional inertial masses are manufactured from sheet steel in a cost-effective forming operation. LuK has developed special production technology in order to provide a cost-optimized and technically sound solution.

The following features represent important milestones during the development of the DFC:

- **the thermal performance of the clutch system**
- **the ball bearing**
- **the forming operation**
- **the seal design**
- **the assembly process**

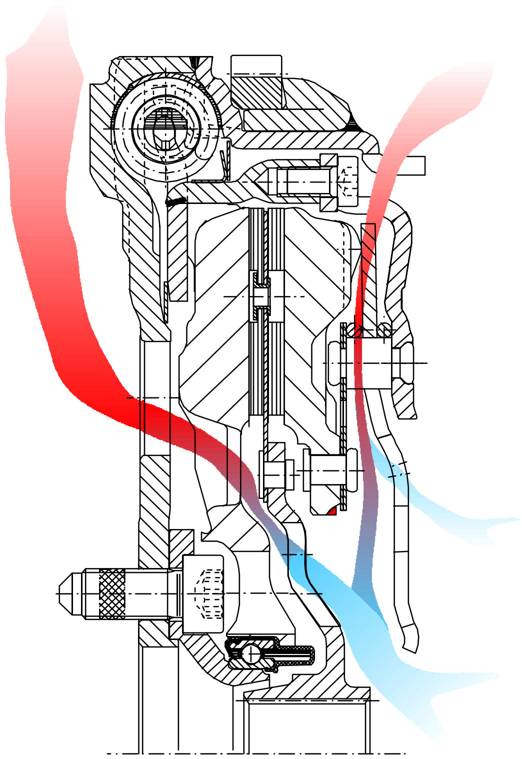
The first three points will be briefly addressed here.

The reduction in clutch diameter and the need to achieve existing performance in a more compact, integrated design make it necessary to pay special attention to the heat build-up in the system.

Flow measurements and calculations were performed in cooperation with the University of Karlsruhe, Germany and its Institut für thermische Stömungsmaschinen. Figure 23 shows the airflow inside a DFC. Individual parts have been specifically designed for best possible airflow through the unit in order to reduce peak temperatures.

The list in figure 23 shows the results of these optimization measures. The surface temperature in a specific "start" test could be reduced by 55 % through improved heat transfer. The number of starts in a special hill start test were increased from 40 to 100, and the test was broken off at 100 starts without failure.

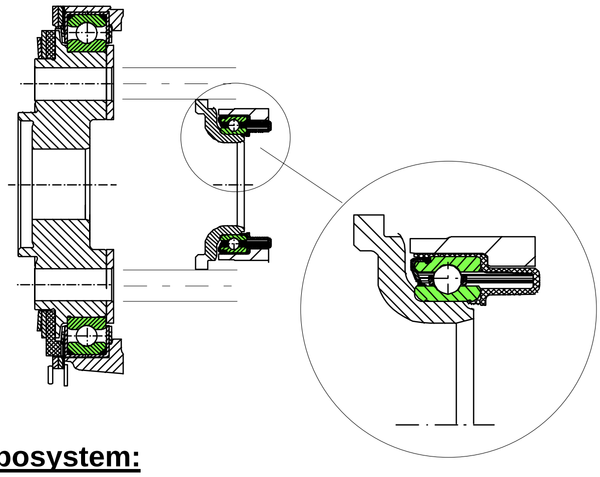
The ratio of outer diameter to inner diameter of the facings was increased to reach a suitable specific load on the facing surfaces. Numerous high stress tests have proven that the DFC system is capable of achieving required durability and overload safety specifications.



heat transfer coefficient α $[\frac{W}{m^2 \cdot K}]$	surface temperature ϑ at time t_1 [°C]	number of start-ups [-]
low	177	45
high	112	100

Figure 23: Airflow distribution and thermal performance in the DFC

The second milestone in the DFC development was the ball bearing. A totally new concept was developed, using experience gained during production of the dual mass flywheel. Optimization of the seal design, increased lubricant volume and use of a special lubricant produced a bearing with the required durability. This was achieved in spite of the drastically reduced package dimensions.

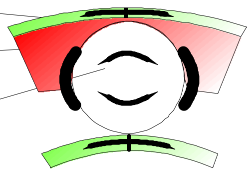


Tribosystem:

static body:
ring, carbon steel

lubricant:
grease
(base lubricant)

dynamic body:
ball bearing, carbon steel



type of friction:

combination of rolling and sliding friction with relative speed of 0 at the end positions

main requirement:

- wear protection under oscillating loads
- long lubricant service life
- no separation of lubricants under centrifugal force
- low cost

Figure 24: Tribosystem roller bearing in the DMFW

It must be stated, that in the DMFW, the load system on the bearing is totally different from that of standard applications. Figure 24 describes the tribosystem and its requirements. As the inner and outer race rotate at the same speed, the ball bearing oscillates with a relative speed of zero at the end positions. At the same time, centrifugal force exerts stress on the entire system. These operating conditions were simulated in specifically designed bearing test stands in which the effectiveness of the special lubricant and the grease cavity design were verified.

The metal forming technique used for the DFC-design was of utmost importance. Using simultaneous engineering procedures, manufacturing processes for these special plate-forming operations were developed parallel to the development of the basic system design. The goal was to ensure the most cost-effective production process for complex formed parts. Figure 25 shows two examples. The inertia ring is created using a 6-stage process whereby the outside collar on the plate blank is folded by 180 degrees.

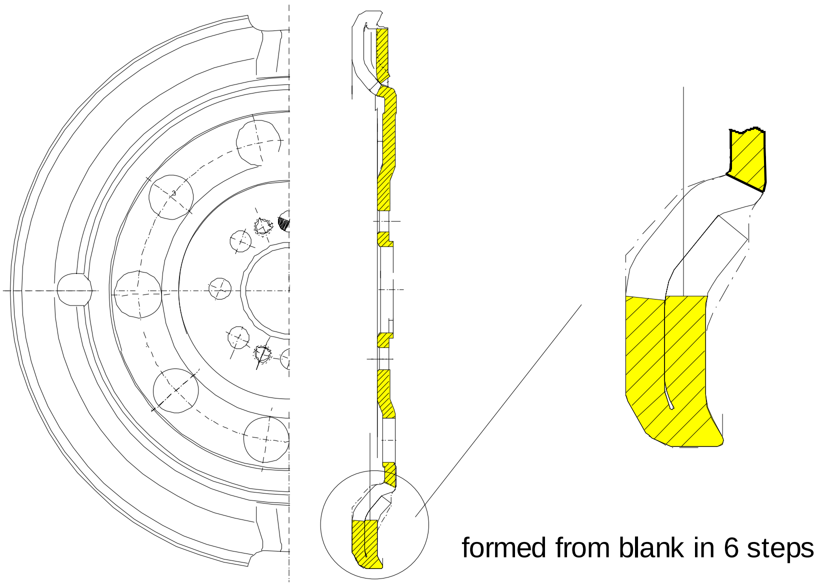
The cover ring with a circle of blind tapped holes is also manufactured from a blank, but in seven steps. The area for the tapped hole is formed by broaching into the cover edge after the forming process. Finally the screw threads are formed during assembly, when self-tapping screws are inserted into the blind holes.

This clearly shows the intensive development required to fulfill these technical demands in a cost-effective way.

Development of the DFC will not be completed for a long time. Presently further cost-reducing solutions are being designed and tested.

Figure 26 shows the DFC II. The basic space optimizing, nested design of the clutch and torsion damper has been maintained, but the positions of clutch and damper have been switched. The arc spring has a smaller operating diameter, but a much larger coil diameter in order to achieve sufficient damping capacity. The clutch has returned to its outer position to allow for a larger effective friction radius.

inertia ring



cover ring

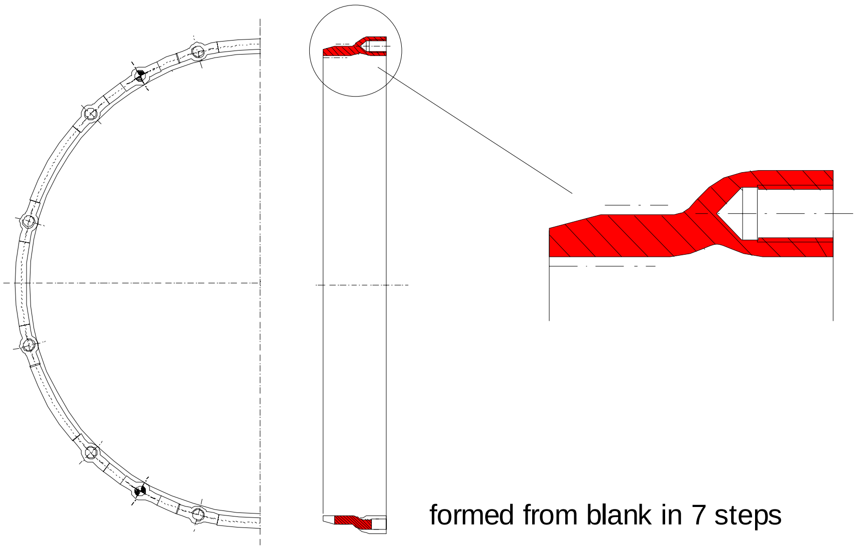


Figure 25: Forming technique for the DFC

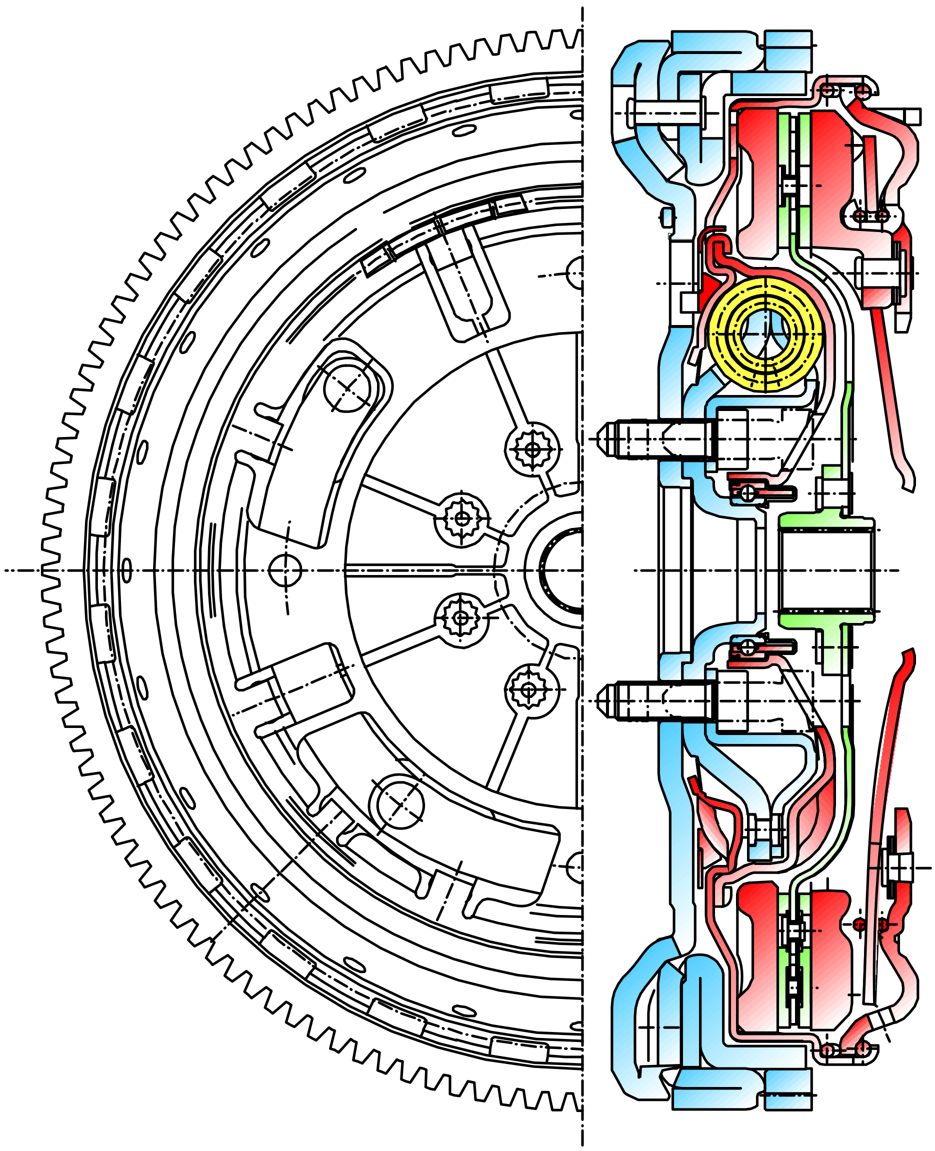


Figure 26: DFC II

The clutch bolt connection has been replaced with a bayonet catch, as shown in Figure 26. Both the secondary housing and clutch cover have interlocking tabs. A split spring ring snaps into place behind the bent tabs on the secondary housing.

The cover and the secondary housing are joined using a preloaded connection. Disassembly of the DFC for service is extremely elegant and simple. The system preload is released with a special tool and the spring

ring removed from its carriers (see Figure 27). Without any further need for tools, the clutch is released and can be changed along with the disc.

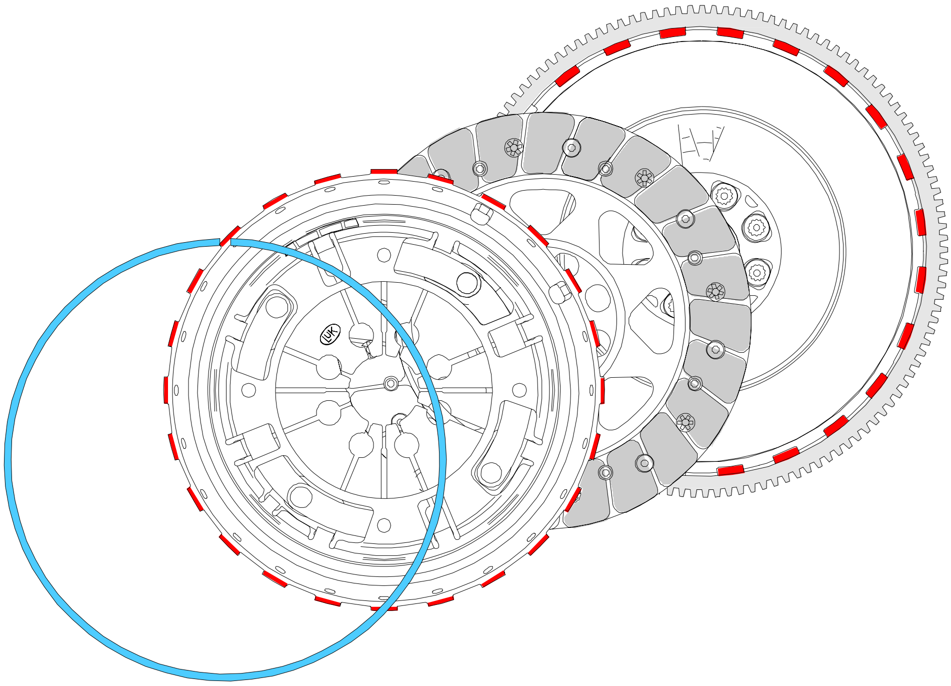


Figure 27: Clutch disassembly with the DFC II

This state-of-the-art clutch design also permits the complete clutch unit, consisting of the clutch with pressure plate and the secondary flywheel ring, to rotate (Figure 28). The load exerted by the diaphragm spring between the flywheel housing and clutch spring ring ensure torque transfer. At the same time, the system is slightly preloaded, which produces, for example, a slip torque of approximately 500 Nm for an engine of 250 Nm torque.

Consequently, the clutch unit also functions as a torque limiter, thus reducing peak torques, which normally exert an unnecessary load on the arc spring damper. The damper design can be simpler and more cost-effective. It is no longer necessary to grind the ends of the arc springs; instead, they can run out directly onto the stops formed in the flange and the damper housing.

Figure 29 illustrates the torque limiter function. Speed curves are measured during a quick engagement with the DFC II system. The center diagram in Figure 29 represents peak torques measured in the DFC with a selectively blocked slip clutch, while the lower diagram represents the clearly reduced peak torque achieved when the torque limitation feature is activated. Of course, this clutch design can also be used with an outer arc-spring design, as in the DFC I.

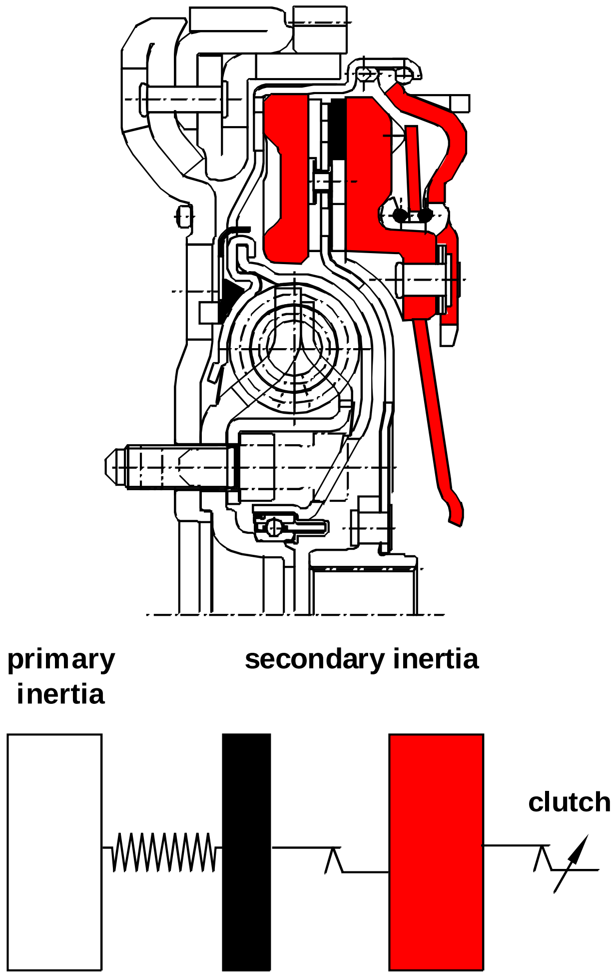


Figure 28: DFC - Torque limiter function

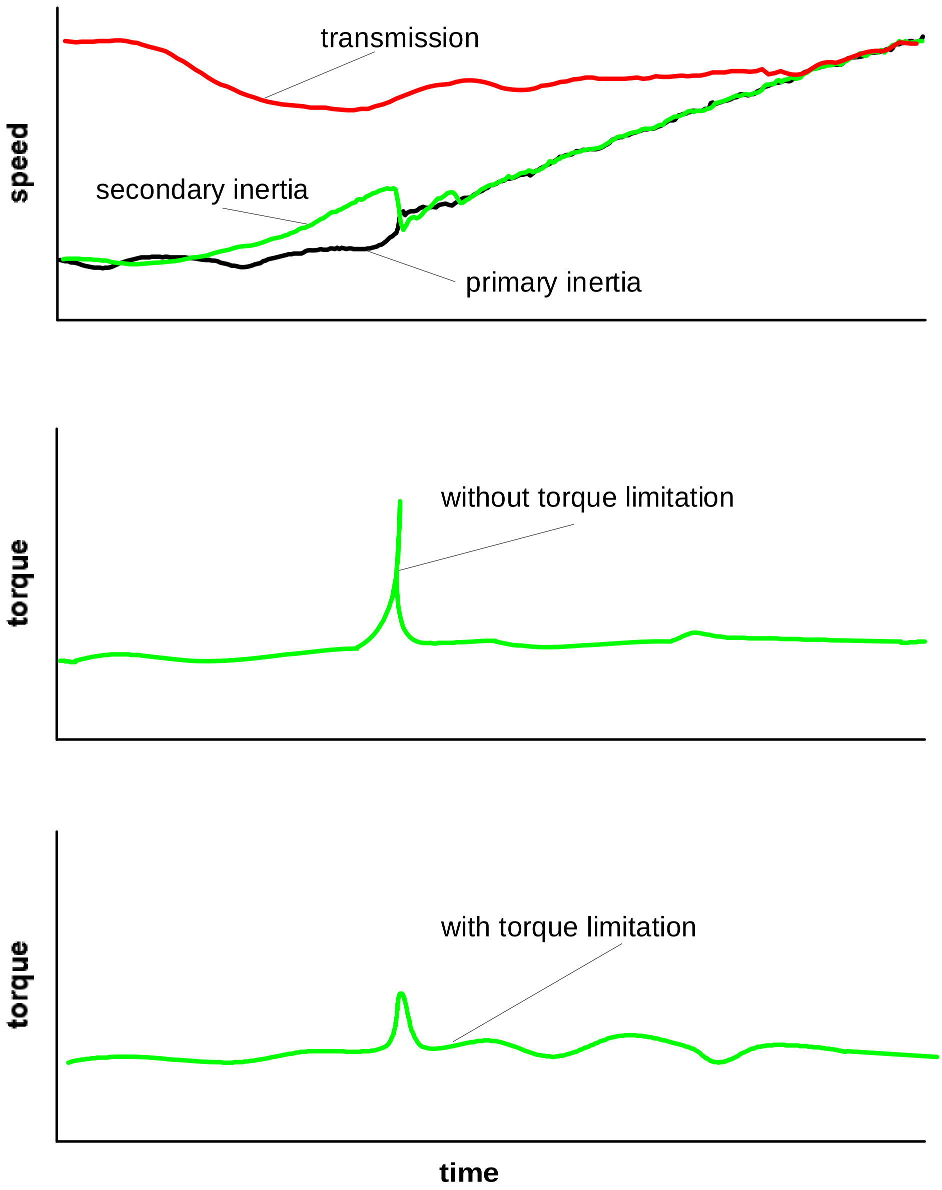


Figure 29: Torque limitation in the DFC

The clutch disc in the area of the damper needs to be specially designed. Integration of the cushion and use of newly developed clutch facing systems produces further cost reductions. Again new forming techniques for sheet-metal parts play an important role.

Endurance testing of the above described system is ongoing.

Figure 30 shows both achieved and anticipated cost reductions for dual mass flywheel systems compared to a conventional solution with the single mass flywheel, torsion-damped disc and clutch.

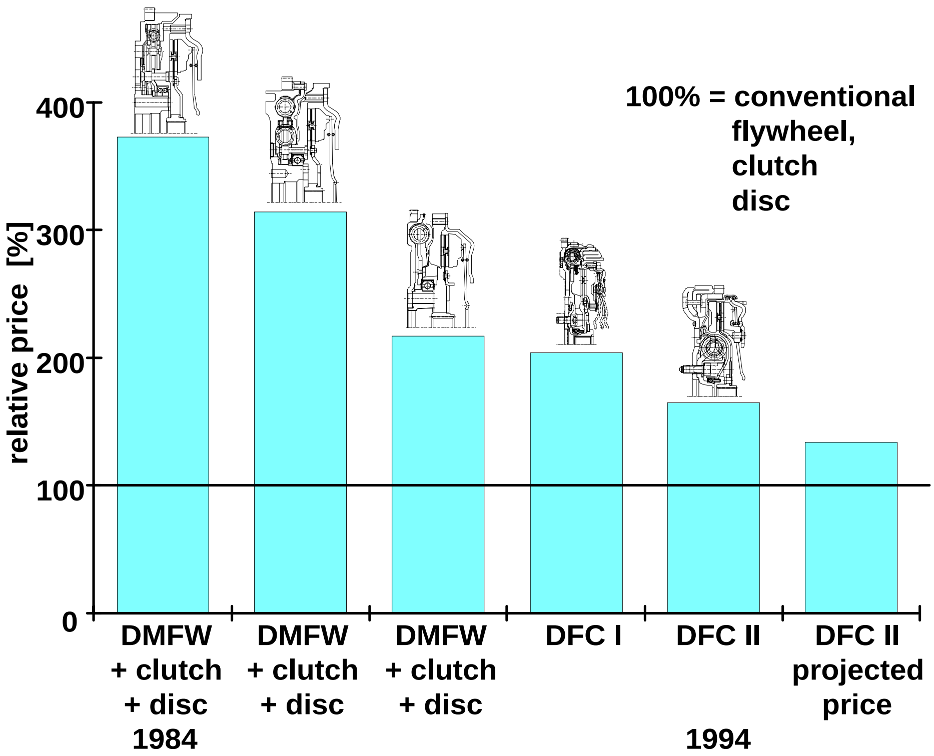


Figure 30: Cost development for dual-mass flywheel systems

A value analysis of the DFC system has demonstrated that these compact units provide all the advantages of the DMFW while incurring only minimal cost increases as compared to conventional systems. This translates into cost benefits for the vehicle manufacturer.

Summary

By use of a special damper design, dual mass flywheel systems provide decisive improvements in vehicle noise control. This increases customer satisfaction. The arc spring is the heart of the dual mass flywheel. It fulfills the requirements for an "ideal" torsion damper efficiently and cost-effectively.

Drive trains with diesel engines are controlled, without the use of idle stages, in all operating ranges. Reducing the mass connected to the crankshaft reduces the load exerted on the crankshaft and offers an opportunity for possible cost reductions in the crankshaft as well. A vehicle with a DMFW offers increased fuel economy if the following conditions are met:

- **The vehicle manufacturer reduces idle speed.**
- **The driver adopts economical driving habits by driving in higher gears at lower engine speeds**

Both of these factors can be achieved without loss of driving comfort.

DFC's or the compact DMFWs offer full dual-mass flywheel function and, therefore, improved noise control for the customer; with only minimal cost increases when compared to conventional systems. The vehicle manufacturer can simultaneously reduce subassembly count and simplify vehicle assembly by selecting modular units which perform all the three main clutch system functions.

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The self-adjusting clutch - SAC

Dr.-Ing. **Wolfgang Reik**

Introduction

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

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

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

Counterbalancing for load reduction

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

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

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

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

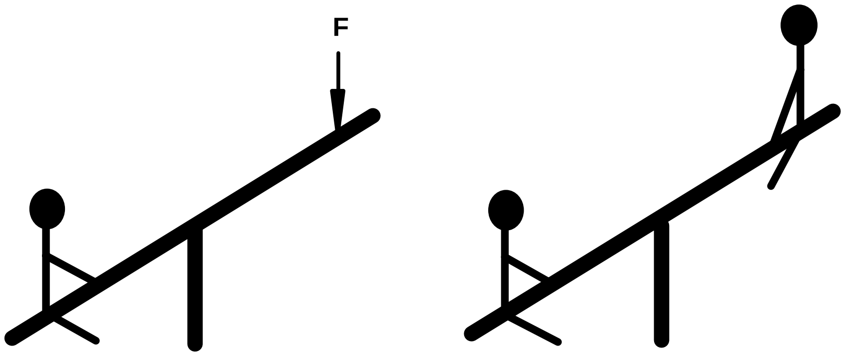


Figure 1: The counterbalance principle

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

Balancing loads in a clutch

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

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

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

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

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

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

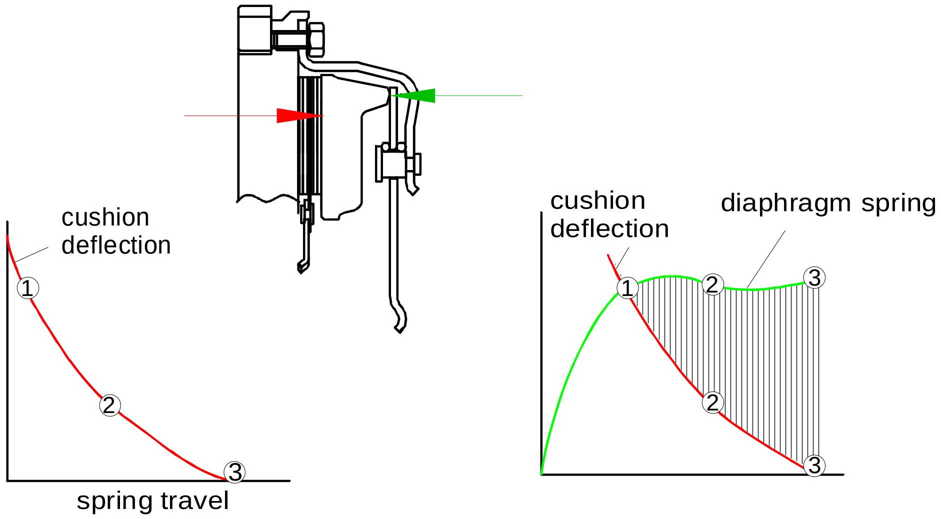


Figure 2: Loads in a conventional clutch

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

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

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

Depending on the relation between thickness and formed height in free position, the diaphragm spring can deliver various characteristic curves (Figure 4). In extreme cases a snap spring can be designed that will snap over center at a certain spring travel. The only way to return it to its original position is to apply a counter-force. At any rate, it is easy to produce a curve like the one shown in Figure 3.

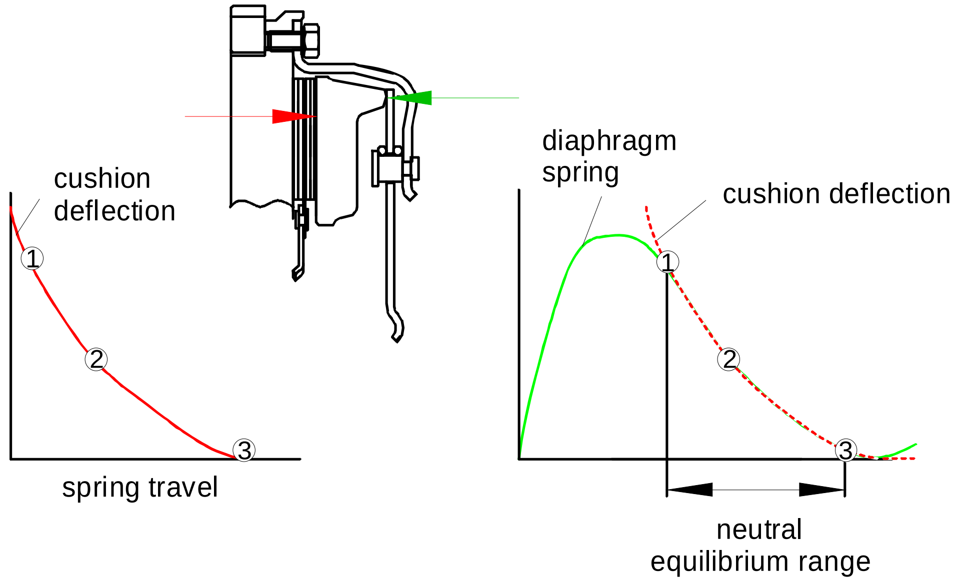
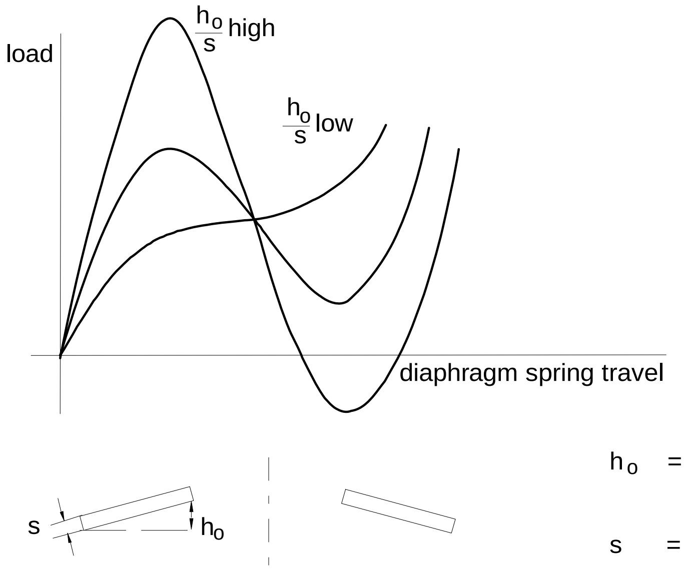


Figure 3: Load equilibrium



h_o = diaphragm spring height in free position

s = material thickness

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

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

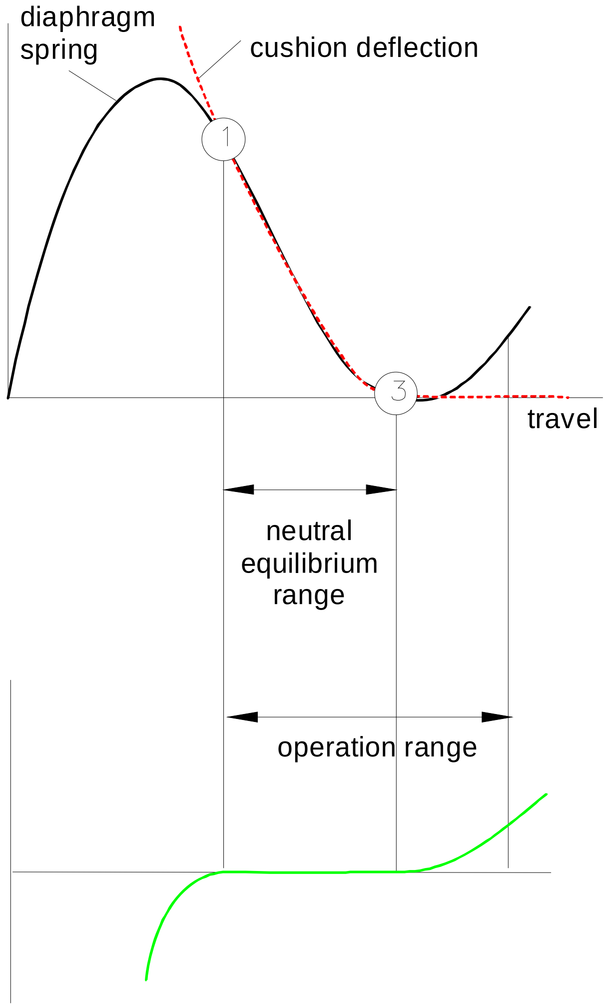


Figure 5: Equilibrium between diaphragm spring and cushion

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

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

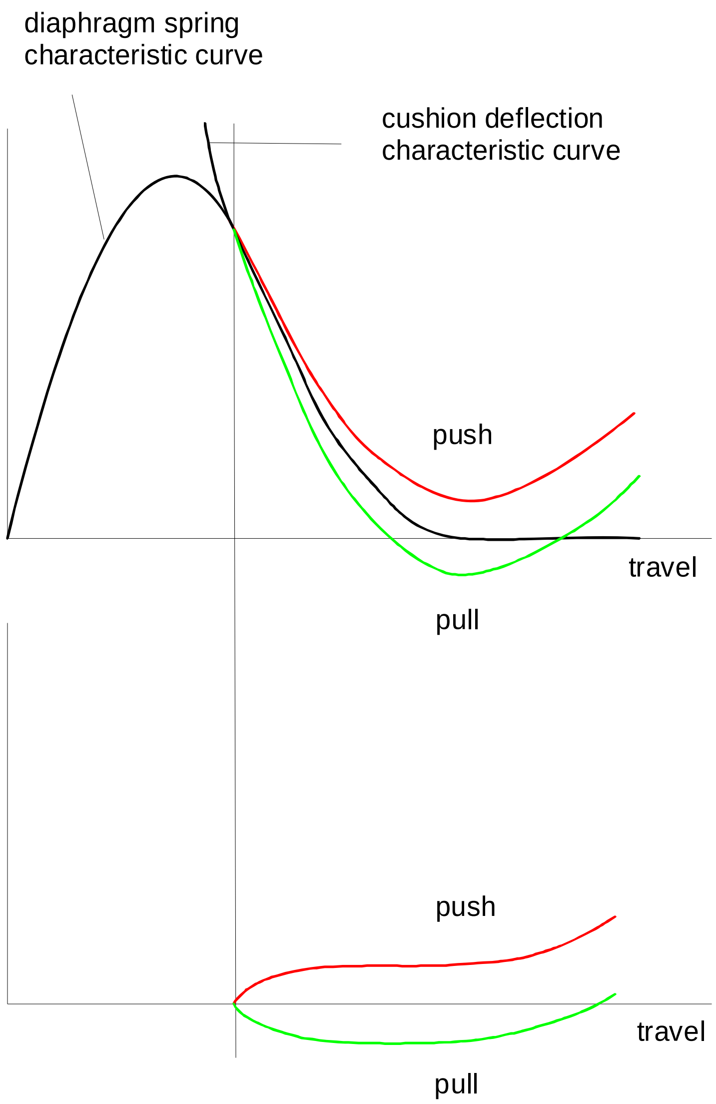


Figure 6: Disturbance of the equilibrium due to deviation from the ideal diaphragm spring characteristic curve

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

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

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

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

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

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

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

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

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

Evaluation of Figure 7 leads to the following conclusion:

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

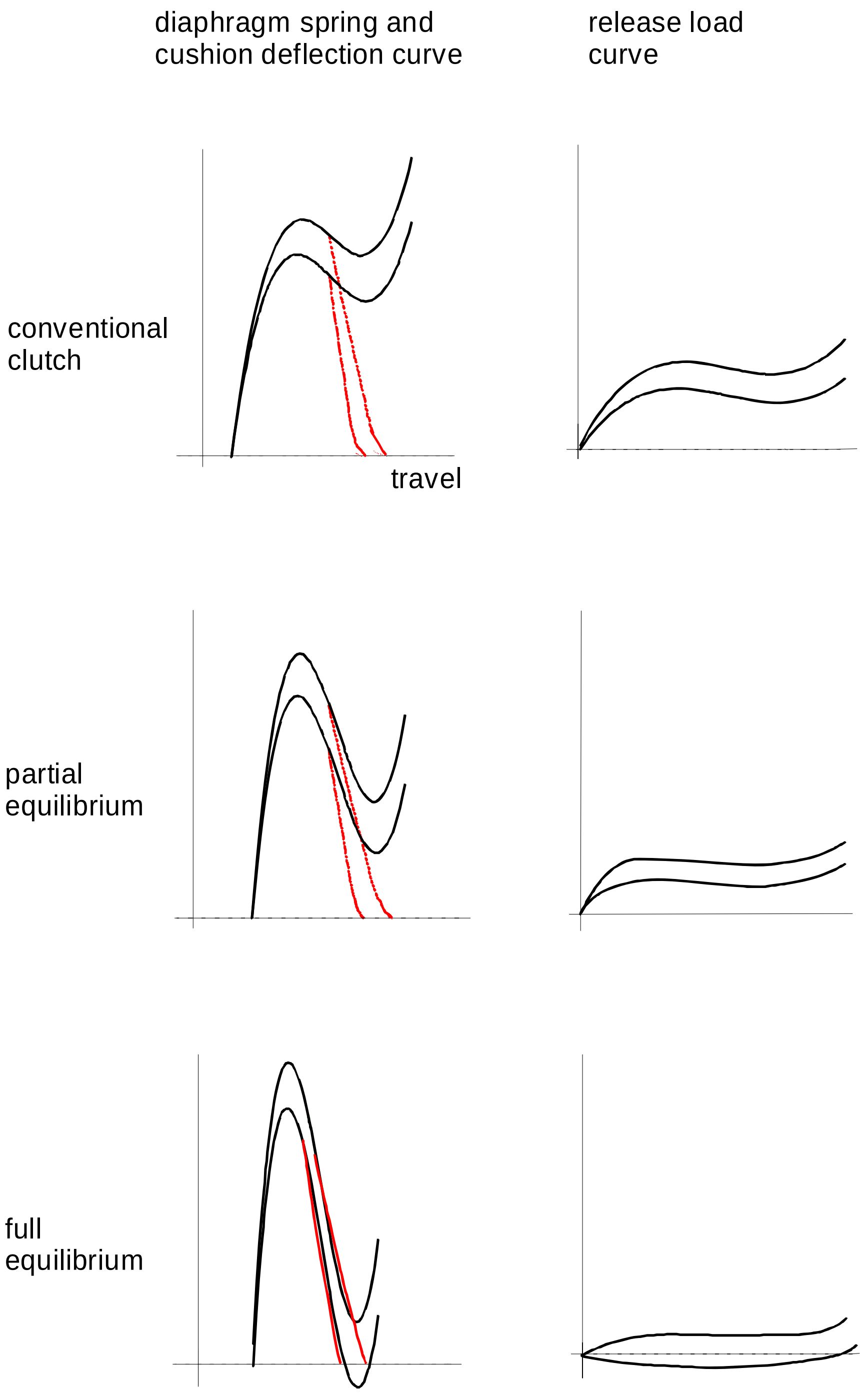


Figure 7: Influence of diaphragm spring tolerance on release load curve

Shifting equilibrium during wear

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

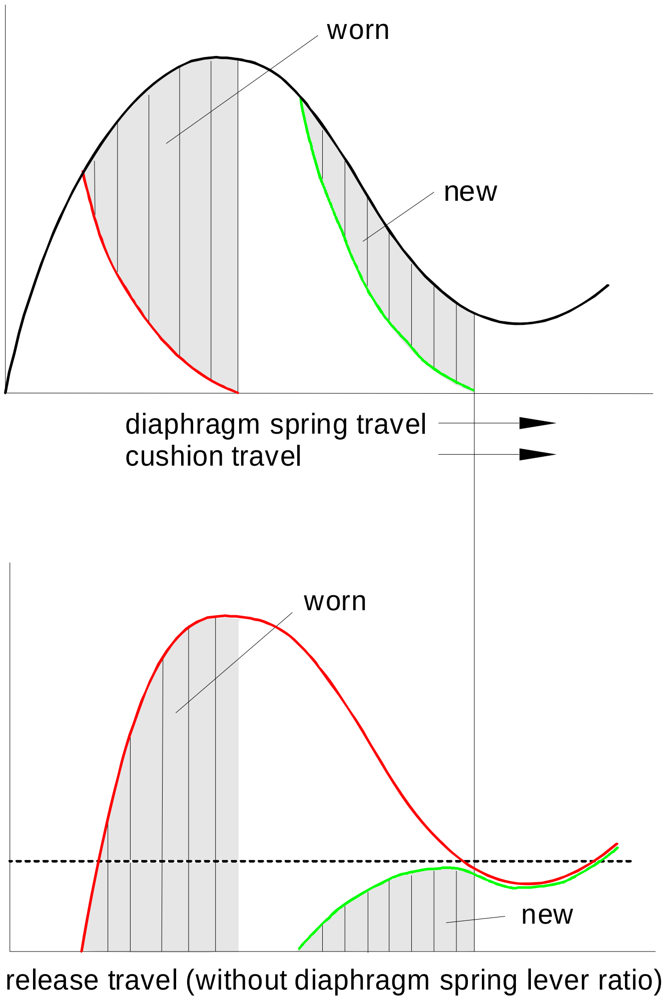


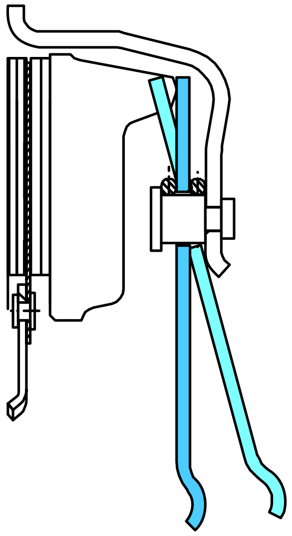
Figure 8: Influence of a shift in the operating point

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

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

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

Basically several options are possible (see Figure 9):



Facing wear changes the following

- release load
- disc thickness
- diaphragm spring position

Suitable sensors:

- load sensor
- travel sensor

Figure 9: Changes during wear

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

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

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

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

Load sensor

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

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

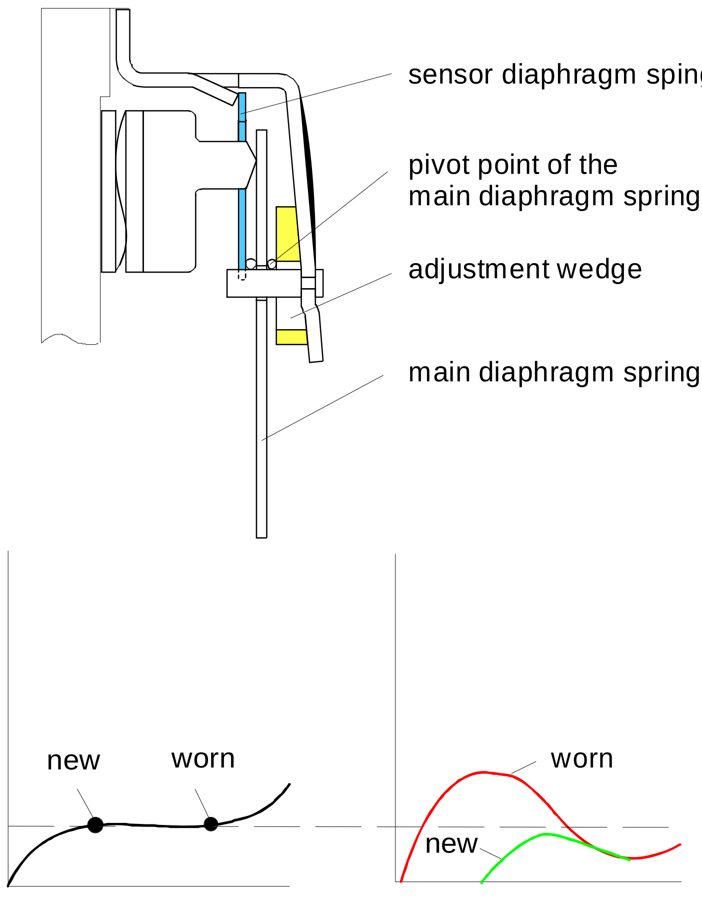


Figure 10: Principle of the self-adjusting clutch

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

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

Design of a wear-adjusting clutch with a load sensor

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

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

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

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

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

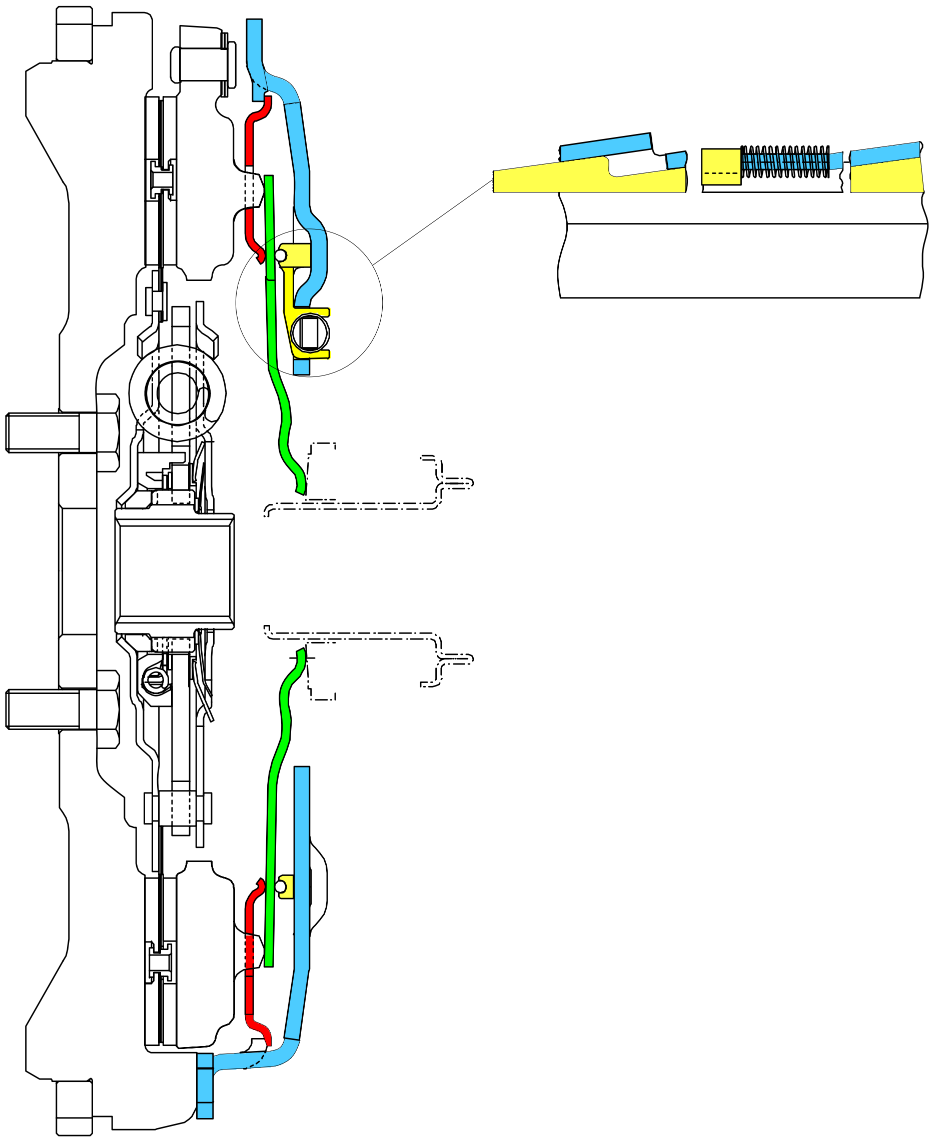


Figure 11: Self-adjusting clutch (SAC)

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

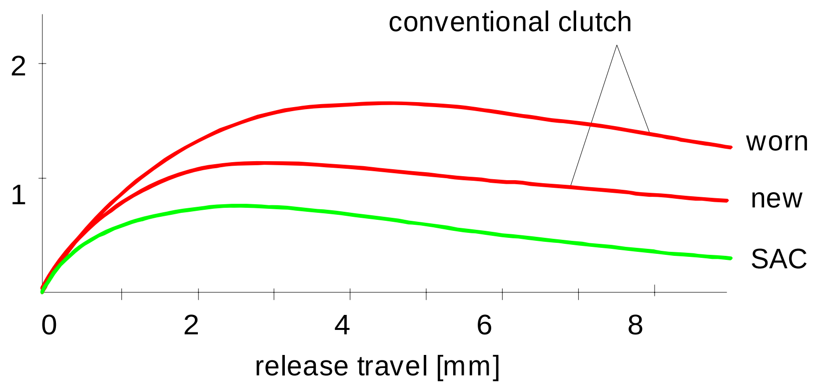


Figure 12: Comparison of SAC release loads with those of a conventional clutch

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

Figure 13: Effective characteristic curves

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

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

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

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

Tolerance considerations

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

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

Features that are more important than in a conventional clutch:

- cushion characteristic
- diaphragm spring minimum
- sensor diaphragm spring load

Features that are less important than in a conventional clutch:

- diaphragm spring maximum
- pressure plate height
- cover height
- disc thickness
- facing thickness

Figure 14: Features for wear-adjusting clutches

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

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

The SAC requires that the designer reevaluate all the tolerances.

Cushion deflection requirements

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

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

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

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

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

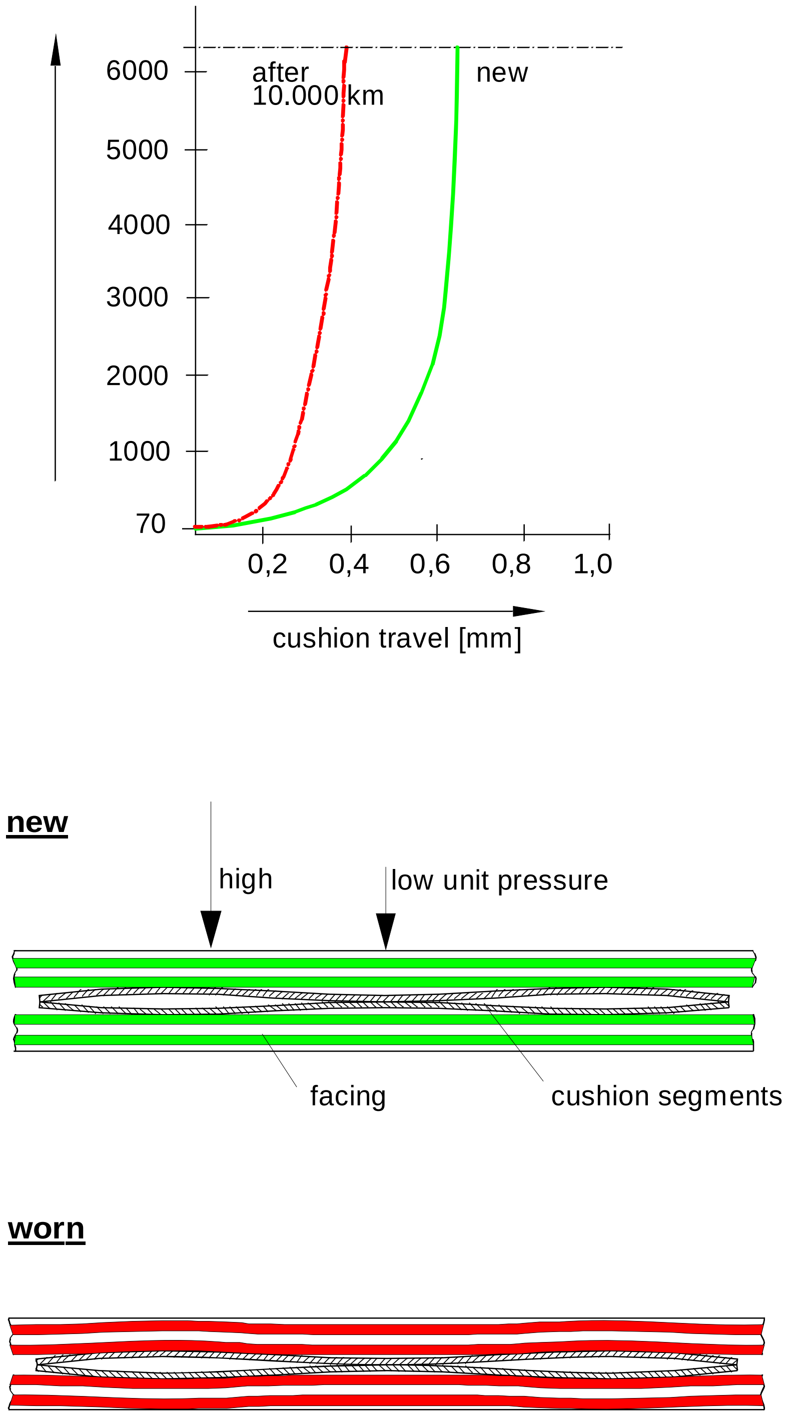
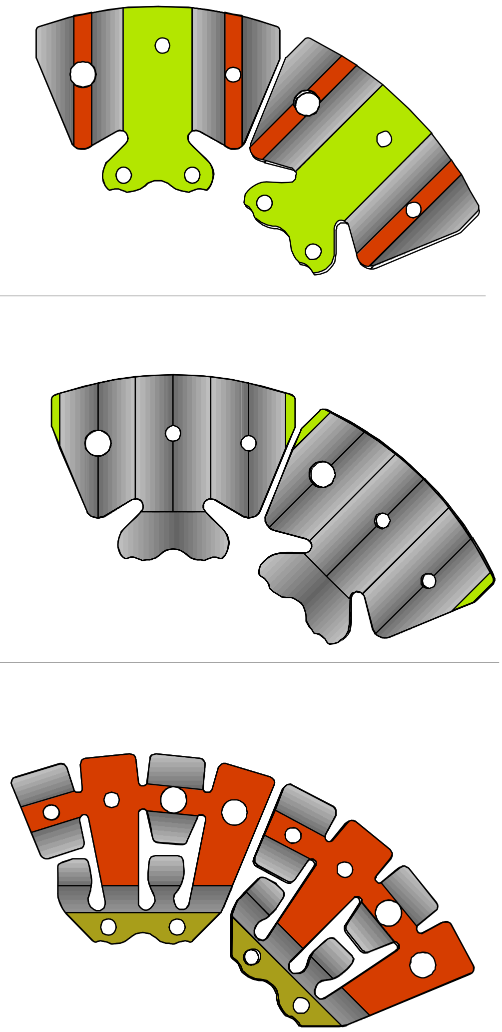


Figure 15: Changes in cushion deflection over the life span of the clutch



conventional segment

triple waving

feathered segment

Figure 16: Segment variations

Release system requirements

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

If the clutch is overstroked , which means the permissible release travel has been exceeded by 30 %, the release load (see Figure 8) increases towards the end of the release travel and causes an unintended adjustment even without wear on the facing. This in turn causes the cushion deflection characteristic curve (Figure 8) to move too far to the right and the intersection point of the two curves, representing the clutch clamp load, to be set too low. Then the clutch can slip.

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

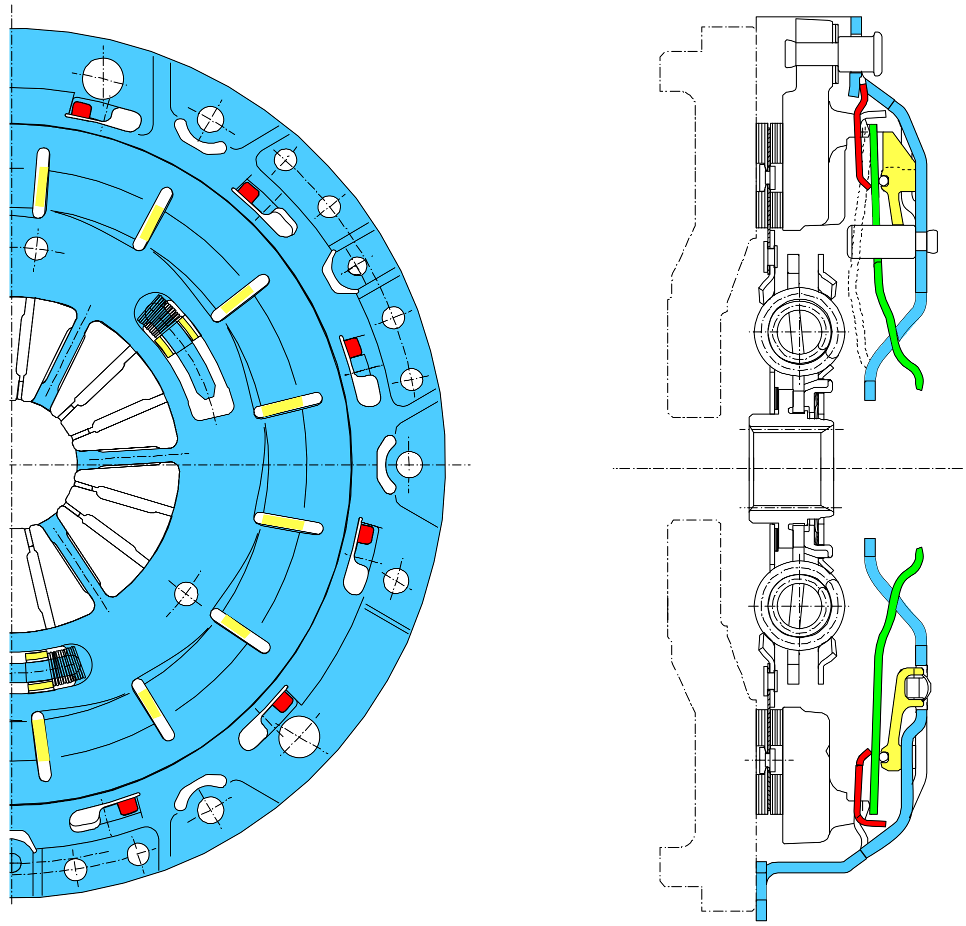


Figure 17: SAC-production model with mechanical stop

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

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

Current developments in durability testing

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

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

Summary

The self-adjusting clutch offers two main advantages:

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

This results in other possible secondary advantages, for instance:

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

The Mechanical Central Release System for the SAC - an Alternative ?

Dr.-Ing. **Ad Kooy**

Introduction

In the past there has been an increase in the application of hydraulic release systems, especially in mid-size vehicles. The reasons are evident:

- a simplified automatic wear adjustment feature
- ease of installation
- good efficiency at high release loads

Figure 1: Advantages of the hydraulic release system over the mechanical release system

This trend has motivated several manufacturers to develop a concentric ring cylinder, positioned around the transmission input shaft, which acts as a slave cylinder (Figure 2). Compared to the semi-hydraulic release system with external slave cylinder (Figure 3), the new system has the advantage of eliminating the release fork. Several automotive manufacturers have introduced these CSC-systems (Concentric Slave Cylinder Systems) into production.

But why not develop a mechanical central release system as an alternative to hydraulic systems? The advantages of the centralized configuration could be used and the elimination of the release fork could give a definite cost reduction in comparison to the conventional hydraulic system. Development of the SAC puts such a possibility in a new light.

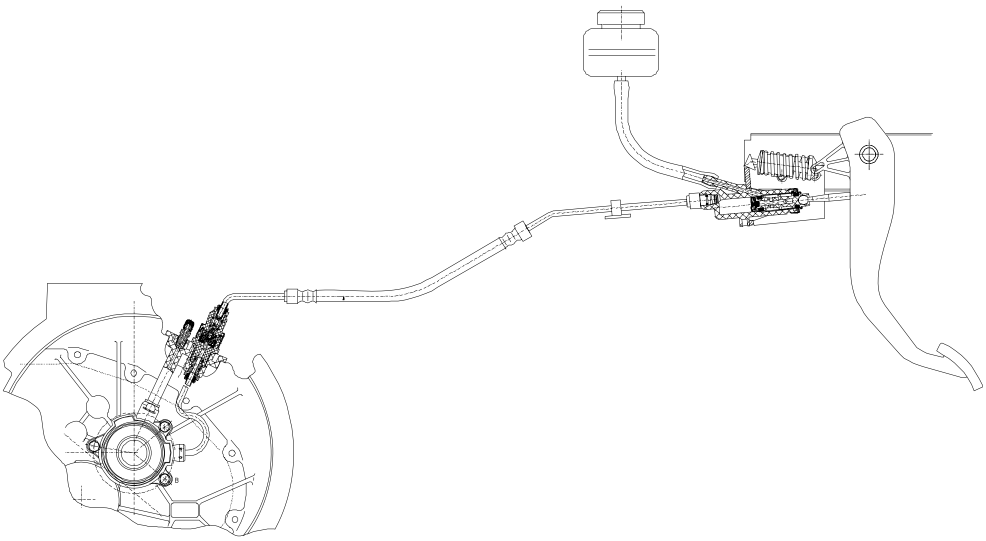


Figure 2: Operating system for a hydraulic release system, INA design

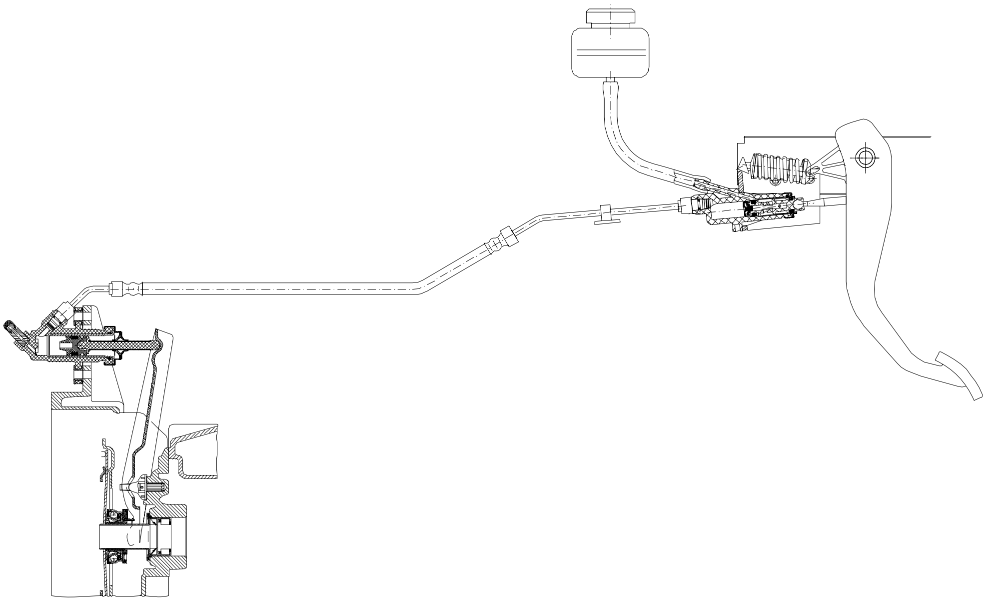


Figure 3: Semi-hydraulic release system, INA design

Release system requirements for a SAC application

The SAC design entails an obvious change in the requirement list for a release system (Figure 4). As demonstrated in the previous presentation, the SAC lowers the release loads. A release system must have the ability to work effectively and to reset itself reliably under such low loads. This requirement favors the advantages of mechanical systems because they develop less friction than hydraulic systems with their seals. Lower requirements can be placed on the stiffness of the release system without sacrificing efficiency.

- Reduces the release load by 50 %

  - Favors the mechanical system thanks to lower friction
  - Lower stiffness is allowable

- During wear the diaphragm fingers travel in the same direction as the release motion

  - simple mechanical readjustment mechanism possible
  - wear travel reduced by 70 %

Figure 4: Influence of SAC on release system requirements

A further difference becomes obvious over the lifetime of the clutch. As the facings wear, the diaphragm spring finger tip travels in the same direction as the release motion. As discussed later, simple automatic adjustment mechanisms can be designed to compensate for this. In addition, by adjusting for wear in the SAC, the actual change in finger height due to the facing wear is reduced by 70 %, which saves installation space.

Requirements for the most basic possible release system must be defined primarily based on these new, SAC-induced conditions (Figure 5).

A low release load needs to be combined with low friction in order to assure reliable reset of the bearing. At such low release loads, sufficient stiffness should not present a problem. To guarantee the function of the SAC, release bearing overtravel must be avoided.

As with all hydraulic systems, this design must provide for an automatic adjustment for facing wear.

Function
- Low friction, sufficient stiffness
- Reset at low release loads
- Prevention of overtravel
- Automatic repositioning of the release bearing in the direction of the engine during wear
- Reduction of pedal vibrations

Durability
- Insensitive to contamination
- Ambient temperatures up to 200°C
- Little wear

Installation
- No need for more space
- Easy cable routing and installation in the vehicle

Costs
- Less costly than conventional release systems because fork is eliminated

Figure 5: Release system requirements

Obviously the release function must be maintained over the lifetime of the clutch, despite high ambient temperatures and contamination from facing particles.

Installation conditions are often very tight, which does not allow for additional space in the vehicle for a new release system. Furthermore, the need for fast, easy installation in the vehicle dictates that the least possible number of subassemblies are used.

With a goal of meeting the previously mentioned requirements, different basic release principles can be compared.

Mechanical release system principles and adjustment systems

In the first step, it was investigated how the actuation of a cable can be used to generate an axial movement of the release bearing without using the usual release fork. Despite the sometimes bad reputation of the cable, it was chosen because the low load, typical for a SAC, clearly improves durability and reduces wear. At first LuK investigated applicable mechanical alternatives systematically.

One of those alternatives was polygonal linkages. Such linkages produce a linear or almost linear movement. Figure 6 shows an example used, among others things, for ship loading harbor-cranes. Selection of the geometrical polygon dimensions and the connection of the cable permits the lever ratio to be altered within a limited range. Space problems would be expected with this concept and also high production costs because a minimum of five joints is required.

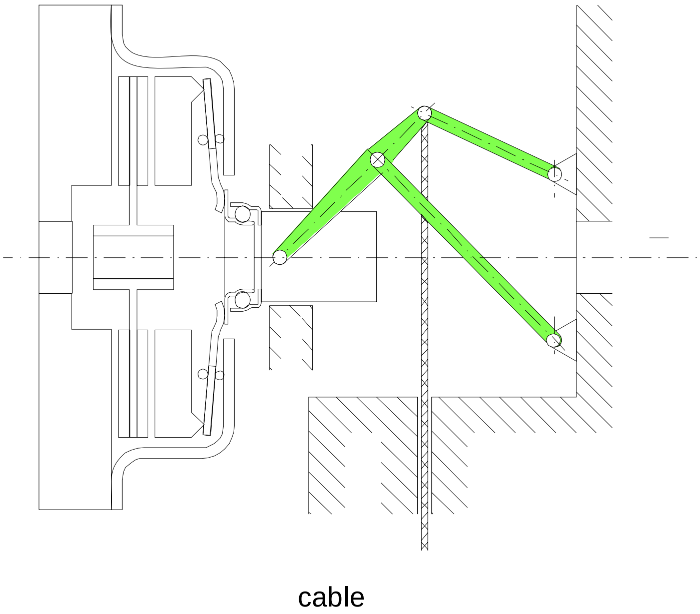


Figure 6: Polygonal linkage for release systems

Another concept is the use of two ramps (Figure 7). When the upper ramp is displaced relative to the fixed lower ramp, the movement is transferred into an axial motion, depending on the ramp angle. Use of balls or rollers keeps the friction low.

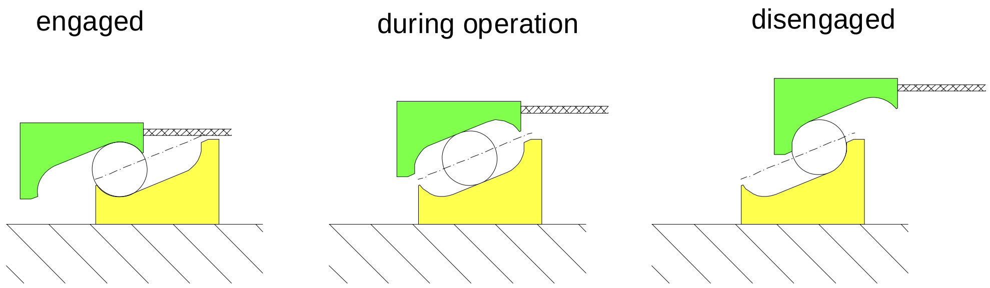


Figure 7: Ramp principle

This basic design can be executed in very different ways. Figure 8 shows a design presently in development. Here two ramp units are located between the release bearing and the transmission wall.

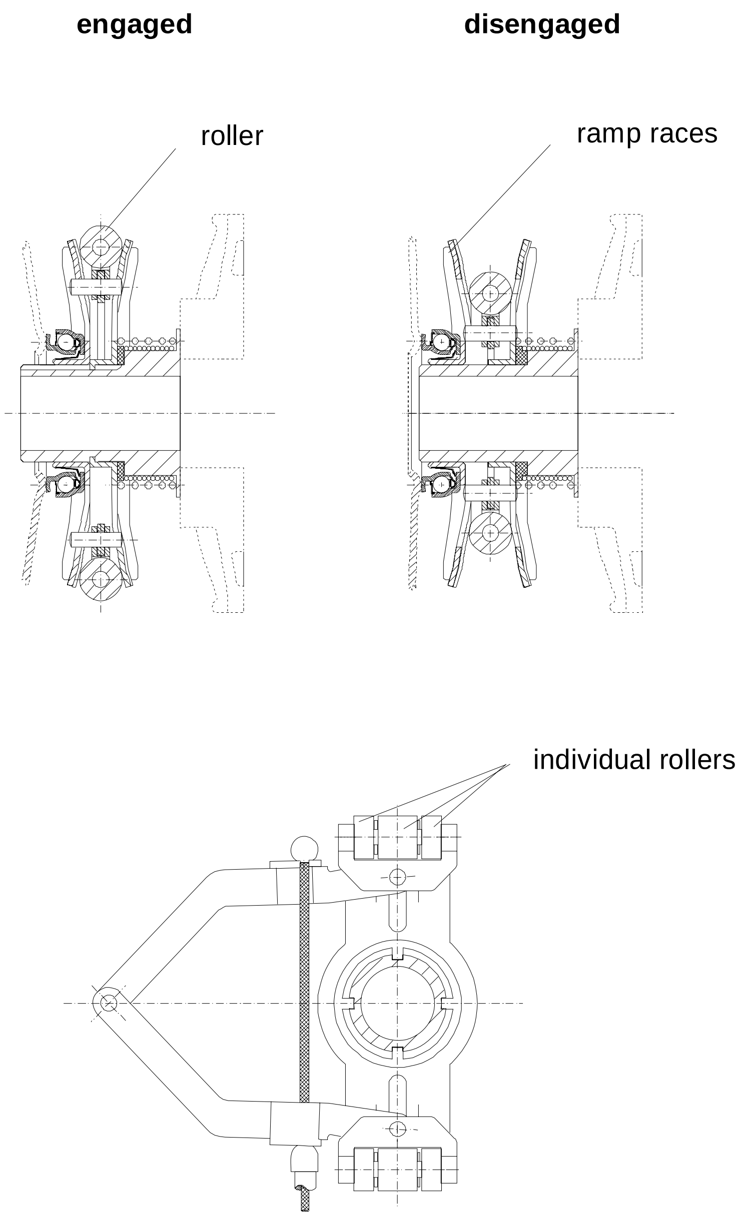


Figure 8: Radial ramp arrangement

When the cable is activated two opposing rollers are synchronously moved towards each other via a mechanical coupling. This spreads the ramp races and produces an axial movement of the release bearing.

In order to achieve a rolling motion and a symmetrical ramp load, each roller unit is divided into three individual rollers. The center roller rolls against one ramp race, and the two narrow outside rollers roll against the opposing ramp race.

Therefore, each ramp race is guided via a three-point support, which prevents a misalignment of the release bearing. This assumes a synchronous movement of the upper and lower rollers shown in Figure 8. Additional components must be added to the design to ensure this.

One further possibility for the configuration of ramp units is shown in Figure 9. Here three circular ramps in the form of asymmetrical grooves were situated in a fixed ramp ring and a rotating ramp ring. When the rotating ramp ring is turned, a screw-type motion occurs similar to the action of a triple-threaded screw; the balls minimize the friction and take over the radial as well as the axial guidance of the release bearing mounted to the upper segment. While the fixed ramp ring is mounted on the transmission wall, the rotating ramp ring can be actuated by a cable producing the axial release movement. This simple principle has proved reliable for a long time in the release system for motorcycle clutches.

Figure 9: Circular ramp arrangement

The three balls create a three-point support which clearly defines the position of both ramp rings with respect to each other. Besides torque, the cable also creates an unwanted transverse load, which, however, only amounts to approximately 40 % of the release bearing load, depending on the ramp angle. This transverse load is transmitted by two of the three balls.

When the ramp angles of both parts are constant, the release bearing will not tilt, even if one of the balls shifts. As shown in Figure 7, the ramps can be designed so that the balls at both ends of the ramp act as stops, which reliably prevents overtravel damage to the SAC and therefore does not require a stop-tab in the SAC cover.

The balls are repositioned each time they are used as a stop. A ball cage is therefore not necessary, even if varying ramp angles are used to reduce the maximum and to increase the minimum release load. In that case the ramps need to be designed so that the contact points of both ramps on the ball are located opposite each other over the complete travel. This allows the ball to easily follow the desired path because it prevents the creation of reactive loads (Figure 7).

The advantage of this ramp ring-type release system lies both in the compact concentric design and in the three-point support, which ensures reliable function with few parts.

In order to compete with hydraulic systems, a mechanical release system must be able to automatically adjust for changing diaphragm spring finger height as the facing wears.

In the SAC, unlike in a conventional clutch, as the facing wears the diaphragm spring fingers travel in the same direction as during disengagement. A relatively simple mechanical readjustment mechanism similar to the SAC-adjustment (Figure 10) can be designed to compensate for the change in diaphragm spring finger height. To accomplish this, ramp rings - preloaded with springs - compensate when a gap occurs between the diaphragm spring fingers and the release bearing. As in the SAC-design, the self-locking feature prevents a return to the previous ramp position by using a flat ramp angle. This mechanism operates automatically and continuously.

The mechanical central release system (MCR) combines this readjustment with the release concept shown in Figure 9.

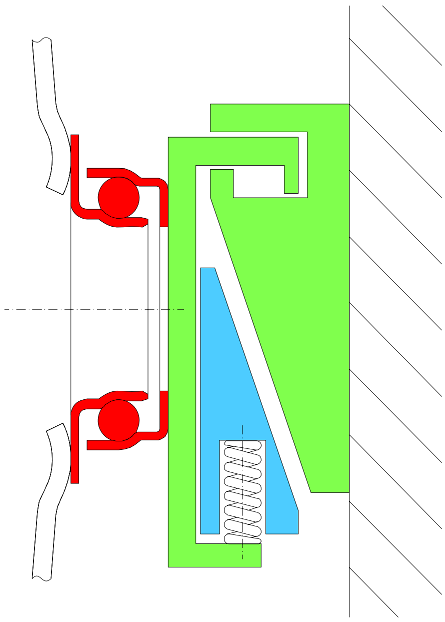


Figure 10: Finger height adjustment concept

The mechanical central release system

Figure 11 shows a layout. The complete release system consists of a few, relatively small detail parts that are inexpensive to make. Besides the release bearing with its diaphragm spring, three formed sheet-metal parts, 3 balls, 2 plastic rings, one coil spring and a simple curved wire are needed. LuK can rely on its long experience in the production of clutches and dual mass flywheels when creating the tools for these stamped parts. How the individual parts work together is described in the following section.

The system can be divided into units with the following functions:

The fixed and rotating ramp rings are responsible for the release function.

The fixed ramp ring is fastened to the transmission via screws or grooved pins. When the rotating ramp ring turns, it moves axially over the three greased balls. At the end of the turning motion, the balls hit stops, which limits the release travel (Figure 12). A wire leading to the housing wall activates the ramp ring. Due to the hook form of the wire, the cable can be easily connected or disconnected from outside. This wire also protects the cable from contamination by facing particles and from excessive transmission-housing temperature.

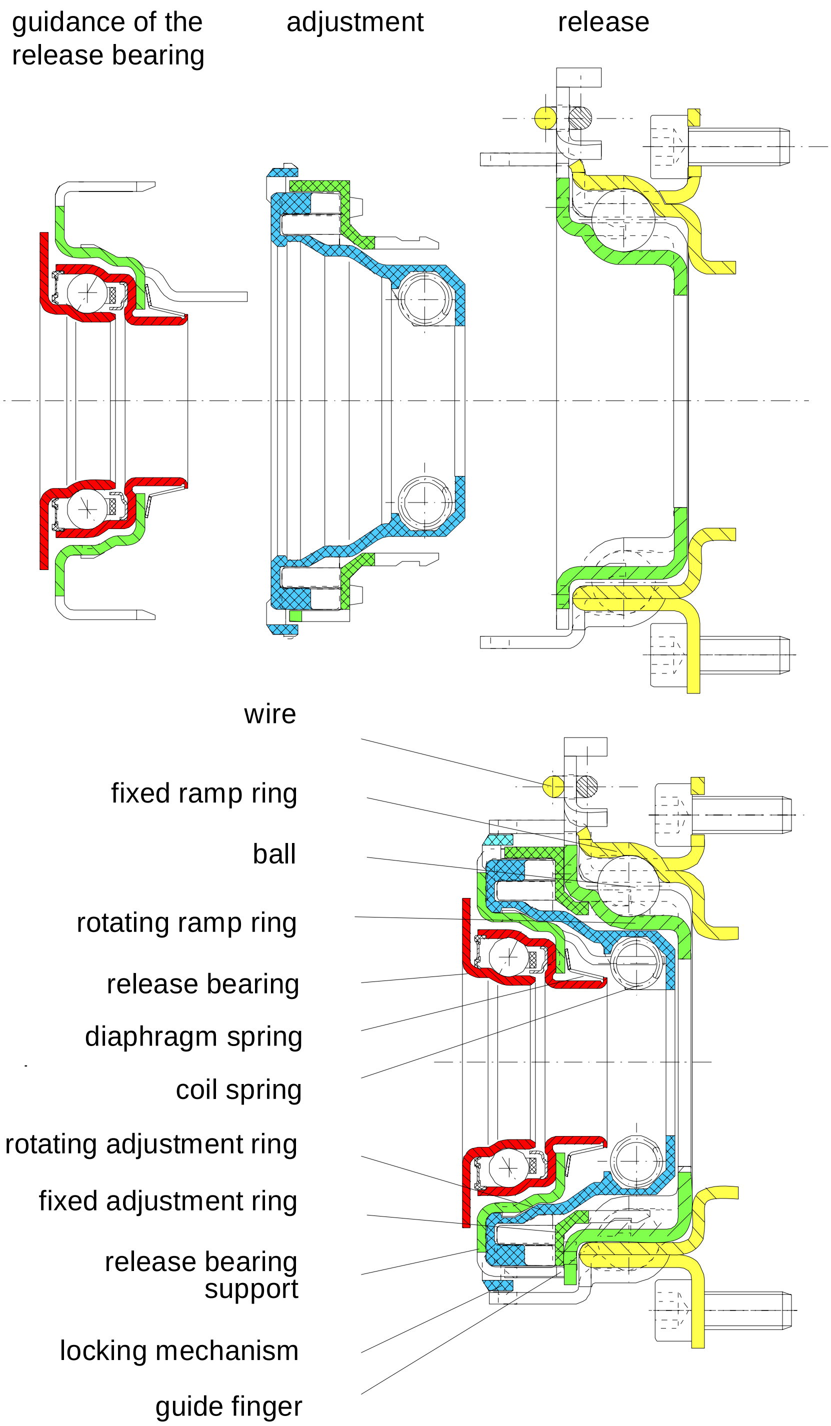


Figure 11: Mechanical central release system (MCR)

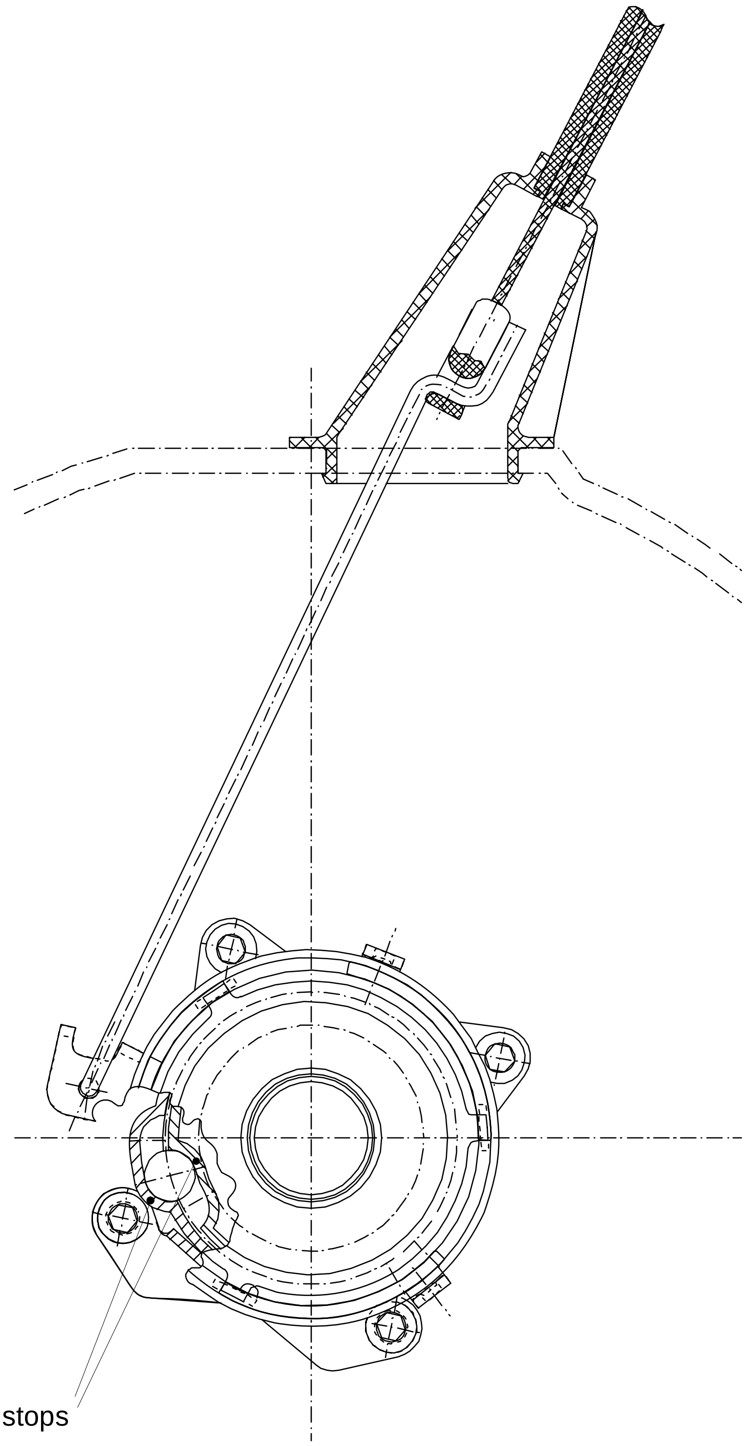


Figure 12: MCR cable connection

A ramp angle varying along the cable travel can be used to change the lever ratio over the release travel and, if necessary, compensate for the release load, which in the SAC typically drops considerably at the end of the release travel (Figure 13).

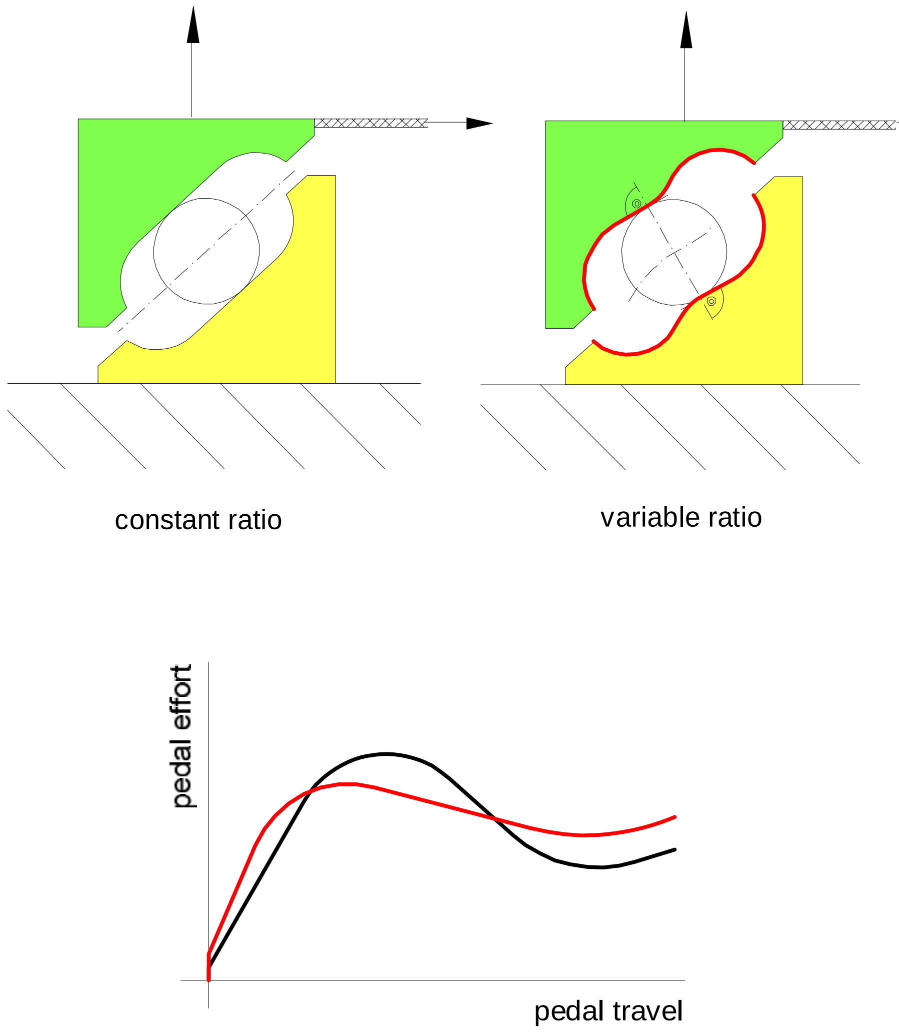


Figure 13: Ramp concept

The fixed adjustment ring is permanently connected with the rotating ramp ring (Figure 11), which together with the rotating adjustment ring provides the adjustment function via two concentrically situated ramp areas. The outer and inner ramp areas consist of two ramps, each extending over 180° with a self-locking ramp angle. After readjustment, each ramp area provides two contact areas separated from each other by 180°. Because the outer and the inner ramp area are off-set by 90°, a stable four-point support is established.

A coil spring assists readjustment by exerting a force directly onto the rotating adjustment ring and via a flange onto the release bearing support. Depending on the available space, the spring can be placed inside, as shown here, or radially outside of the adjustment ramps. With a readjustment angle of approximately 100°, about 6 mm adjustment can be achieved.

The coil spring also produces a release bearing preload.

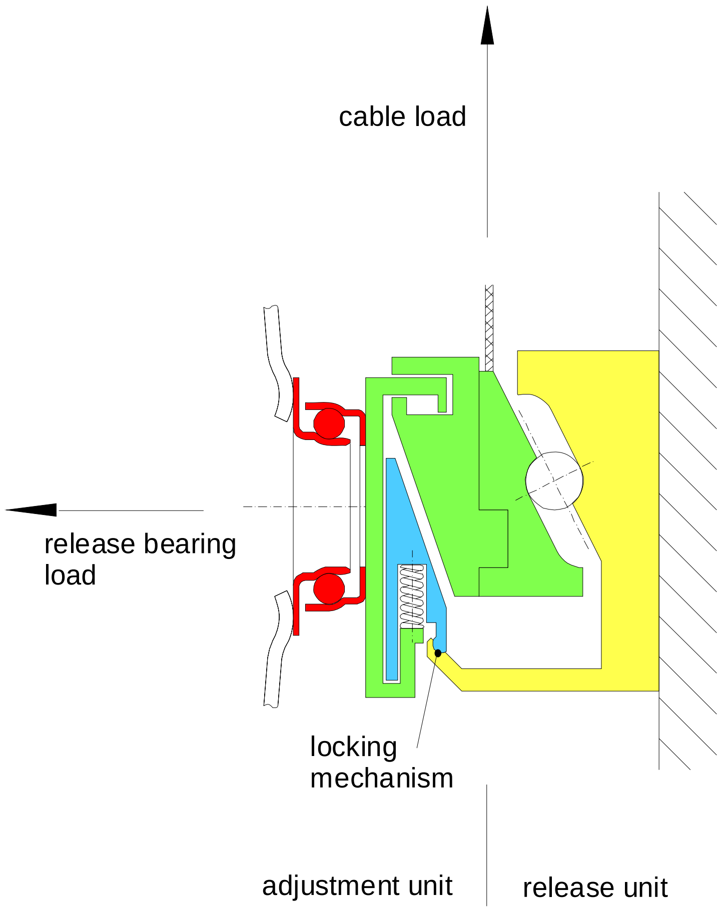


Figure 14: MCR block diagram

Prior to installing the MCR, the adjustment mechanism must be reset and locked (concept see Figure 14), which requires that the MCR be removed from the vehicle. The adjustment ring is turned back and then locked in the rotating ramp ring in order to block the axial movement of the rotating adjustment ring. The MCR assumes its shortest axial length and can be installed in this position. After the clutch housing is installed, the mechanism unlocks automatically during the first clutch operation by separating the rotating ramp ring from the rotating adjustment ring, which releases the locked parts and frees the adjustment function.

Once activated, the ramp-ring reset mechanism functions to adjust for the installation tolerances. It moves the ramp ring away from the lock-up position and effectively prevents further lock-up. The mechanism can easily be locked again manually if the transmission housing is removed from the vehicle for service. In addition, the mechanism locks the release system during transport to the automotive plant.

The self-centering release bearing is mounted in the traditional fashion on the release bearing support via a small diaphragm spring. Because the three-point support on the three-ball ramp prevents the release bearing from tilting, the conventional quill can be eliminated. This means that the outer bearing diameter can be smaller, which reduces costs and bearing drag torque. The three guide fingers on the release bearing support directly transfer these drag torques onto the fixed adjustment ring, which in turn isolates the adjustment function of the rotating ring from the drag torque.

Tolerance compensation

The axial length of the component is based on the crankshaft length, the flywheel height, the diaphragm spring height, the release system length, and distance to the transmission wall. The finger height adjustment assembly can also be used to compensate from tolerance of this axial length.

This reset mechanism cannot compensate for the tolerances of the cable length and its linkage, which produce a variation in pedal height. A separate initial adjustment mechanism is required, like the one that has been developed in cooperation with Küster (Figure 15).

Basically the design consists of two nested sleeves which are preloaded with a light spring and integrated into the cable sheath.

At installation the spring is pushed to solid. Serrations in a sliding sleeve form a positive locking function, which prevents the sleeves from being pulled apart. At minimum length, the sheath is shortened by more than 40 mm and permits easy connection of the protruding cable. After the transmission has been installed, the safety catch between the sleeves is released by activating the locking sleeve. Inside the sheath a spring-supported length adjustment is established. After the locking sleeve has been released, the serrations again lock with each other and are in operating condition. This process can also be easily accomplished during service.

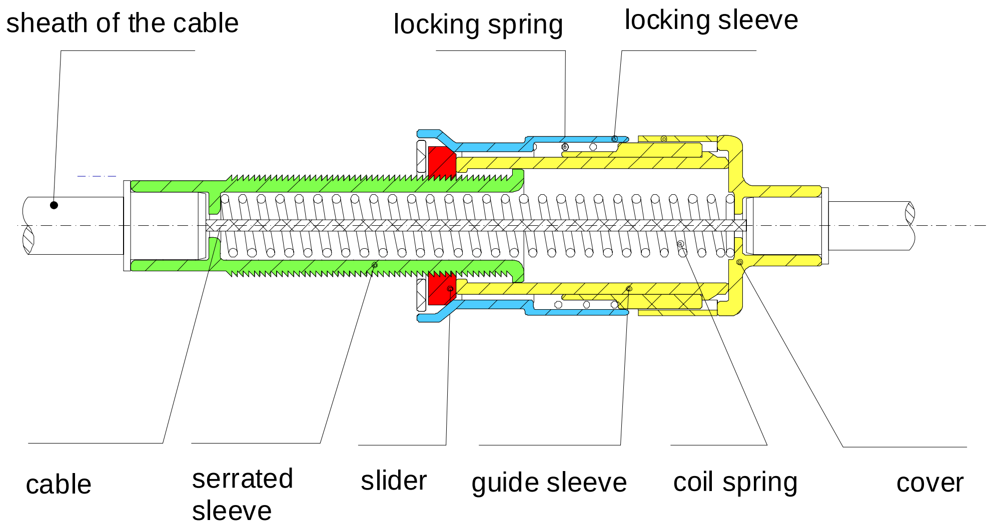


Figure 15: Initial cable adjustment mechanism, Küster design

This mechanism must be sealed against the somewhat lower air pressure inside the vehicle. Otherwise it will attract dirt particles, which in turn increases the friction inside of the cable.

Counter to popular belief, wear on cables causes less than one millimeter change in length, therefore the cable itself does not have to be readjusted during its lifetime, although this could be easily done using the initial adjustment mechanism.

It also is possible to adjust the pedal height tolerance with a locking mechanism placed between cable and pedal (Figure 16, Kirchhoff design).

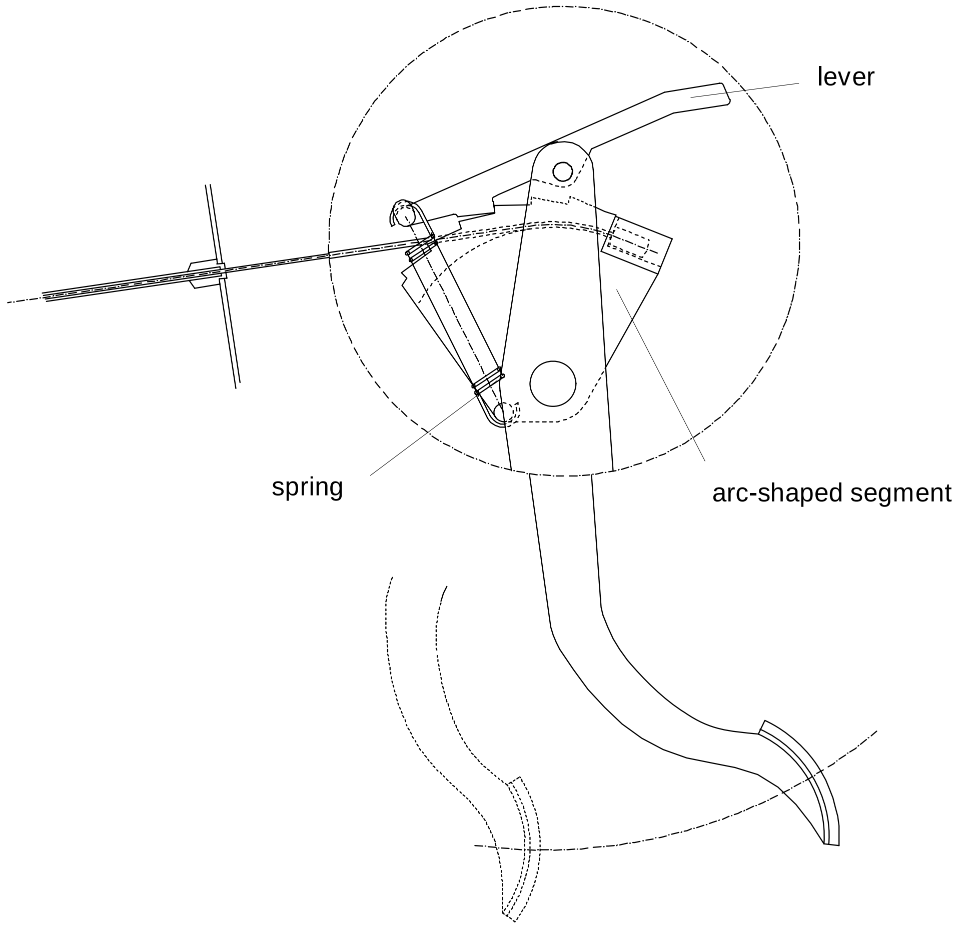


Figure 16: Initial pedal adjustment, Kirchhoff design

During installation, the cable is hooked into an arc-shaped segment. Interlocking teeth in the lever and the segment prevent rotation.

After the cable is attached, the lever is depressed to release the serrated teeth, and the attachment segment preloads the cable slightly via a spring. When the lever is released, the spring-loaded positive connection is reestablished and the original adjustment is completed. Because this mechanism can be installed inside the vehicle, it is not exposed to higher temperatures, as it would in the engine area. There are also no air pressure differences, which in turn reduces contamination of the cable by dirt particles.

There are various concepts for initial adjustment and easy attachment of the cable, as demonstrated with both previous examples. The vehicle type does influence the selection. In any design, the smallest possible distance between serrations should be used in order to limit lash, which would translate into release system losses.

A sample MCR device was installed in a LuK-vehicle, where it functioned to full satisfaction (Figure 17).

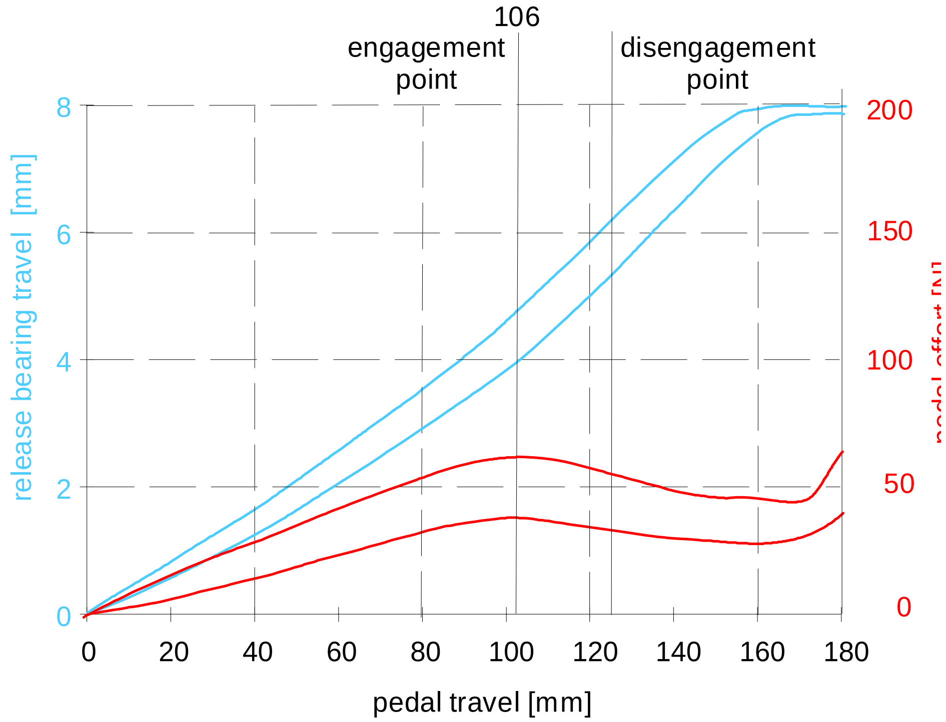


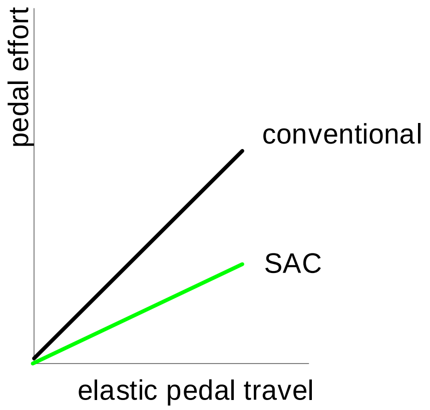
Figure 17: MCR function test in the vehicle

Isolation of pedal vibrations

Pedal vibrations attract unfavorable attention by creating noises or a tingling sensation in the foot. Elimination of the release lever and installation of the SAC with MCR improves vibration isolation (Figure 18).

Eliminating the release lever gets rid of a relatively heavy part that has its own natural frequency. In addition, the lash-free three-point support prevents tilting, so the finger run-out of the clutch no longer causes pedal vibrations.

Remaining vibration sources include axial crankshaft vibrations and the resulting natural frequencies of the pressure plate.



- The release fork is eliminated from the vibration system
- Run-out of the diaphragm spring finger does not result in excitation because of release bearing guidance without free play
- Softer spring improves the isolation of vibrations

Figure 18: Influence of MCR with SAC on pedal vibrations

In conventional release systems, a rubber spring is often located between the release bearing and the pedal to reduce such vibrations. While the rubber spring makes it possible to achieve vibration isolation above the resonance frequency of the release system, the efficiency of the release system is reduced because the application of springs automatically leads to more flexible and therefore less efficient release systems. Application of the SAC, with its resulting 50 % reduction in release load, provides lower spring rates with increased vibration isolation at the same efficiency.

Figure 19 shows one possibility for integrating such a spring elegantly in the release system. No additional parts are required.

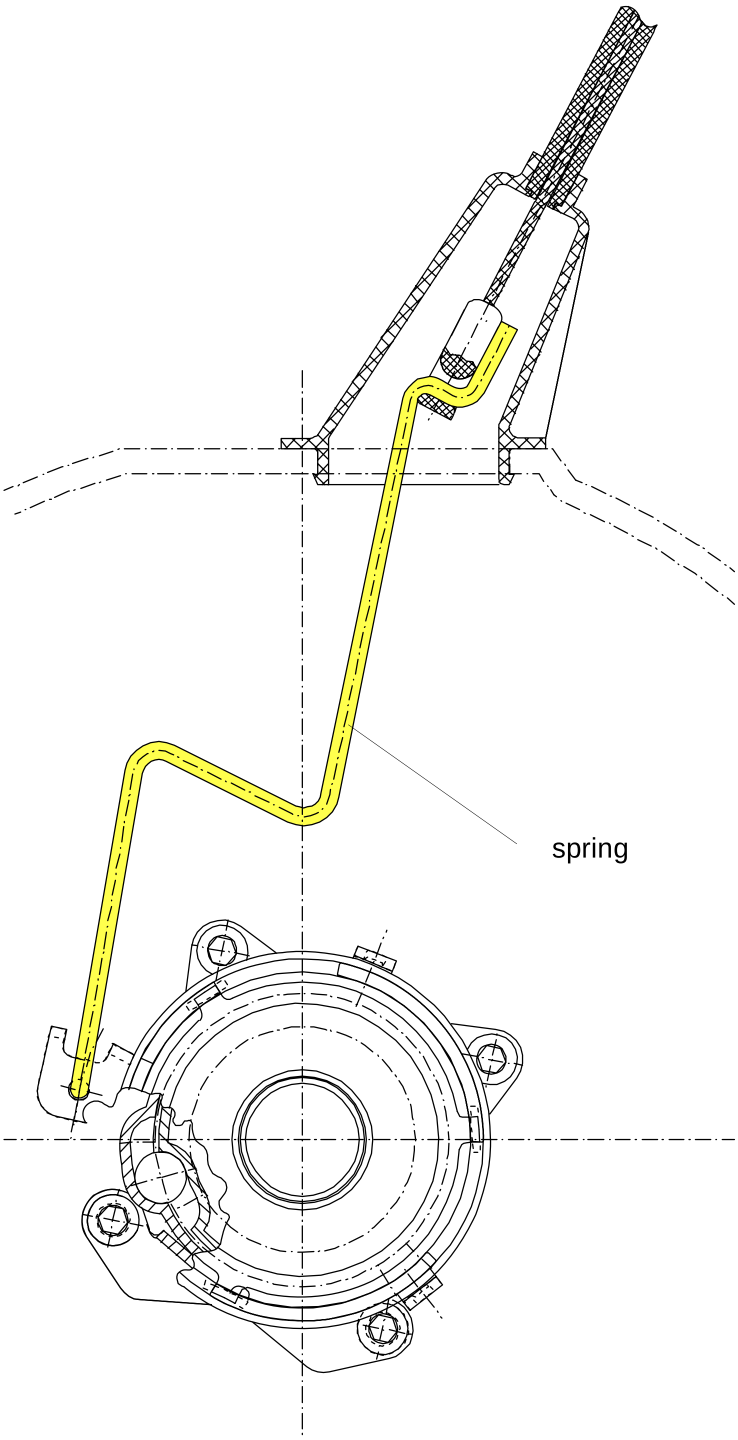


Figure 19: MCR with spring

Comparison of the mechanical central release system (MCR) with hydraulic systems. When comparing these very different systems (Figure 20), it is apparent that the functions achieved are almost the same. The MCR is prone to travel losses (without load) due to the mechanical lock feature in the initial adjustment, but so are hydraulic release systems due to the travel until the orifice connecting it to the reservoir closes

criteria	MCR	hydraulic systems
application	with SAC only	universally
travel losses	8 % (10 mm) (initial adjustment)	8 % (10 mm) (orifice)
losses in elasticity	10 %	5 - 10 %
friction, new friction, old	20 % 30 %	30 % 30 %
damping of pedal vibrations	possible	possible
complexity of design	few parts	few parts
cable routing	limited because of design requirements	free
functional reliability	simple mechanical system	seals represent a risk
costs	- 20 %	base

Figure 20: Comparison of MCR with hydraulic systems

Losses in elasticity are somewhat higher for the MCR, which does not matter in actual application. In new hydraulic systems, the friction produced by the seals is higher than in the MCR. With the cable, however, friction can increase over the lifetime of the clutch, but it will be less than in conventional clutches because of the lower SAC release load.

One advantage of the CSC is the easy routing of the hydraulic lines. On the other hand, the piston seals are subject to higher risk because abrasive wear from dirt particles can endanger the seals and their function.

The difference in costs is significant. In comparison to the conventional hydraulic release system, the release fork, the bearing support for the release fork and the quill located on the transmission can be eliminated.

For obvious reasons the costs for the MCR will lie considerably below those for the conventional hydraulic release system and probably below the costs of a CSC as well.

State of development and outlook

Durability tests have been performed with functional samples for 10^6 release operations (Figure 21). They provided information about suitable material combinations for the ball/ramp ring. C15 has high forming capabilities and in the case-hardened state achieves a suitable surface hardness and hardness penetration. When greased balls were used in tests, no significant wear was found on the ramps. Since the balls do not rotate, a costly grease seal was not required.

Test stand experiments

Durability tests to 10^6 load cycles at 120°C to 165°C

- No wear between ramps and balls
- Wear of cable causes less than 1 mm change in length

Vehicle tests

- Vehicle test reached 20 000 km - no problems
- Friction losses of 25 %

Figure 21: MCR - test stand experiments and vehicle tests

In the meantime, 20,000 km have accumulated in one vehicle test. The SAC reduced the maximum pedal load to about 70 N. Friction is still a bit high at 25 %, because the cable has not yet been optimized.

To summarize (Figure 22), it has been determined that application of the SAC has created a new range of requirements for the release system.

A new type of mechanical central release system has been introduced, which functions similar to the CSC. The elimination of the release fork and the quill clearly cuts costs when compared to conventional hydraulic systems.

- The SAC enables the development of new release systems.
- The new style of mechanical central release system functions as well as its hydraulic counterpart.
- The mechanical principle provides for simple, reliable function.
- A clear cost saving due to elimination of the release fork and the quill as compared to the conventional hydraulic system is achieved.

Figure 22: Summary MCR

For this new product, the first prototypes are planned for the end of 1995.

The Automated Clutch - The New LuK ECM

Dipl. Ing. **Burkhard Kremmling**
Dr. Techn. **Robert Fischer**

Introduction

Manual transmissions have the advantage over automatic transmissions in that the driver has free choice of gears and does not feel dictated to by an automaton.

Already in the 60s, automotive manufacturers began to offer automated clutch operating systems designed to simplify vehicle operation. In the past, interest in these systems has been very limited.

These early systems were functionally inadequate, maintenance-intensive and prone to frequent repairs; disadvantages that could be eliminated with modern vehicle electronics.

In the meantime, automated clutch operating systems have been used in formula 1 and rally vehicles, which proves that they are equal to the most demanding conditions.

Traffic density is constantly increasing, and currently has reached a level where automated clutch systems become interesting.

The data illustrated in Figure 1 are taken from the Allgemeine Deutsche Automobil-Club (ADAC) study [1] in which all significant traffic jams on the German Autobahn system were recorded during the summer vacation travel period.

Figure 1 shows the number of traffic jams that were longer than 20 km. This number increased by 20 % in only a year, i.e. from 75 traffic jams in 1992 to 90 traffic jams in 1993.

The ECM system relieves drivers of having to concentrate on operating the vehicle and thus allows them to turn their attention to the actual traffic situation. This leads to the conclusion that ECM will decrease accident frequency.

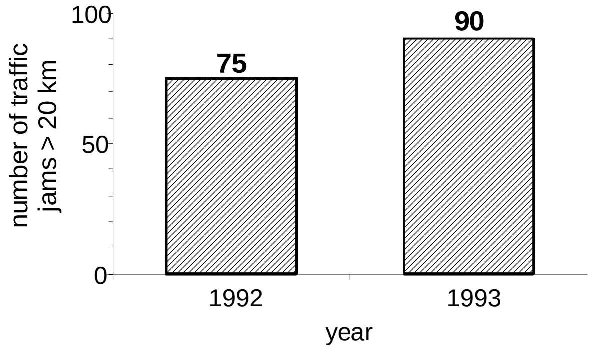


Figure 1: Number of traffic jams longer than 20 km on the German Autobahn system during the summer vacation season (June - September); comparison for the years 1992 and 1993 [1].

When LuK's ECM was presented initially at the last colloquium in 1990 [2], the potential and technical possibilities of the system were demonstrated.

LuK initially developed the ECM in conjunction with hydraulic actuation systems.

In 1993, LuK succesfully introduced the ECM into production with the BMW ALPINA B12 (Figure 2). So far more than 60 % of all ALPINA B12 vehicles have been ordered with the ECM - the so-called SHIFT-TRONIC option - and the number is rising. This is proof that customers want the system.

Atlas Fahrzeugtechnik in Werdohl (AFT) has taken over the development and production of the control device for the ALPINA ECM system. AFT has proven its competence as a development partner and supplier.

Although the hydraulic actuation system has the advantage of dynamic response, it has the disadvantage of being very complex. Consequently, the cost reduction potential of hydraulic actuation systems is less than for other kinds of operating systems.

In order to expand the market base for electronic clutch management, the costs of the system have to be reduced. Consequently LuK has conducted a cost-benefits analysis.

Figure 2: ALPINA B12 SHIFT-TRONIC

Functions of the ECM

Although the advantages offered by the ECM system should be familiar, here is a summary of its functions .

Increased comfort in stop-and-go traffic

The system simplifies stop-and-go driving because it is no longer possible to kill the engine when the vehicle starts off or stops.

Improved maneuverability

LuK has developed a strategy similar to the operation of an automatic transmission, whereby, the vehicle can creep forward when in gear even **if the driver is not pushing on the gas pedal**. The great advantage of this "creep strategy" is that it is easier for drivers to inch forward because they only have to operate one pedal - the brake pedal.

The control system completely dissipates the creep torque with a slight time delay when the foot brake or parking brake is activated. This feature eliminates the disadvantage of increased clutch wear and fuel consumption due to creeping.

Rattle and boom prevention

Controlled clutch slip can be used to eliminate irritating noises such as gear rattle and body boom.

Improved tip-in/back-out performance

Tip-in/back-out performance can be improved using a special clutch control to eliminate surging (chuckle).

Potential advantages for transmission developers

The ECM system provides a significant cost reduction potential for transmission developers:

In conventional manual transmissions with pedal-activated clutches, it is possible for drivers to misuse the clutch by changing gears with the clutch partially closed. The transmission developer has to account for a certain number of improper shift operations when designing the transmission.

The ECM system ensures that the driver can't forget to disengage the clutch when he changes gears. This means that the synchronizer does not have to be so robust, which results in cost savings and reduces shift effort.

When using a pedal-operated clutch, it is also possible that the driver's foot can slip from the clutch pedal during start-up. This kind of "jack-rabbit start" causes short-term torque peaks throughout the entire power train that could be many times the maximum engine torque.

With the ECM, this kind of "jack-rabbit start load" does not occur. This, coupled with a special tip-in/back-out strategy, reduces torque peaks in the drive train. As a result, cost can be reduced because the transmission and axles do not need to be "over-designed".

The potential for reducing fuel consumption and emissions

The ECM system can be equipped with options that will significantly reduce both fuel consumption and harmful emissions (see the chapter Outlook of this paper).

What are the limits on system costs?

LuK assumes that the electronic clutch management system has not been widely used in the past because the costs for the system were almost as high as the costs for an automatic transmission.

In order for the ECM to establish itself, system **target costs** must be **significantly reduced** to the point where they are more comparable to the costs for a manual transmission (Figure 3).

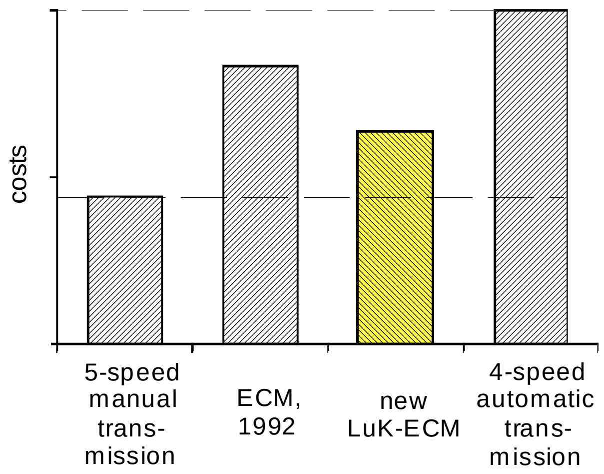


Figure 3: Cost estimate: manual transmission / ECM / automatic transmission

Result: The new LuK ECM

Cost effective hardware was the primary requirement in achieving a drastic cost reduction. This process involved limiting the **performance** of the **clutch actuator** to the level absolutely required of the system, coupled with a **reduction in the number of sensors** needed.

As a result, the requirements imposed on the control strategy increased considerably, which means that instead of high performance or numerous sensors, more intelligent control is needed.

One important development goal of the new LuK ECM has been to use the existing **production transmission without modifications**. This simplifies logistics for the automotive manufacturer, minimizes development effort and decreases investments.

Additional development goals included:

- using the same gear shift lever
- keeping weight low
- maintaining a compact, variable package size

Figure 4 shows the results in the form of an overview of the new LuK ECM.

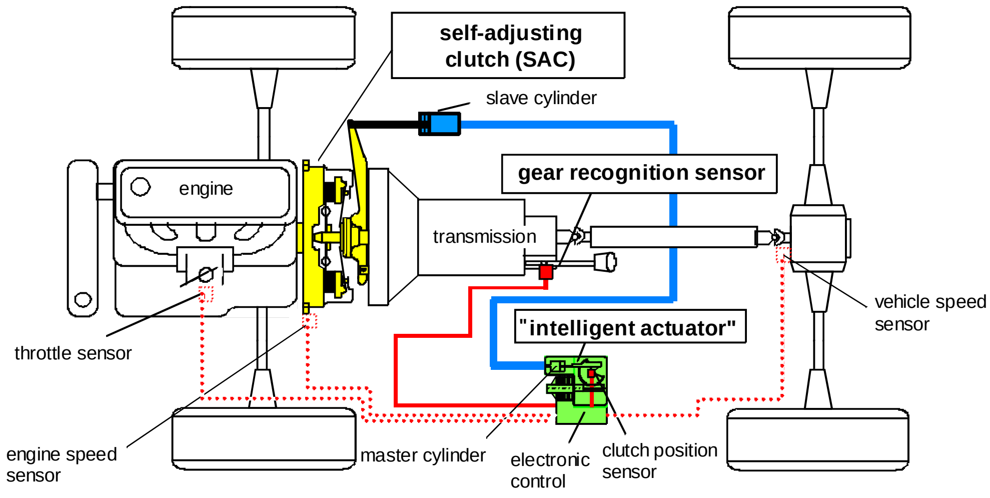


Figure 4: System overview of the new LuK ECM

The LuK Self-Adjusting Clutch (SAC)

The breakthrough to a compact and cost-effective actuator was the self adjusting clutch (SAC), developed by LuK. This design, which was introduced in one of the previous papers, is in production with a pedal-operated clutch at the time of this symposium.

There are limits to how low the actuation load required by the pedal-operated SAC clutch can be set because too low of a pedal load is subjectively perceived as unacceptable.

The requirement of a minimum pedal load also limits the minimum permissible clutch actuation load. The actuator-operated SAC clutch associated with the ECM system also requires a certain minimum actuation load in order to overcome friction. The minimum actuation load of the SAC clutch for the ECM can, however, be significantly lower than that for the pedal-operated SAC clutch.

Figure 5 shows the actuation load as a function of travel. The load curves for a conventional clutch are plotted in red; the solid curves show the new condition and the broken line curves represent the wear condition. In comparison to the conventional clutch, the yellow lines show the load curve for a SAC clutch designed for the ECM system.

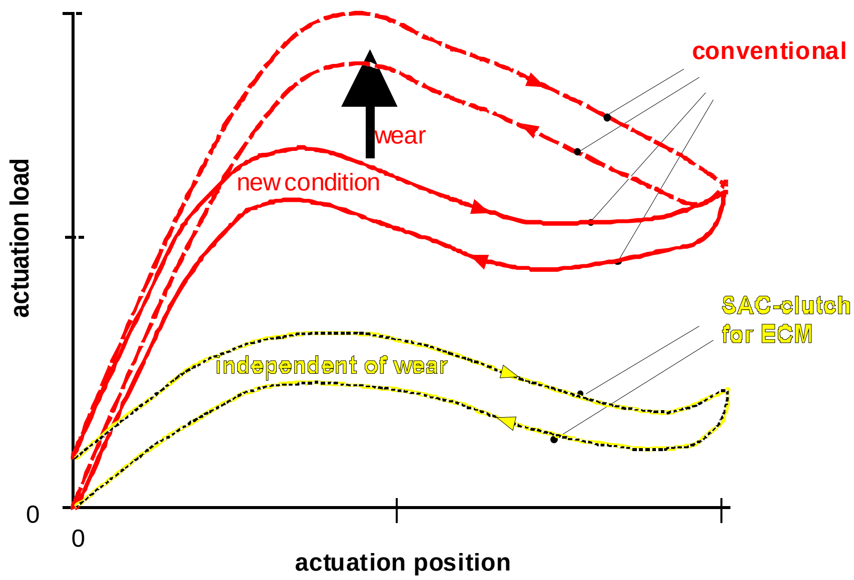


Figure 5: Comparison of the actuation load curves for a conventional clutch and for the SAC clutch designed to be used with the ECM system

Based on the special conditions described above for the ECM actuator-operated SAC clutch, a **load reduction of about 2/3 compared to the maximum actuation load of a conventional clutch** can be achieved.

In comparison to the conventional clutch, in which the actuation load increases as the facing wears (red curves), the actuation load of the SAC clutch (yellow curve) is **constant over the entire service life** of the component.

An additional advantage of the SAC clutch is the option of increasing the facing wear reserve without increasing the actuation load at the same time.

The clutch actuator:
A mass produced electric motor

By reducing the clutch release load, it is possible to use a low power electric motor (Figure 4). LuK's development partner, BOSCH, which manufactures the small motor used, produces more than 4 million of these units per year.

There is a hydraulic master cylinder located on the actuator housing. The master cylinder is connected with the clutch slave cylinder by hydraulic lines, which allows the actuator to be installed almost anywhere in the vehicle. The only requirement is that its ambient temperature does not exceed 100 °C.

The basic actuator design makes it possible to use a cable instead of master cylinder and slave cylinder.

The dynamic response of the actuator described here is lower than for the hydraulic clutch operation. This apparent disadvantage of slower dynamic response can, however, be compensated for, using the motto "brains instead of brawn", illustrated in Figure 6, by using an intelligent control system.

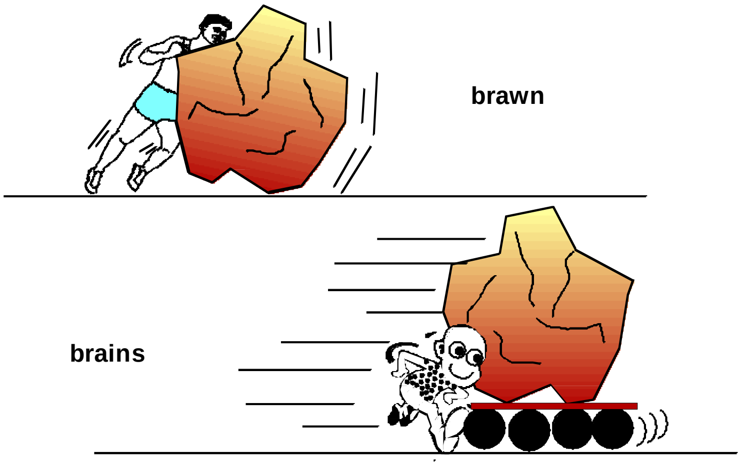


Figure 6: Brains instead of brawn!

Solving several problems at once: "torque follow up system"

As the result of design safety factors and tolerances, which must be accounted for when designing the clutch, the maximum torque that can be transmitted by a clutch, amounts to two to three times the maximum engine torque. Nevertheless, while driving, the average engine torque only amounts to a fraction of the maximum engine torque. If the clutch is "fully closed" (this means, able to transmit the maximum torque), the transmittable clutch torque is many times higher than the actual engine torque.

The basic idea behind "torque follow up system" is: instead of closing the clutch far enough to accommodate the maximum transmittable clutch torque, it is possible to only close it to the point where the transferable clutch torque is only slightly greater than the actual engine torque.

Effect during gear change

As soon as the driver lets up on the gas, the engine torque is reduced and the "torque follow up system" function automatically adjusts the clutch by opening it slightly. This means that when the system recognizes the drivers desire to shift gears, the clutch is already partly open. Consequently, the reduced adjustment speed of the new ECM system is sufficient to completely disengage the clutch, even for fast gear changes.

Effect during tip-in/back-out

Figure 7 shows simulation results for tip-in in 2nd gear.

These graphs compare the following control variants:

- Clutch closed (top pair of graphs)
- controlled slip (middle pair of graphs)
- "torque follow up system" (bottom pair of graphs)

In each set of graphs, the top graph shows the engine and transmission speed curves as a function of time. Each bottom graph shows the longitudinal acceleration.

As shown in Figure 7 above, rapid changes in engine torque are followed by unpleasant surge oscillations, which are sometimes called "chuckle".

The controlled slip system (Figure 7, center), which was introduced at the last LuK Colloquium in 1990, is capable of preventing surge oscillation. The disadvantage of TCI is that it requires a continuous, relatively great amount of slip. This control variant makes it necessary to drive with continuous slip, even in speed and load ranges where slip would not be necessary to eliminate noise.

The effect of "torque follow up system" (Figure 7, bottom) is similar to that of a torque limiter. Slip occurs only when there are rapid changes in engine torque and **only** for a **short time**. In comparison to controlled slip system, "torque follow up system" offers advantages with respect to fuel consumption and wear.

The graphs on the left in Figure 8 show the simulation results already represented in Figure 7. In comparison to these results, the graphs on the right hand side illustrate the measured results for the same driving situation (Tip-in, 2nd gear). It is clear that there is a good correlation between the simulation data (Figure 8, left) and the measured data (Figure 8, right).

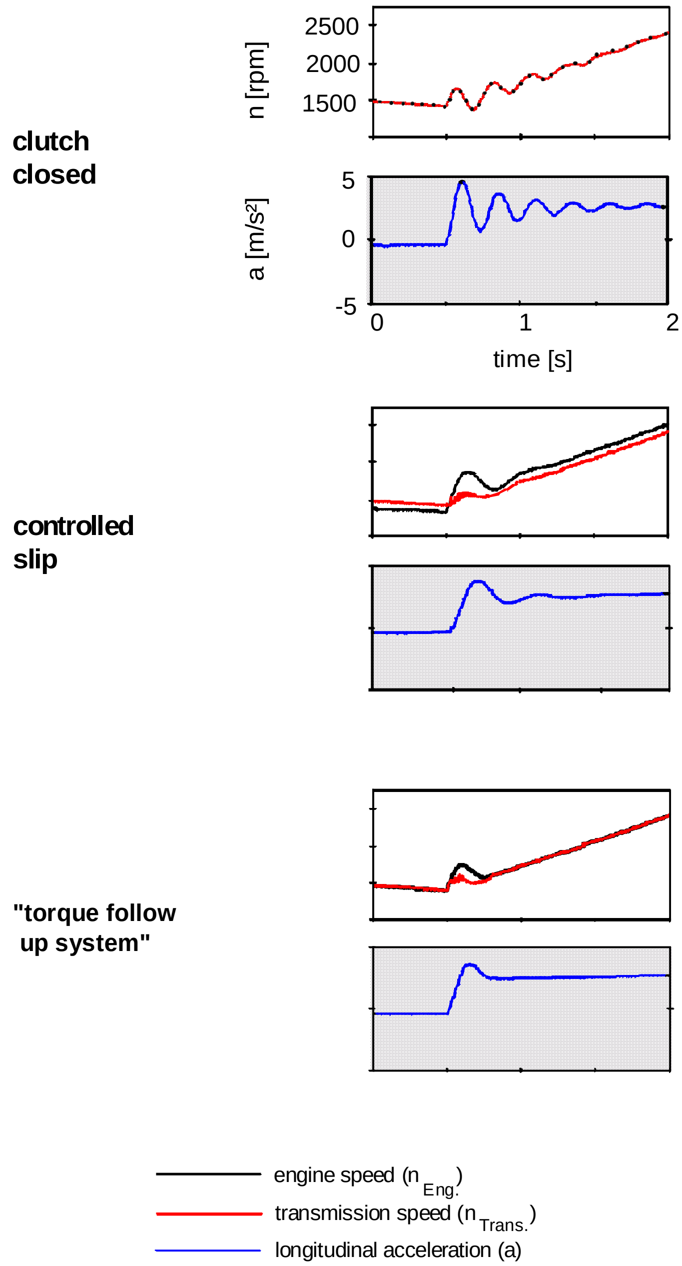


Figure 7: Simulation of a tip-in cycle in 2nd gear. Comparison between the control variants "closed clutch", "controlled slip" and "torque follow up system".

"Torque follow up system" utilizes hysteresis, which means that, in contrast to controlled slip system, the clutch torque is only shifted if the engine torque is changed by a certain minimum value. The result of this design is a significant decrease in the electric motor operating time although the system, nonetheless, provides good tip-in/back-out performance.

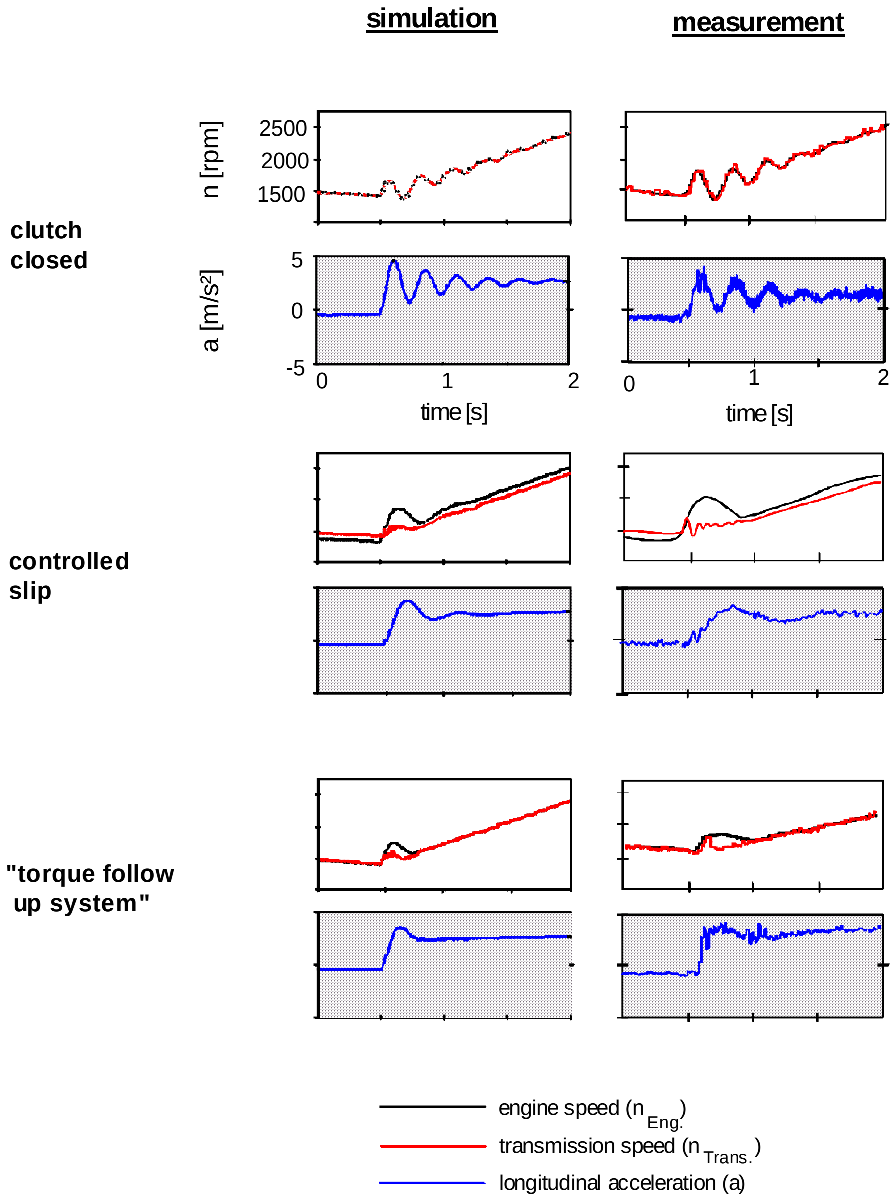


Figure 8: Tip-in in 2nd gear; comparison of simulated and measured data

Integrating the control device in the actuator housing

Two factors play an important role in achieving the next significant step in the development of the clutch actuating system:

- the introduction of a low power electric motor
- installation of the actuator at a remote site (i.e., removed from the release fork or the concentric slave cylinder) in an area whose ambient temperature does not exceed 100 °C.

As a result of these factors, the actuator and the complete ECM control device with the engine electronics have been combined to form a single unit, which is called the intelligent actuator.

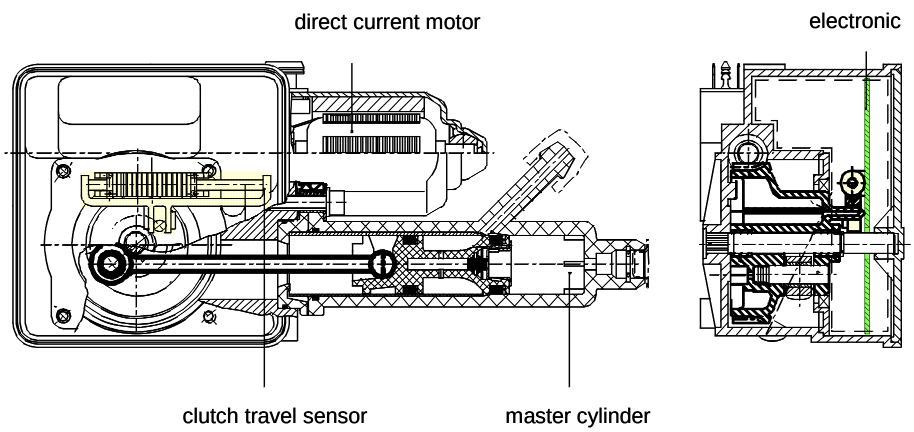


Figure 9: Unit consisting of an actuator and a control device (intelligent actuator)

The most important advantages of this integration include:

- reduced cable complexity and expense
- fewer electrical connections
- fewer components
- increased protection against system malfunction
- reduced system costs
- reduced additional weight.

The complete actuator/control unit was developed by BOSCH.

No transmission modifications required!

In order to **avoid modifying the production transmission**, the following demanding problems had to be solved:

- **moving the clutch travel measurement** from the slave cylinder to the "intelligent actuator" (Figure 4), which means placing it upstream of the master cylinder
- **eliminating the transmission input speed sensor**
- **moving the gear recognition sensor** to the shift rod from its previous location directly on the transmission.

These measures have simplified the adaptation of the ECM to a new vehicle, but the software requirements increase at the same time. Moving the clutch position measurement from the slave cylinder to the intelligent actuator will be used as an example to explain this situation:

The master cylinder attached to the actuator, i.e. the "new" measuring position for the clutch position, and the slave cylinder are connected by a hydraulic line that varies in length depending on the vehicle in question. The fluid used in the system (brake fluid) is subject to changes in volume due to temperature influences, which results in significantly inaccurate clutch position measurement signals. Losses due to compressibility, which are for the most part dependent on the air entrained in the brake fluid, also cause false measuring results as well as fluid losses due to the compensation orifice in the master cylinder.

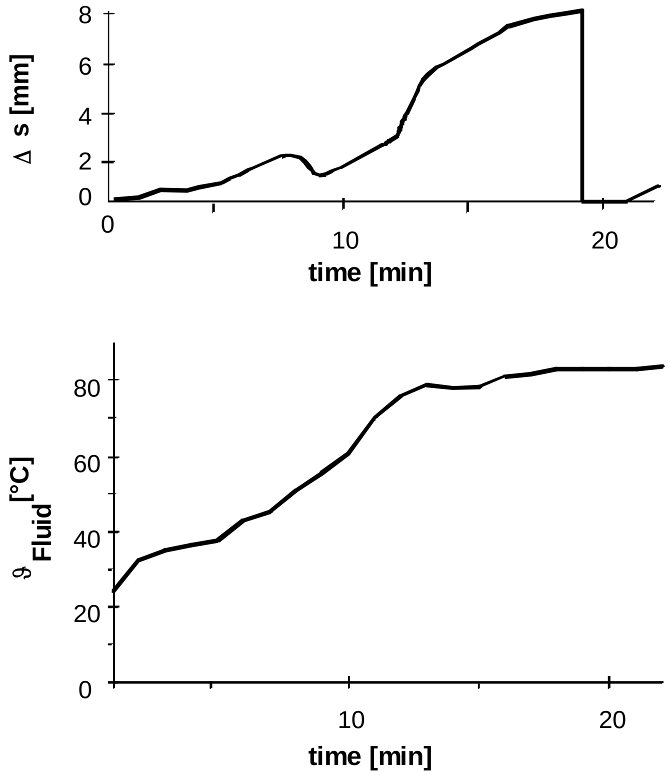
Figure 10 shows the measured data curves for the ECM vehicle. The measurements were stored with the vehicle parked immediately after a brisk drive.

The figure illustrates the false signal values for the clutch position, which are attributible to moving the sensor. The top graph shows the deviation Δ s between the true clutch position measured on the slave cylinder and the virtual clutch position measured on the master cylinder and plotted as a function of time.

At the point in time t=0, the vehicle was parked and the measurement was started with the engine running. The relatively low fluid temperature of about 25 °C at the beginning of the measurement cycle (shown in the bottom graph in Figure 10), is attributable to cooling of the line between the master and slave cylinder as a result of air flow while driving.

However, heat built up from the engine and exhaust system cause the fluid temperature to rise significantly again (Figure 10, bottom). As a result, the brake fluid expands in volume, which causes the deviation Δs between the master and the slave cylinder to increase.

As shown in the measurement, the position deviation Δs amounts to about 6 mm at time $t \approx$ 18 min. If one takes into account that the entire adjustment range of the clutch only amounts to about 20 mm, it is obvious what a great effect this has on the measurement.



Δ S = Deviation between the position of the slave and the master cylinder

$\bar{\vartheta}_{Fluid}$ = Average fluid temperature in the line between the master and the slave cylinder

Figure 10: Falsification of measuring signals for clutch position as a result of shifting the measuring point from the slave to the master cylinder.

At t $\approx$ 19 min, the engine is shut off and the clutch is closed. The excess brake fluid escapes via the compensation orifice in the master cylinder into the reservoir. When t $\approx$ 21 min, the clutch is opened again. Directly thereafter, the position deviation Δs is initially 0, but then it increases again immediately.

LuK has developed a strategy whereby the **clutch characteristic**, that is the relationship between the clutch position and the transmitted torque is **constantly** adapted. This involves determining the actual clutch torque during clutch slip phases based on dynamic torque equilibrium. This adaptation process has made it **possible** to **fully compensate** for the significant deterioration in signal quality described above. This strategy has been implemented in the ALPINA B 12 production design, where it functions without any problems.

Additional sensors required: Reduced to only one

The LuK ECM only requires one additional sensor for gear recognition. All other sensor signals required by the control system can be picked off the existing control devices (Figure 4). As cited above, gear recognition is now measured on the shift rod rather than on the transmission.

The advantages of reducing the number of ECM-specific sensors include:

- fewer electrical connections, which reduces the sources of potential problems
- simplified installation and easier wiring

Two analog potentiometers are used as the sensor elements in the gear selection recognition system. These sensors have proven themselves for long-term automative applications as throttle sensors. They indicate two shift lever directions - "shift" and "select". Analog measurement eliminates the need for system adjustments to compensate for production variations and changes in the transmission kinematics between the shifter and the gear recognition sensors, for example, due to temperature-related material expansion and to wear. Compensation for these changes are made continuously during operation based on extreme position checks.

As already mentioned, a series of **additional signals are required** for the ECM system controls, as shown in part by the broken line in Figure 4. These signals can all be picked off from **existing control devices**:

- engine speed
- vehicle speed
- throttle position
- engine torque
- parking brake engagement
- brake engagement

Eliminating shift lever changes

The new LuK system recognizes the drivers desire to shift gears based on the travel of the shift lever.

The sensor designed to recognize the desire to shift gears that was included in earlier ECM systems can be eliminated in the new LuK ECM with the exception of special cases.

An intelligent software program prevents incorrect system responses due to leaving a hand on the shift lever or uneven drive surfaces. The system also opens the clutch at the appropriate time when the driver actually does signal the wish to change gears.

Simple torsion damper design

A holistic view of the mechanical and electronic systems has led to the recognition that the use of a **simple torsion damper** will make it possible to reduce the speed ranges where slip is necessary to eliminate noise. The torsion damper can also be designed only to accommodate partial loads.

Comparing different systems

Figure 11 compares various clutch automation systems. The illustration at the top of the drawing shows a hydraulic actuation system (LuK's 1st generation ECM). The center illustration shows a conventional electric motor actuator **without** the SAC clutch and the bottom illustration shows the new LuK system.

The comparison shows that a hydraulic actuator (top of Figure 11) requires a number of cost-intensive components that contribute to a weight of about 5 kg, such as:

- an electric pump
- a hydraulic valve body
- an accumulator
- a pressure sensor
- a proportional valve.

With conventional electric motor actuation **without** the SAC clutch (Figure 11, center), the 6 kg weight is higher than for the hydraulic actuation system.

If the size of the new LuK system (bottom of Figure 11) is compared with the conventional electric motor system, (Figure 11, center), the simplifications due to the SAC becomes apparent, as illustrated by the following advantages:

- simpler installation and wiring
- fewer electrical connections and consequently, fewer sources for potential problems
- fewer detail parts
- considerable cost reductions
- significantly decreased weight (about 2 kg)
- lower space requirements

The reduction in sensors by eliminating the transmission speed sensor has further simplified the new LuK system.

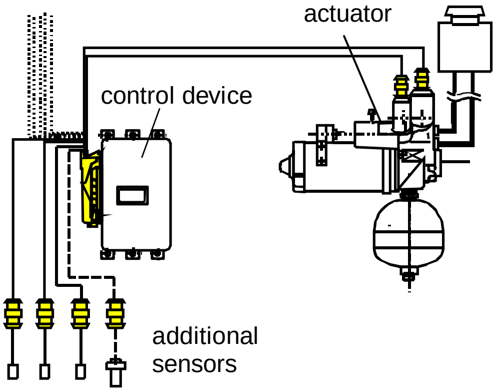
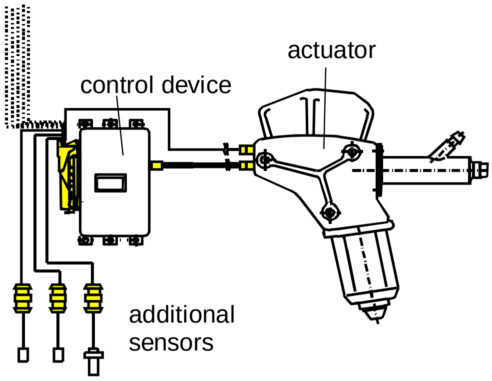
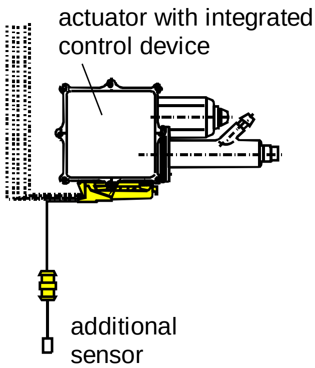
actuation system	design	weight [kg]	sensors
hydraulic system		**≈ 5**	**3 ... 4**
conven-tional electric motor		**≈ 6**	**3**
new LuK system		**≈ 2**	**1**

Figure 11: A comparative view of various systems

Outlook

The ECM system offers several possibilities for new customer options.

Increase in driving stability

By supporting other systems, for example:

- anti-lock brakes
- traction control
- control of engine braking torque

the ECM can improve driving stability.

Integration of anti-theft device

An anti-theft device can be easily integrated into the ECM simply by locking out the starter and maintaining the clutch in the disengaged position.

Decreasing fuel consumption and harmful emissions

The ECM offers several options for decreasing fuel consumption and harmful emissions.

Shift recommendations

A display or a control light on the dashboard indicates to the driver when it is most advantageous to shift up or down in order to save fuel.

Official test cycles do not allow for checking the effects of driving style because they specify when to shift. Consequently, LuK has performed tests on a special test route designed to compare driving styles. These tests compare the following driving conditions:

- typical country driving
- driving in city traffic
- driving on the highway.

The same vehicle (a mid-size passenger car with a 3-liter gasoline engine) was tested with an ECM control system with shift recommendation and with a conventional clutch pedal.

Figure 12 shows how fuel consumption can be reduced with ECM if the driver conscientiously follows the shift recommendations compared to operation without ECM and without shift recommendations. During city driving, fuel consumption with shift recommendation is about 21 % lower, and about 19 % lower for country driving. On the Autobahn, there is no significant advantage because here the vehicle operates primarily in 5th gear.

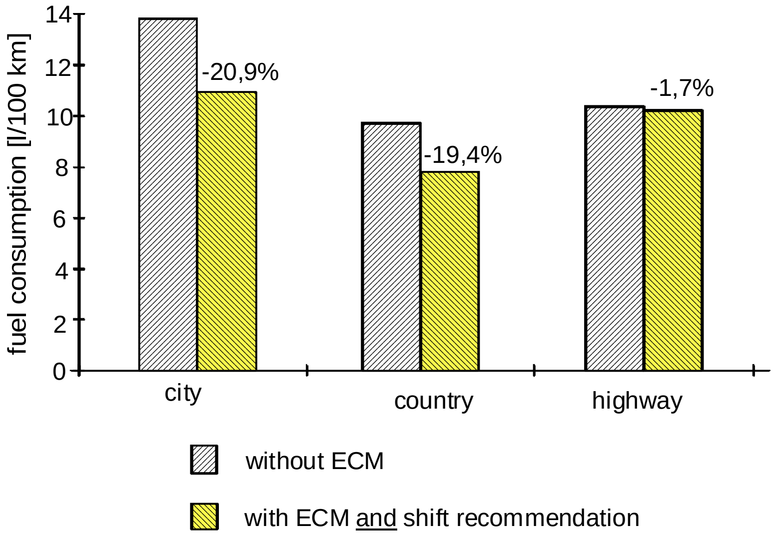


Figure 12: Comparison of average fuel consumption:

- without ECM (normal clutch operation)
- with ECM and shift recommendation

Shift recommendation is more likely to be accepted with ECM than without because changing gears is significantly easier with ECM.

The psychological advantage of ECM with shift recommentation compared to automatic transmission operation is that the system only makes a recommendation, but the driver makes the decision to actually shift gears.

Start/stop function

In addition to reducing fuel consumption and harmful emissions, ECMs offer the so-called start-stop function, which shuts off the engine during longer stop phases. The engine starts again when the driver presses on the gas pedal or shifts into gear.

Free-wheeling function

The free-wheeling function opens the clutch while coasting (when the driver is not pressing down on the gas pedal).

One possible variation affects continued engine operation during idle mode. With a diesel engine with direct fuel injection, which features low fuel consumption during idle mode, it is possible to achieve very significant fuel savings.

Another variation that will achieve even more important fuel savings opens the clutch and turns off the engine during free-wheeling; it restarts the engine as soon as the driver steps on the gas.

Automatic shift transmission

A further ECM design stage can automate shift operation as well as clutch operation.

Summary

The LuK ECM, with all its functions, is significantly less expensive than other systems. The total costs for the ECM are much closer to the costs of a conventional manual transmission than to those of an automatic transmission, which provides the basis to introduce the ECM into a larger market.

The primary **basis** for implementing such drastic cost reductions is the introduction of the LuK **self-adjusting clutch.** The reduction of the actuation load by 2/3 of the actuation load of a conventional clutch has made it possible to use a very compact electric motor.

Using a low power electric motor with very low heat output enables both the **actuator** and the **control unit** to be integrated in a **single housing**. This integration has resulted not only in cost savings, but also in other advantages, for instance, a much simpler system design, fewer detail parts, reduced effort for wiring, and increased system reliability.

The new **LuK ECM** can be applied **without changing the conventional manual transmission** and in most cases it can be installed **without modifying the shift lever.**

As a result of the changes described here, the number of sensors installed in the vehicle can be reduced, leaving **a single sensor** for gear recognition. All other signals can be picked off either in the actuator/control device unit or from existing control devices such as the engine control.

References

[1] ADAC press report of 1993-09-15

[2] 4th International LuK-Colloquium, 1990. "Torsional Vibrations in the Drive Train", "Torque Control Isolation (TCI) - The Smart Clutch"

Torque Converter Clutch Systems

Dr. techn **Robert Fischer**
Dipl.-Ing. **Dieter Otto**

Introduction

Modern vehicle drive-train engineering must exhaust all potential drive-train options in order to provide maximum acceleration and fuel efficiency with high overall efficiency and optimum comfort. At the same time, attention must be paid to ever stricter emission standards. These requirements often work at cross-purposes with each other, which means that improving emissions often entails increasing weight, fuel consumption and decreasing acceleration, not to mention incurring constantly increasing costs [1].

Despite this trend, LuK has developed a torque controlled clutch system - called the TorCon System - that increases driver comfort, reduces fuel consumption and emissions, improves acceleration and even results in a 4-speed automatic transmission that is superior to a conventional 5-speed automatic. This means that wherever an expensive 5-speed automatic transmission is used due to fuel consumption and acceleration requirements, the same results can be achieved with a 4-speed automatic transmission.

LuK's design philosophy is centered on holistic system design, and the automatic transmission area is no exception. This approach meets the demands that automotive industry have come to expect of it's system suppliers.

Given the parameters it is not possible to fall back on large test facilities and a fleet of test vehicles. Yet LuK is confident that it is a competent development partner and can ensure the introduction of new transmission systems into production with the shortest possible development lead times. The following demonstration of LuK's development philosophy shows why this is possible.

LuK, as a component supplier, has given special thought to the total system, not just to the parts supplied by LuK. In concrete terms, this means that when dealing with automotive transmissions, it is also necessary to look at control systems, engines, vehicles and external influences (see Figure 1).

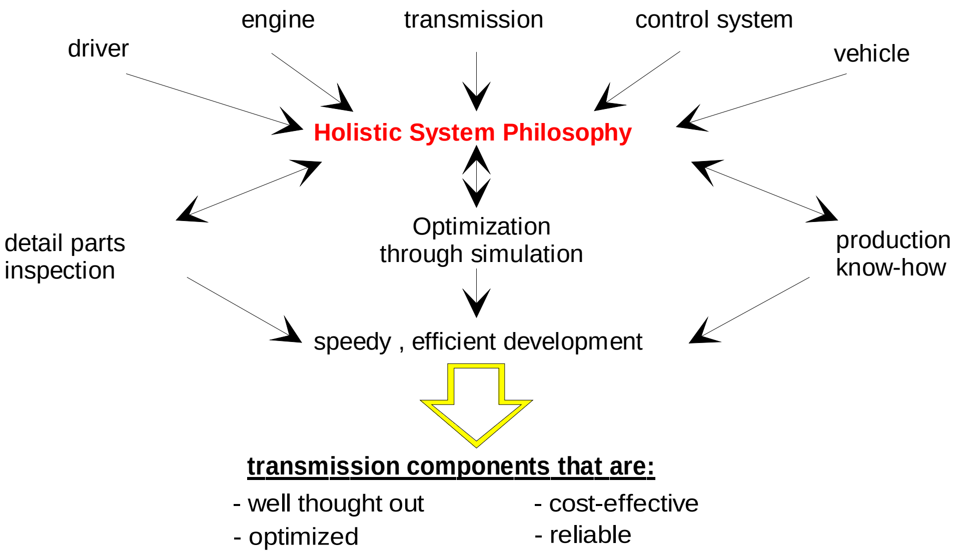


Figure 1: Holistic System Philosophy

LuK greatly appreciates the importance of simulation and detail component testing as important tools for cutting development lead-times and costs. LuK also possesses considerable production know-how (for instance, LuK produces over 2 million conventional torque converter clutches per year for the automatic transmission market).

LuK conducts extensive basic tests in order to ensure reliable product function. Based on these tests, the structure of the development model gradually becomes more complex compared to the previous model [2]. In this way, general knowledge can be integrated and extensive, time-consuming - not to mention expensive - vehicle tests (Figure 2) can be significantly reduced. Nevertheless, some information must be obtained from vehicle tests and vehicle tests serve to confirm projected data.

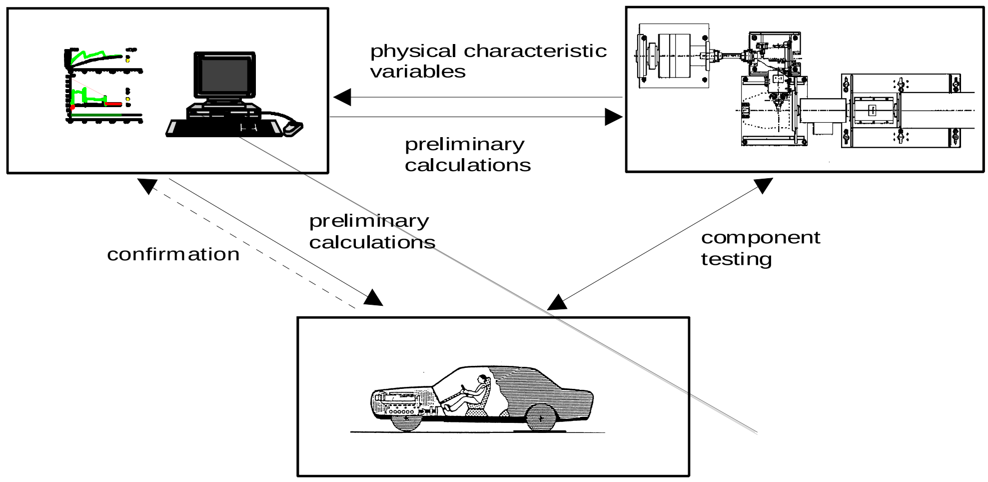


Figure 2: Integrated Development Tools

An example of this kind of development is an analysis of the relationship between friction linings and hydraulic fluid. The torque converter clutch is, of course, a wet clutch and basic knowledge of this kind of clutch is important in developing transmission systems. Typical problems, such as shudder, only occur at relatively high mileage levels. In order to reduce the time it takes to gather data on long-term performance, a small test stand has been developed that reduces the fluid volume used in the test to ¼ liter, a value that corresponds to the relation between the friction surface and the quantity of fluid present in the automatic transmission. Considerably more fluid has been used in traditional test stands, with the result that, if oil additives are damaged at the friction surface, it takes a relatively long time for any consequences to show up because of the dilution effect of the fluid. A small test stand enables fairly rapid results to be obtained concerning the interaction of hydraulic fluid and the friction lining when exposed to specific loads. One can also determine the stress that the fluid and the facings can be exposed to over time. These findings go directly into new designs, for example the slipping torque converter clutch. This results in new torque converter clutch designs that withstand extensive customer durability tests without any problems.

The Physics of the Torque Converter

The torque converter consists of a fluid coupling with an impeller, a turbine and a stator.

Without slip it cannot transmit torque.

Given a constant output speed, the higher the slip speed, the higher the torque. Figure 3 shows this relation for a stalled converter.

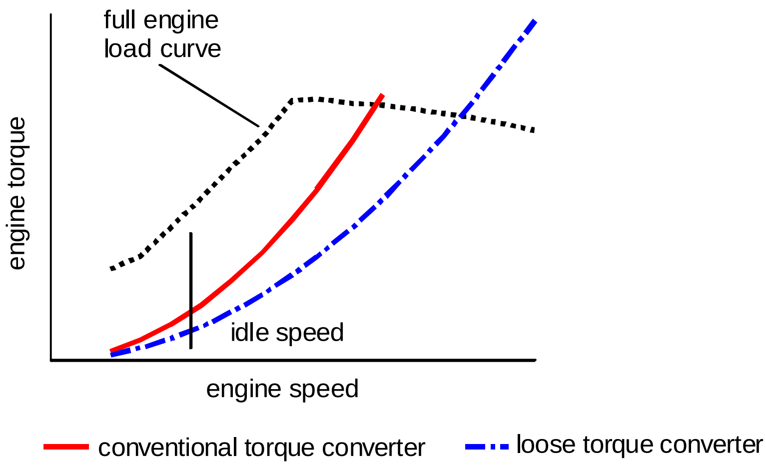


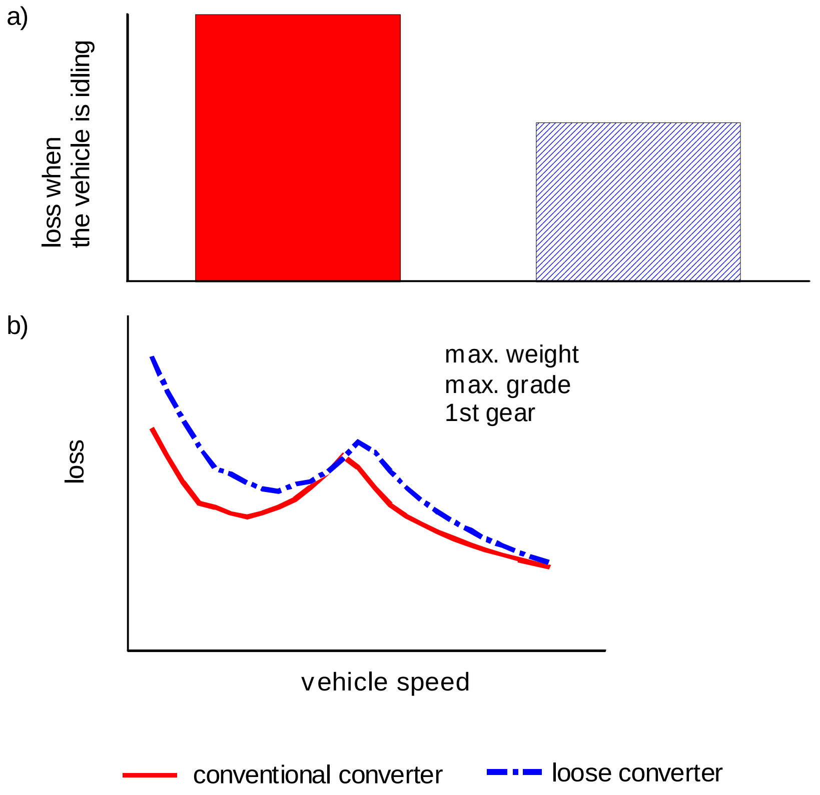
Figure 3: Converter stall torque in relation to the engine torque curve

One says a torque converter is "looser" if, compared to another torque converter, it has a higher slip at the same torque level, which also means that it transmits less torque at the same slip level. A looser torque converter exerts less resistance on the engine. If the driver demands greater torque, the looser converter builds up higher speed differentials.

Higher speed differentials result in an "elastic" connection between engine and transmission, which causes a delay in the vehicle reaction to changes in throttle position. This means that the vehicle is no longer immediately responsive to the throttle.

The advantage of this feature is that most emission tests begin with a cold phase. If it is easier for the engine to reach high speeds in this phase, then it heats up faster and emission levels improve considerably.

At any given engine speed, the looser torque converter exerts less torque in resistance to the engine. If the vehicle engine is idling, it has to overcome converter torque. This means that the energy loss is lower for loose converters in a stationary vehicle than it is for standard designs (Figure 4a).



Figures 4a and 4b: Converter Losses

For any given output torque demand, for example, for a given vehicle speed on a given grade, slip increases with the loose converter, as shown in Figure 3. This means that loss increases as well (see Figure 4b).

In contrast to a clutch, a torque converter can multiply engine torque. This torque conversion can be higher for a loose torque converter than for a conventional design with the same diameter.

A higher torque ratio means that the tractive force increases along with acceleration (Figure 5).

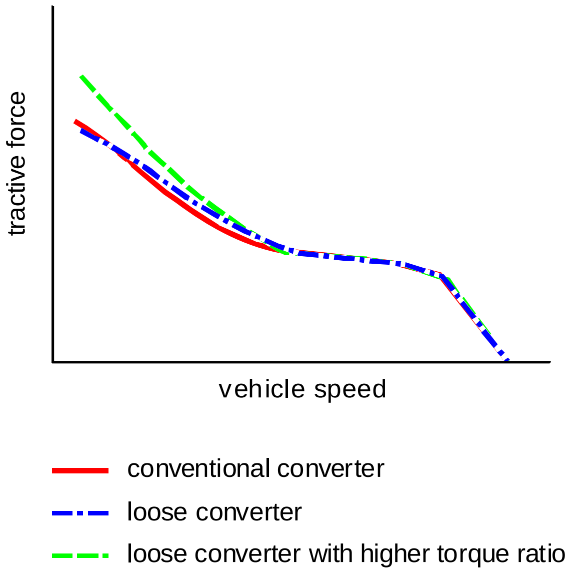


Figure 5: Effect of converter design on tractive force

Losses for a given driving condition (weight, grade) also decrease, if the torque ratio is higher in comparison to those of a loose converter without a higher torque ratio. Nevertheless, they remain higher than with a conventional converter (Figure 6).

All converters produce large amounts of slip at low speed and under extreme load. When slip values are high, load losses are no greater for a loose converter than for a conventional converter because the higher torque ratio improves the efficiency (See Area A in Figure 6).

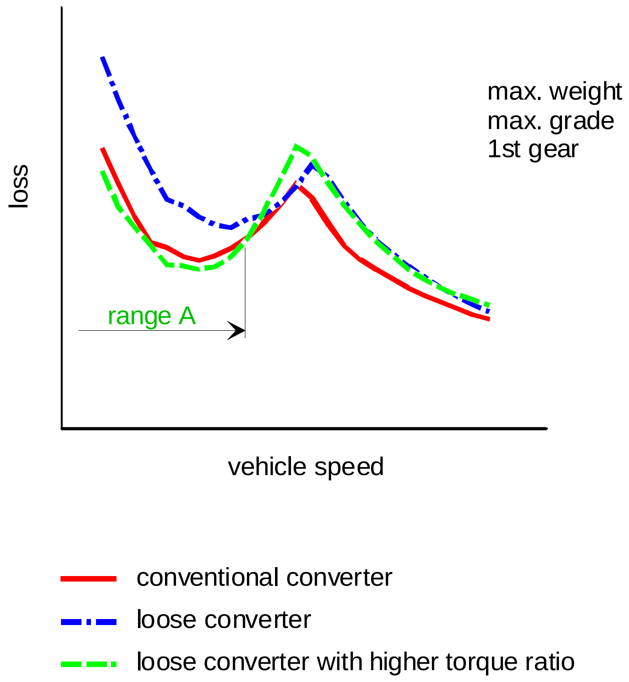


Figure 6: Effect of converter design on losses

Although reducing idle losses, lowering emissions in the cold phase and improving acceleration would seem to require a loose converter, in applications without torque converter clutches conventional converters are used in order to reduce losses during normal driving operation and to achieve an acceptable power response.

Torque Converters with Traditional Torque Converter Clutches

Losses in a traditional torque converter can be limited by using a torque converter clutch (TCC) with a conventional spring damper. Because of comfort problems - boom, rattle and tip-in/back-out reactions -, these TCCs can only be used in higher gears at average speeds, despite the use of the torsion damper. Even then, certain compromises with regard to comfort must be expected.

To clarify the problem of boom and rattle, figure 7 shows vibration amplitudes for the engine and the transmission output as a function of the engine speed. Depending on engine excitation and vehicle sensitivity to boom, a torque converter clutch can only be used at higher speeds. It is well known, however, that most of the time engines are running at relatively low speeds, which means that any reduction in fuel consumption is limited.

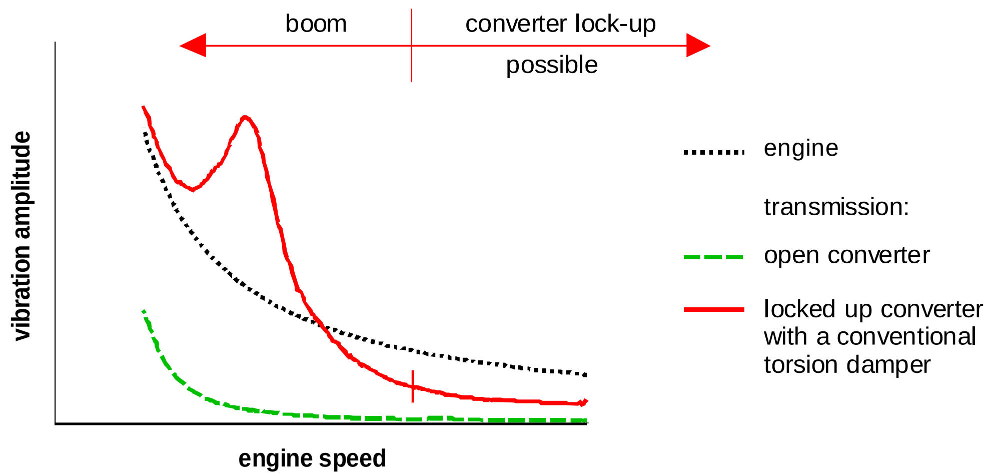


Figure 7: Torsional vibrations in the engine and the transmission

Tip-in/back-out performance is also a problem as well as changing gears and engaging and disengaging the torque converter clutch (Figure 8). If the driver steps on the gas when the torque converter clutch is engaged, he gets surge vibrations instead of the desired increase in tractive force. Then the torque converter clutch opens up, which in some situations even briefly cuts off torque transmission, before the driver finally gets the desired increase in tractive force.

Closing the torque converter clutch again can produce a drive train vibration.

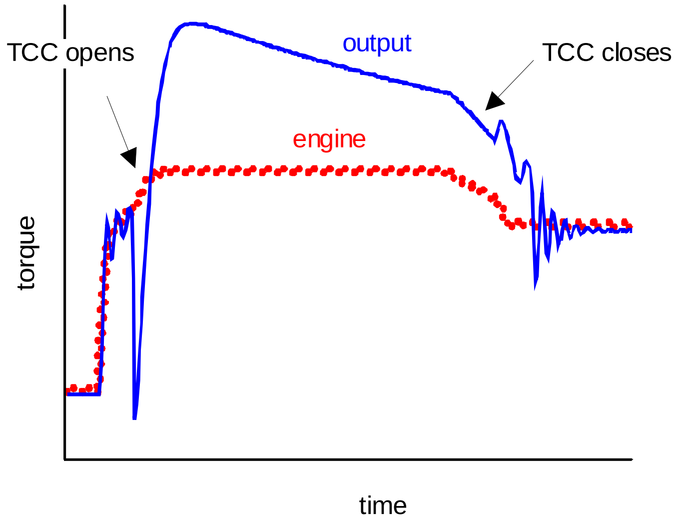


Figure 8: Tip-in/back-out cycle with a conventional spring damper

Comfort problems can also occur when changing gears, so it is customary to open the torque converter clutch when changing gears.

These effects are most apparent in the lower gears, so it is customary to use traditional torque converter clutch systems only in the upper gears.

This means that the torque converter cannot be locked up to reduce losses when driving up a steep grade in first gear. The losses in the torque converters are converted into heat, so the "looseness" of the converter is restricted by the capacity of the cooling system.

Furthermore, the power response demands during acceleration limits how loose the converter can be.

The Turbine Damper [3]

A Significant Step in Conventional Technology:

The problems of boom and rattle cited here for conventional torque converters can be reduced with torsion damper modifications. It would appear possible to achieve improvements by using a torsion damper with a lower spring rate. This is actually the case in some drive trains (Type A, Figure 9). Nevertheless, there are drive trains where this solution does not work (Type B, Figure 9).

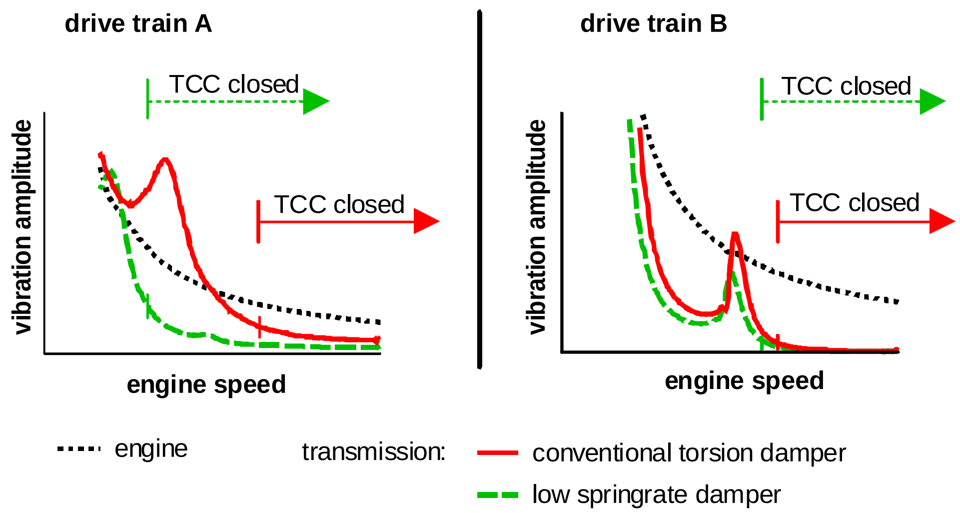


Figure 9: Effect of a low spring-rate damper

Why is this the case? The answer lies in the analysis of characteristic vibration modes and frequencies (Figure 10). In type A drive trains, boom is a function of the second characteristic mode. Relative torsion damper movement is fairly high during this mode, which means that damper modifications will have an effect. For Type B drive trains, on the other hand, the third characteristic mode is the problem. In this case, there is very little relative movement in the torsion damper, which means that changing the stiffness (spring-rate) has little effect. Based on the characteristic curve, one can assume that the stiffness of the transmission input shaft will have to be reduced.

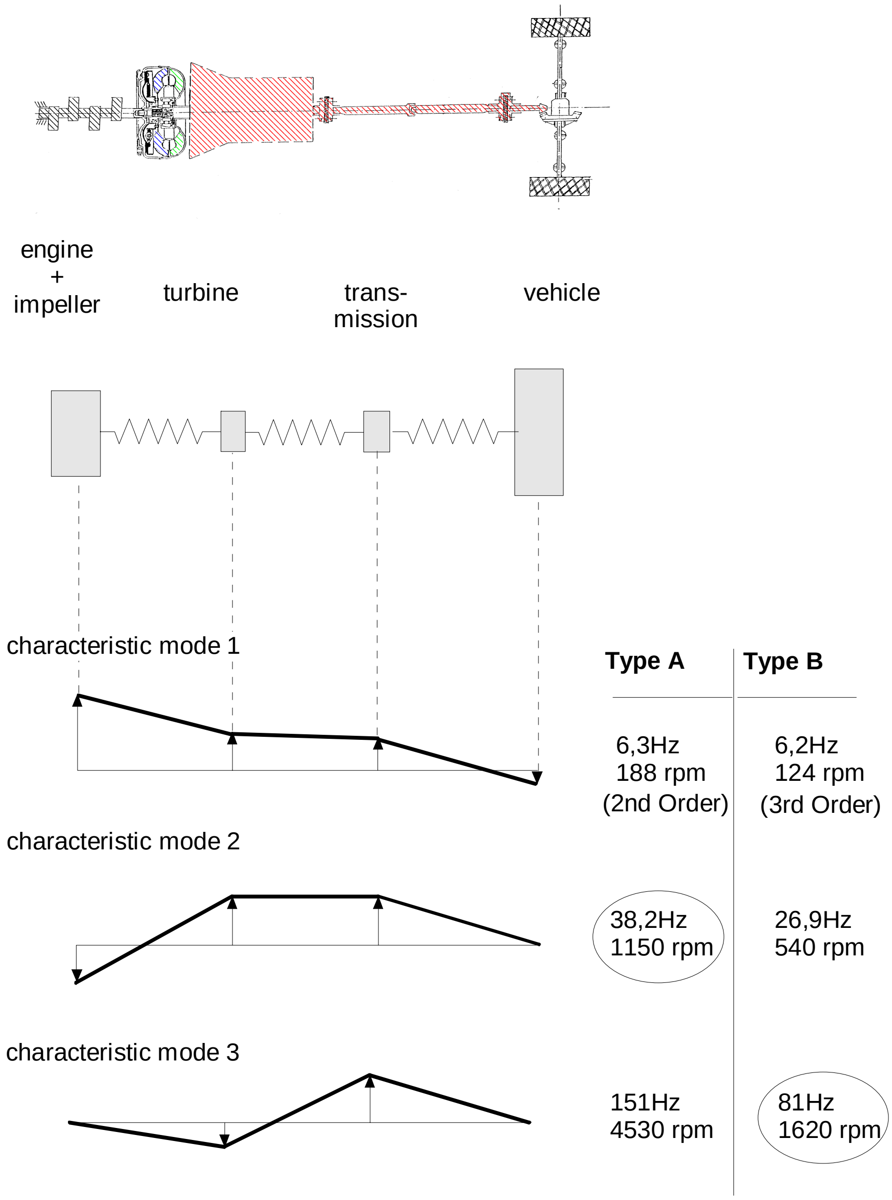


Figure 10: Natural frequencies in a vehicle with an automatic transmission

The amplitude curve in Figure 11 shows that a significantly softer transmission input shaft in a Type B drive train will achieve significant decreases in boom resonance, which cannot be achieved using a torsion damper with a lower spring-rate (Figure 9).

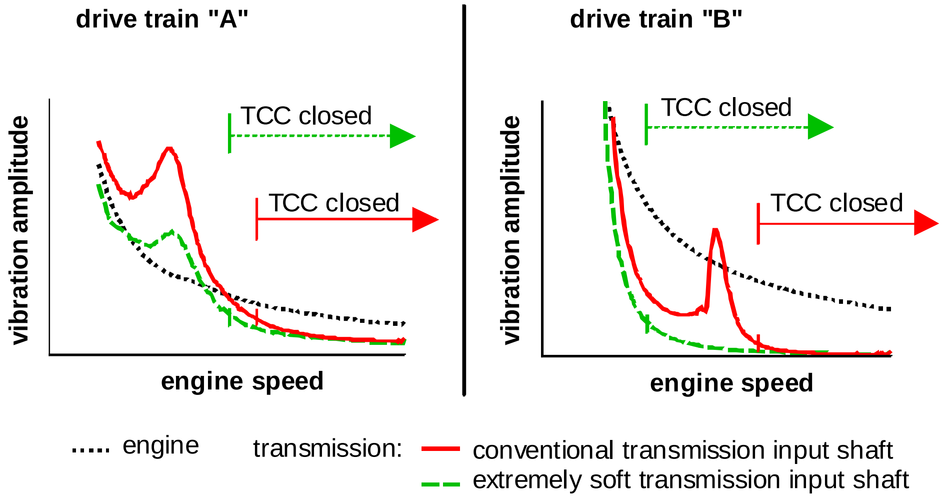


Figure 11: Effect of an extremely soft transmission input shaft

The stiffness of the transmission input shaft itself cannot be reduced to the required level, so a serial torsion damper is installed (Figure 12). The torsion damper between the engine and the turbine is removed and a damper is placed between the turbine and the transmission input shaft. LuK calls this design a turbine damper. It is important to note that with this design, power still flows through the torsion damper even when the torque converter clutch is open.

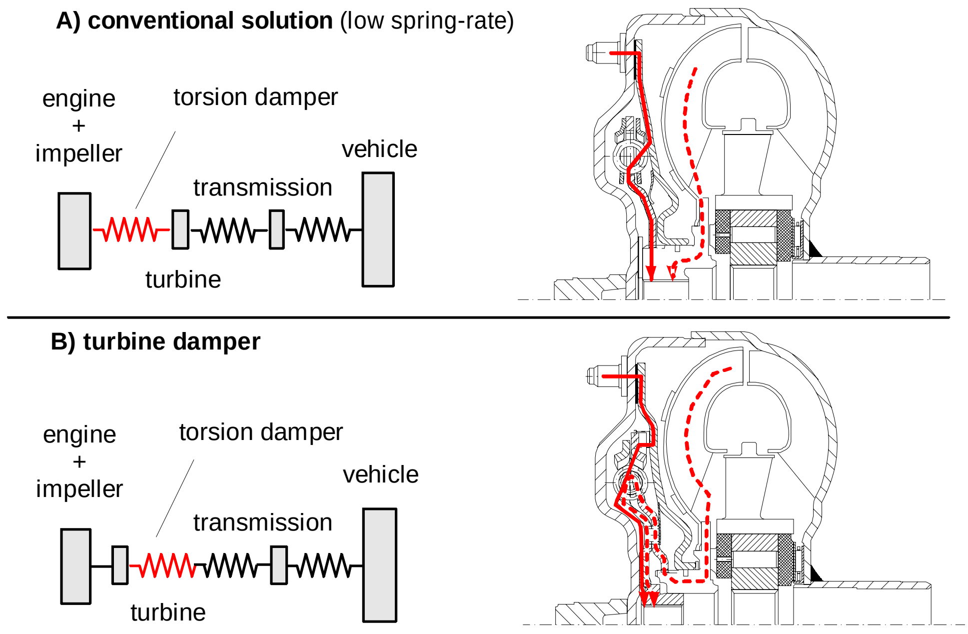


Figure 12: Turbine damper design

The choice of whether to use a LuK low spring rate conventional damper or a turbine damper depends on the drive train design. In comparison to other systems, both designs allow the clutch to be closed at a significantly lower engine speed. Depending on the customer control strategy - the tip-in/back-out peformance must be considered - a significant fuel saving can be achieved.

A Holistic Concept: The LuK TorCon System

What does the LuK TorCon System entail?

The LuK torque control clutch system (TorCon) consists of the following components: a conical slipping torque converter clutch, a mini-torsion damper, an adaptive control strategy and a loose converter.

Slipping torque converter clutches have been the subject of considerable debate for many years, have been introduced into production, and have been abandoned. One major problem is shudder due to hydraulic fluid damage. In some cases, control strategies result in comfort problems or in increased slip.

LuK has been aware all along that the slipping torque converter clutch has a very high potential, but also that full realization of this potential involves a three-step process:

- careful analysis of the interaction of mechanical, hydraulic and electronic systems (hy-mech-tronics)
- assignment of each function to the system that can best perform it
- ensuring that the three systems interact as effectively as possible.

Slipping Torque Converter Clutches

Theoretically, a slipping clutch offers the advantage that in addition to preventing high frequency vibrations such as boom, it is also capable of isolating low-frequency vibrations like those caused during tip-in/back-out cycles.

By reducing boom excitation, the slipping torque converter clutch, like the turbine damper, can be engaged at a lower engine speed than with traditional systems (Figure 13).

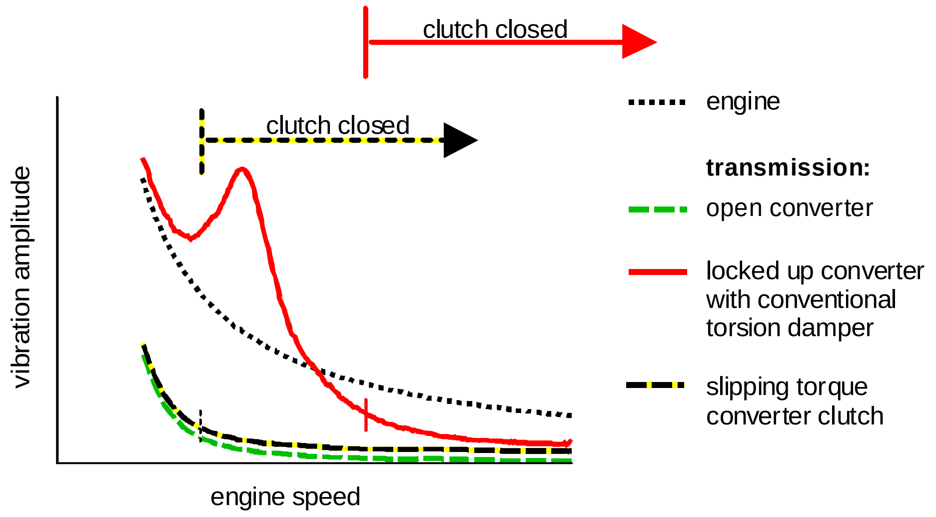


Figure 13: Effect of slip on vibration behavior

In comparison to traditional systems, a slipping torque converter clutch significantly improves tip-in/back-out performance and the supply of tractive force (Figure 14). No surging occurs when the driver steps on the gas because the torque converter clutch slips. The additional slip causes the converter torque to increase and prevents any break in torque transmission. As a result of increased torque conversion, torque increases continuously beyond the engine torque. The torque converter clutch can be engaged sooner, even in lower gears and at lower speeds.

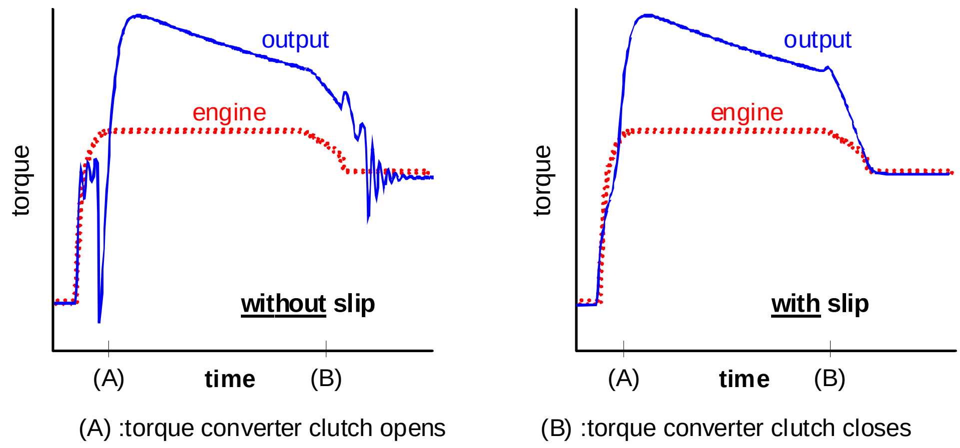


Figure 14: Tip-in/back-out performance with and without slip

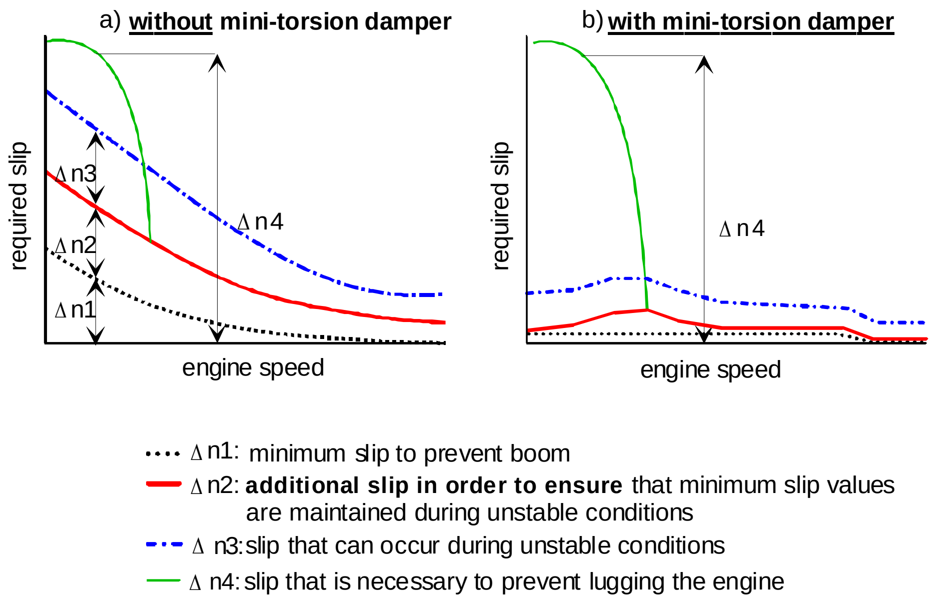
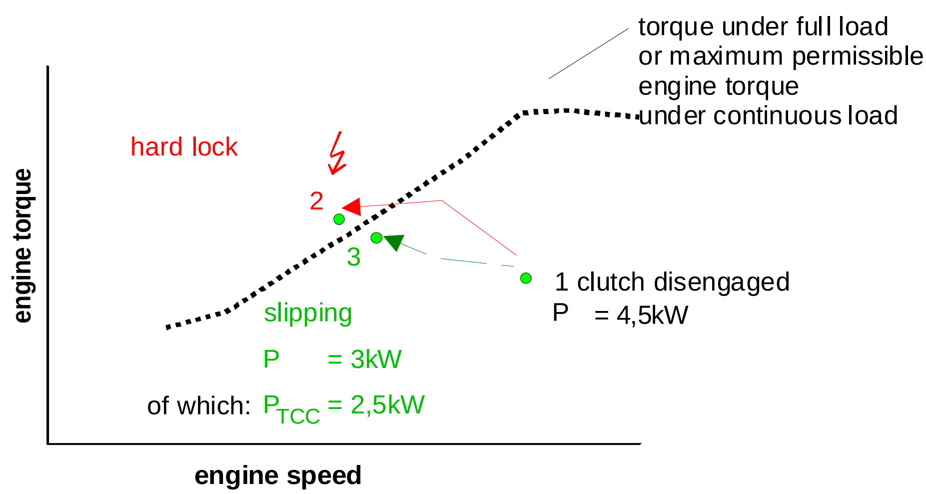


Figure 15: Slip requirements for designs with and without mini-torsion dampers



1) Converter clutch disengaged, **Total Losses = 4,5kW**

2) Converter clutch hard locked, required torque exceeds permissible values.

3) Converter with slipping clutch, **Total losses = 3 kW**,
Losses at the converter clutch = 2,5 kW,
Engine operating point for maximum fuel efficiency

Figure 16: Using torque control isolation to reduce overall losses

If a slipping torque converter clutch has so many advantages, why are so few units in operation today? The answer is that slip control also has its problems (See Figure 15 a for further explanation).

1. Preventing boom at low speeds usually requires a relatively high amount of slip. This means that total loss is also high. Decreasing slip often results in short-term sticking, which causes boom in many cases (n_1).
2. Low slip is difficult to control. Problems often occur if the control parameters are very stringent, but easing up on the parameters can result in significant deviations in slip values. In many cases, a open loop control system has distinct advantages, but even in such cases, slip fluctuations are hardly avoidable. Sticking can occur (with a possibility of boom) or slip can be too great (high loss).
3. The control system is imprecise. The lower the control torque, the more difficult it is to achieve exact control.
4. The system has a response time. Under unstable conditions, the control system requires a certain response time. During these conditions, slip values vary. The system must maintain a certain slip level in order to prevent boom, which again results in higher slip values (n_2, n_3).
5. With many engines, it is important not to lug the engine at low speed under heavy load or if the tractive force is insufficient, when the torque converter clutch is engaged (Figure 16). Driving at low speeds with high load can also cause problems with the cooling system. There are two options if it is important not to lug the engine at high loads. One can increase the engine speed by completely opening the torque converter clutch or one can let the torque converter clutch slip more. In fully open condition, loss will be unnecessarily high. If more slip is allowed, total losses decrease, but losses will increase in the torque converter clutch (n_4), which will increase clutch cooling requirements.
6. Heat build-up in the torque converter clutch must be dispersed. The essential problem with slip clutches is durability. Usually they will tolerate the heat built up during operation for a while, but after a few thousand kilometers, they begin to exhibit shudder problems. These problems are usually attributable to oil breakdown rather than to any problem with the friction lining. Petroleum additives are damaged by local overheating and over time affect all fluid in the system. Even when heat build up is relatively low, the facing should be well cooled. The design should provide maximum protection from local overheating! In addition to the heat caused by slip during vibration isolation, as well as the type of losses described in Points 1, 2 and 5, losses occur when the torque converter clutch engages and disengages. The lower the

speed and the higher the load at which the torque converter clutch engages, the larger the loss and greater the heat produced - especially if the system is designed for a comfortable converter clutch engagement.

One can counteract these problems by using a **simple torsion damper** (designed for partial load), a **conical design** and an **adaptive control strategy**. Figure 17 features a bubble chart that illustrates the interaction of these system components. The thick-lined bubbles represent customer requirements, and the shaded bubbles represent the components in the TorCon System.

The Mini-Torsion Damper

Advantages of a simple torsion damper (see also Figure 15b):

- Problem 1 (boom): the torsion damper will filter out the impulses caused by brief sticking, so no boom occurs.
- Problem 2 (control parameter problem): provides a partial solution. The torsion damper prevents brief sticking from causing boom.
- Problem 3 (control precision at low torque levels): The torsion damper assumes the vibration isolation function if the clutch briefly sticks, therefore the torque converter clutch can transmit a higher torque at low torque levels without control imprecision causing problems.

Slip can be maintained at a lower level. Slip prevents excitation in the damper resonance range, which means that friction elements are unnecessary in the damper. The mini-torsion damper is lighter and cheaper than conventional torsion dampers.

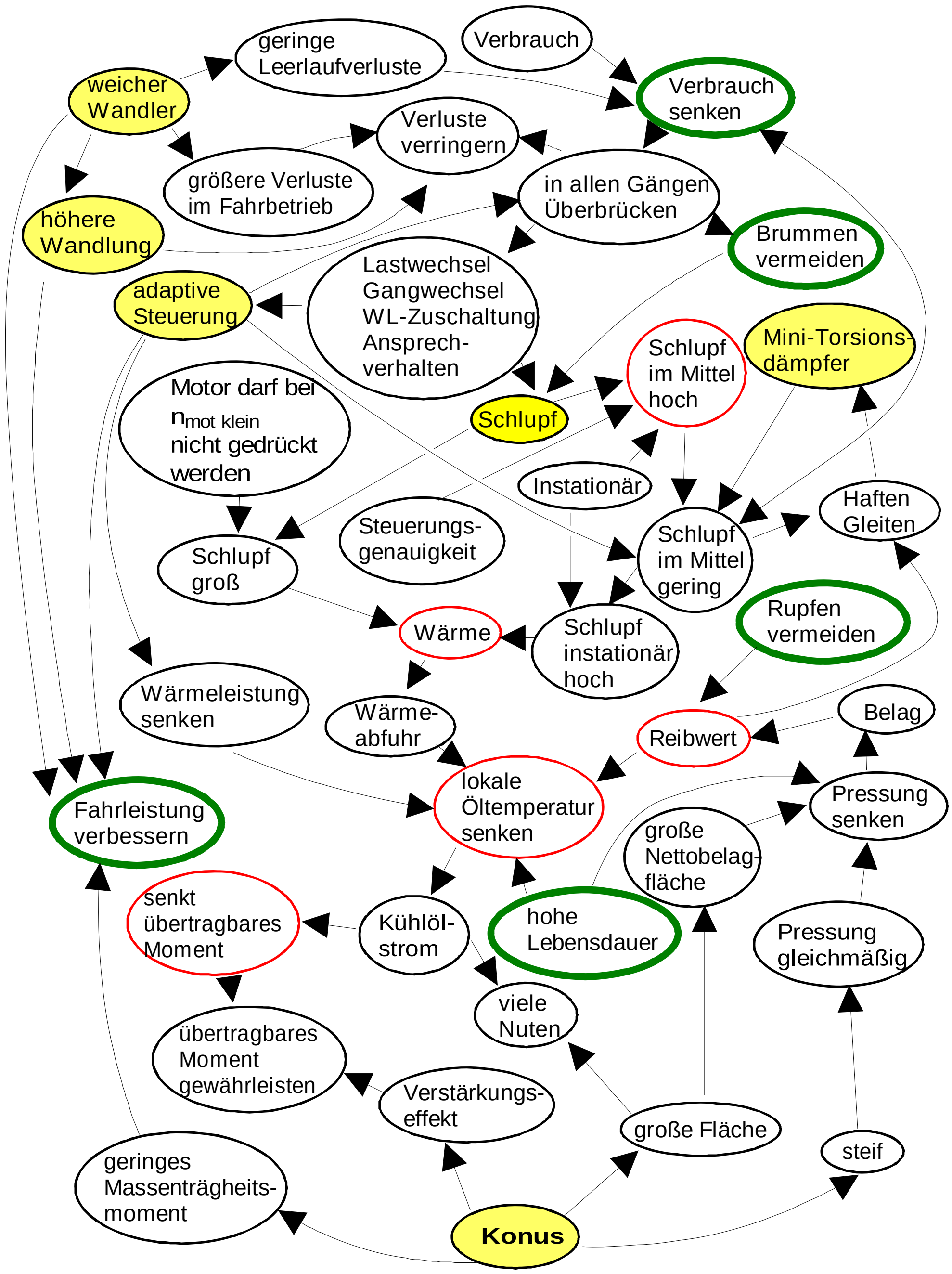


Figure 17: Bubble chart showing the interaction of slip, the mini-torsion damper, the conical design, and the adaptive control strategy

The Conical Design

The advantages of the conical design are primarily attributable to the stiffer design and the increased friction area:

- It is easier to dissipate heat build-up, which means that the maximum oil temperature will be lower for the same amount of heat. This factor helps solve problem 6 (dissipating the heat) and with that problem 4 (control response), problem 5 (lugging the engine) and the rest of problem 2 (the control parameter problem).
- More uniform unit pressure decreases lining load.
- Transmittable torque increases. Many of today's existing single-disc bypass clutches are operating at their capacity limits. The effect of coolant flow further decreases transmittable torque. The conical clutch compensates for this.
- The weight and the mass moment of inertia are decreased because the stiffer design allows use of thinner material.
- The converter ballooning decreases, which improves control capability.

The maximum local temperature has a significant effect on the service life of the oil. Lining cooling reduces the temperature in this area (Figure 18).

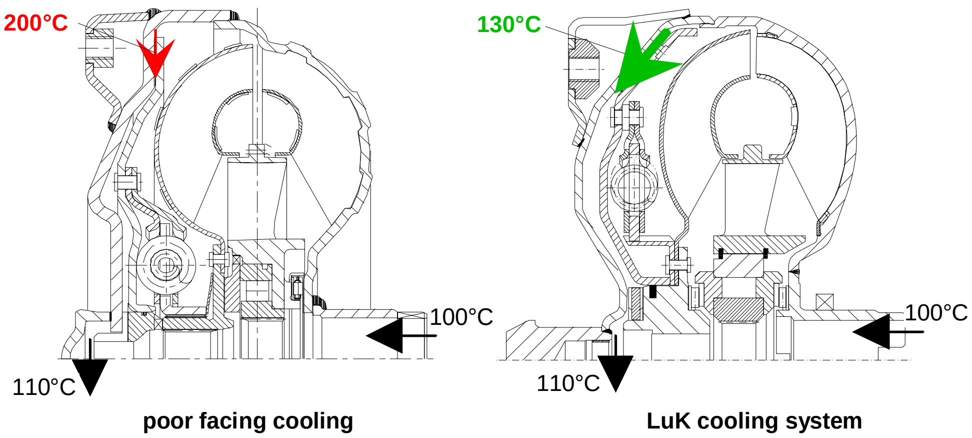


Figure 18: Facing cooling and maximum friction lining temperature

Cooling the lining decreases the bearing surface, but this is no problem with the conical design because lining unit pressure is more uniform (Figure 19). Furthermore, although the flow of oil decreases the transmittable torque, it is compensated by the amplification effect of the conical design.

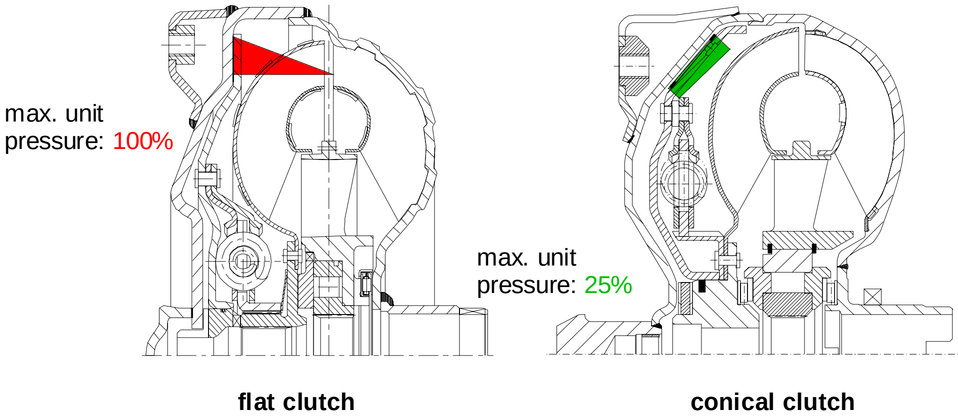


Figure 19: Unit pressure exerted on the facing

Control System Development

LuK has designed the *dyfasim* simulation program to support control system development. Figure 20 shows the basic program structure. In a typical simulation run, desired speed curves are specified and the automatic "driver" tries to follow the specifications.

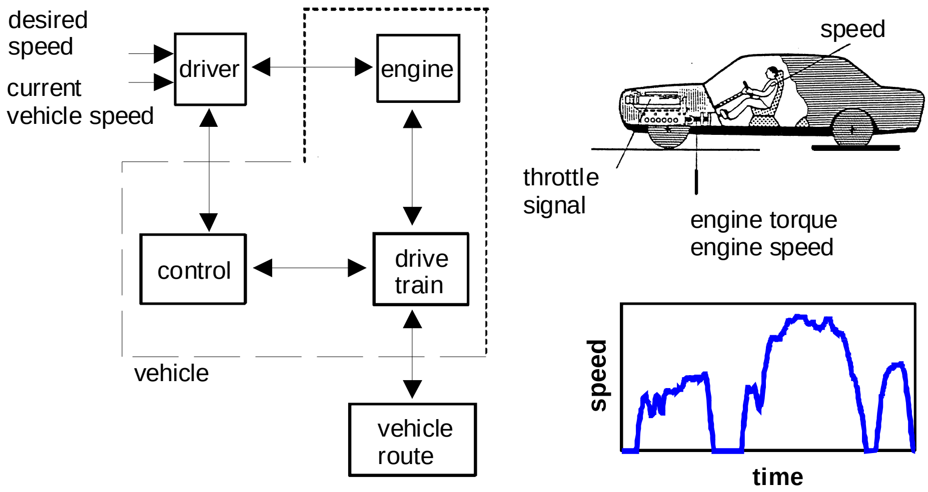


Figure 20: Total system study

Based on these calculations, it is possible to predict element load, shift quality, fuel consumption, etc. (Figure 21).

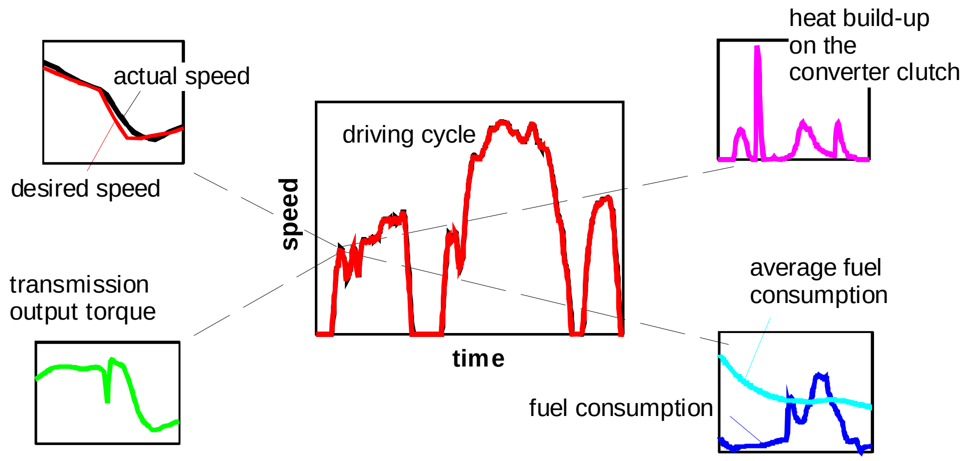


Figure 21: Simulation analysis of transmission system performance

Inclusion of the original control code in the simulation program significantly cuts down on development time (Figure 22).

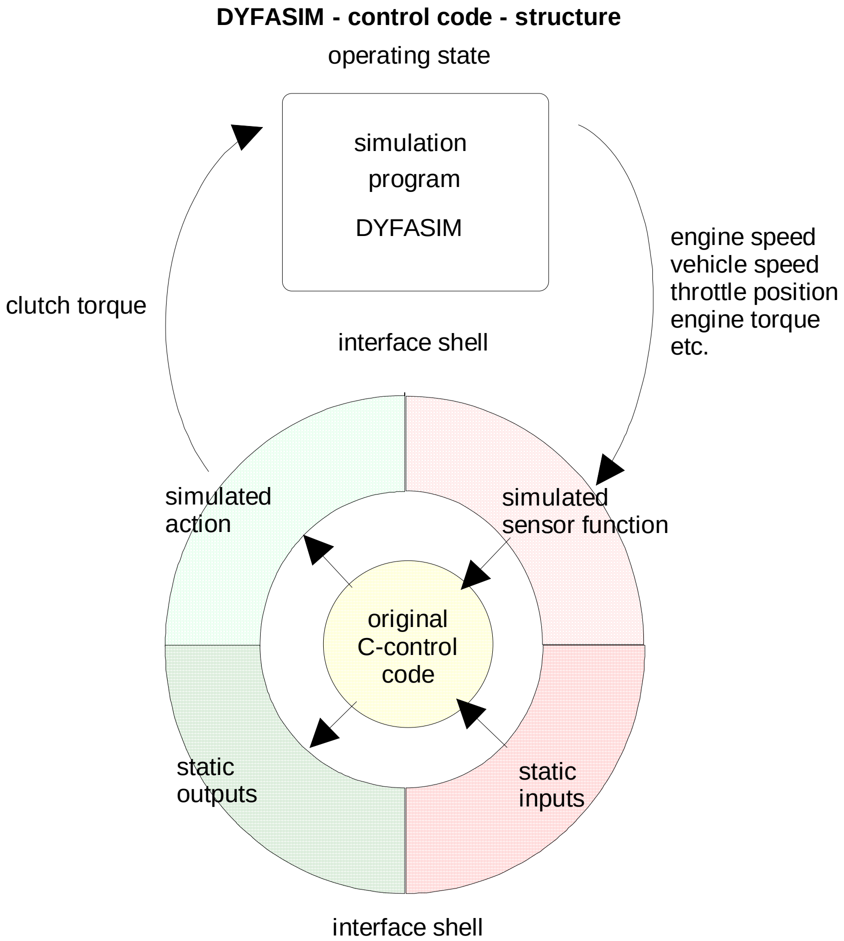


Figure 22: Integrating the control code into the simulation program

This procedure allows the development and testing of the control code long before the hardware is completed. The measured signals are not distorted by noise, so it is possible to conduct much more precise analyses. It is also possible to test performance in various situations and driving cycles, and the same program can be used to test control philosophy, to calculate fuel consumption, to test the control code, etc. Of course, it is not possible to precalculate all phenomena using the simulation program.

Subjecting every vehicle to an automatic long-term study reveals problems that might be overlooked, but long-term studies usually conjure up an image of paper printouts by the pound. In order to avoid the paper overload, measurements are taken at 100 Hz, but values are averaged and transmitted at a per second rate (Figure 23). If any unusual situation occurs, an automatic trigger function causes the system to store measured values at the 100 Hz rate for a specific time interval. Examples of special situations include exceeding a temperature threshold or an uncomfortable output torque or driver activation of the measuring button in the vehicle. This plotting technique drastically reduces the volume of measured data recorded without significantly decreasing the value of the results. To the contrary: the amount of real knowledge gained is increased because the evaluator doesn't have to plow through stacks of paper output or to settle for classified results.

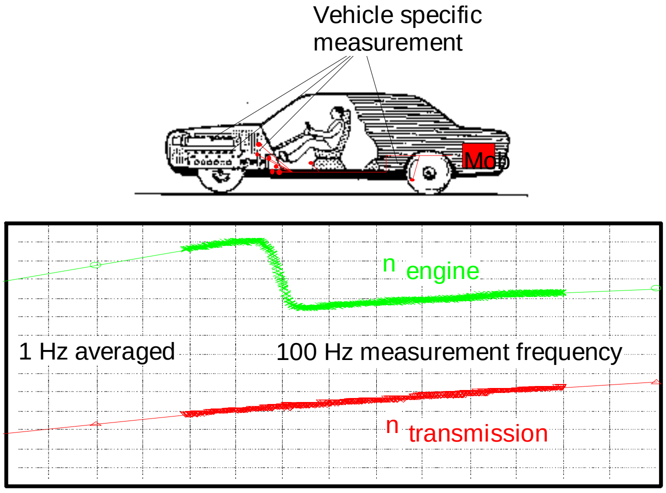


Figure 23: Long-term measurement

This development tool enables LuK to develop new transmission system products to a production-ready stage in a relatively short time and to ensure their reliability.

The Control Strategy

What control strategy was selected for the TorCon System? The problems that occur during slip control were mentioned above. The basic problem with any feedback control system is that a control deviation must occur before the controller can respond. Furthermore, there are ranges in which the default value is unachievable, for instance it is impossible to impose any higher slip than would occur with an open converter.

During shifting cycles, there is a negative effect if the controller works against the gear shift sequence. If the slip is set too low during upshifting for example, the converter clutch can stick at the end of the shift cycle, which compromises comfort. It is possible to come up with solutions to all these control problems, but they are not always optimal solutions. The LuK control concept utilizes torque control and an adaptive system to compensate for system deviations. Converter clutch torque is determined based on the engine torque.

$$\mathbf{M_{TCC} = M_{Engine} * converter\ clutch\ factor}$$

This means that there is no slip setpoint, which allows the controller problems described above to be avoided.

This control philosophy allows a definite reduction in slip to be achieved, as shown in the cumulative frequency graph (Figure 24).

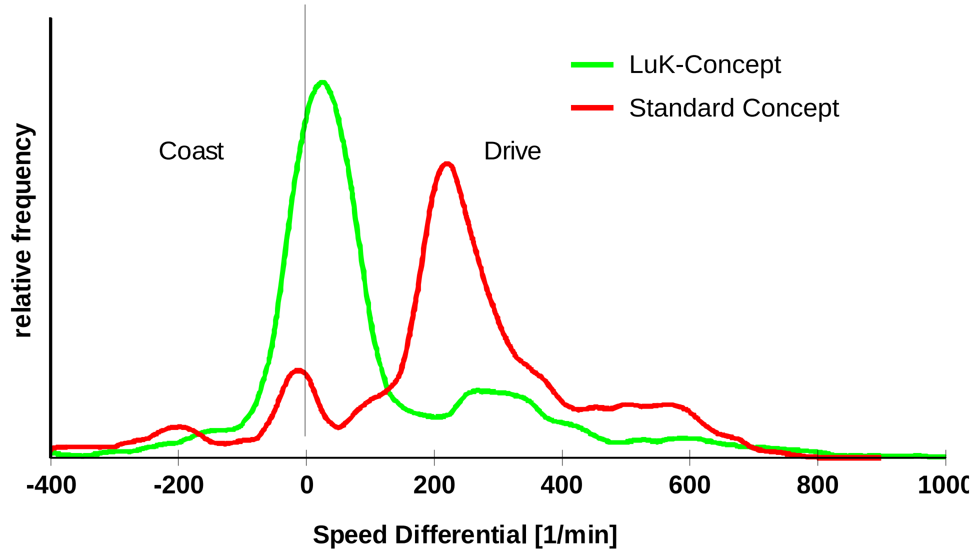


Figure 24: Slip distribution (EU without stationary phases)

Energy values determine whether the TCC is fully opened or whether it is allowed to slip. For instance:

When driving on a steep grade with a heavy load (3600 kg, 12%), the converter clutch cannot be fully closed at low speeds because of insufficient tractive force reserves or because the engine would lug. In such a case, the system constantly checks whether the total power loss is lower if the converter clutch slips or if it is fully open (Figure 25).

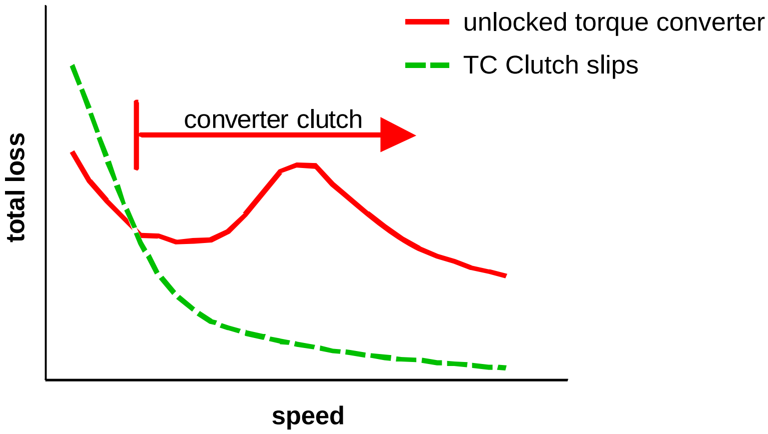


Figure 25: Driving on a steep grade

If the driver wants to increase tractive force, he increases the throttle position. Initially the engine torque increases. If this torque is insufficient, the driver again increases the throttle position to signal his demand for additional acceleration. Traditional systems usually involve downshifting to decrease the transmission ratio and increase tractive force. With TorCon, the system first checks to determine whether opening the converter clutch will increase tractive force. This would be the case if the converter would be in the conversion range after doing so. If this is the case, the converter clutch opens; otherwise, the system downshifts. The control system constantly monitors this function. In order to improve this interaction, it makes sense to adapt the transmission shifting curves to this concept. It is particularly effective to combine this tuning procedure with a loose converter (see next section). The following shifting curve graph approximates this philosophy (Figure 26).

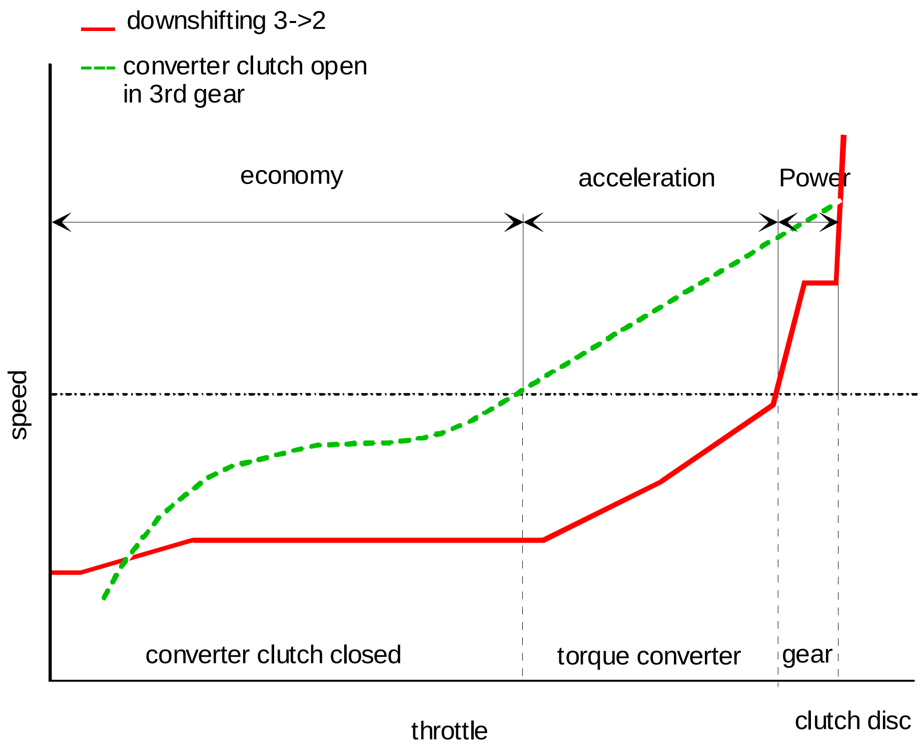


Figure 26: Control philosophy for the LuK TorCon System

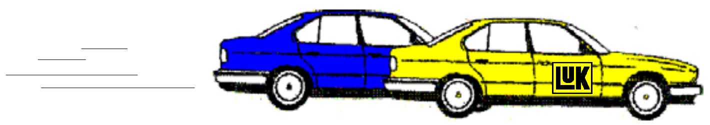
Introducing slip, a conical design, the mini-damper and adaptive control strategy achieves considerable improvement in fuel consumption, but adding a loose converter results in even further improvement.

The Loose Converter

This presentation started with a reference to converter design. Because it is impossible to engage a TCC in all operating ranges of current transmission systems, the converter has to be stiff enough to ensure driving comfort. The TorCon concept makes it possible to utilize the advantages of the loose converter without having to accept its disadvantages. The advantages include improved tractive force and less power loss in the stationary vehicle. Introducing a continuous-operation torque converter clutch eliminates the disadvantages, which include higher losses under load over a wide driving range, as well as the poor power response.

This design also achieves other significant advantages (Figure 27).

Driving performance is significantly improved, as is fuel economy. Emissions are improved disproportionately: testing cycles begin with a cold phase, but with an open converter clutch and a loose converter, the engine reaches operating temperature more rapidly, which has a positive effect on emissions. **Despite these definite advantages, TorCon does not increase the system cost, weight or overall mass moment of inertia compared to the current production standard.**



Consumption : (EU Test) — **1,1 L/100km less**

Acceleration : (0 - 100 km/h) — **0,8 seconds better**

Exhaust emissions : — **11 ... 36% reduction**

Figure 27: Statistics for the TorCon System

A cost-effective solution with many advantages:

Combining the TorCon System with a 4-speed transmission

Combining the TorCon System with a 4-speed transmission achieves similar advantages with respect to driving performance and fuel consumption as can be achieved with a 5-speed transmission and a traditional torque converter clutch, but overall weight and costs are lower than with a 5-speed transmission (quite aside from development cost savings and investment for production capacity).

A tractive force curve shows that combining a loose converter with the wider gear ranges of the 4-speed transmission results in higher tractive force than with a 5-speed transmission and a traditional converter (Figure 28).

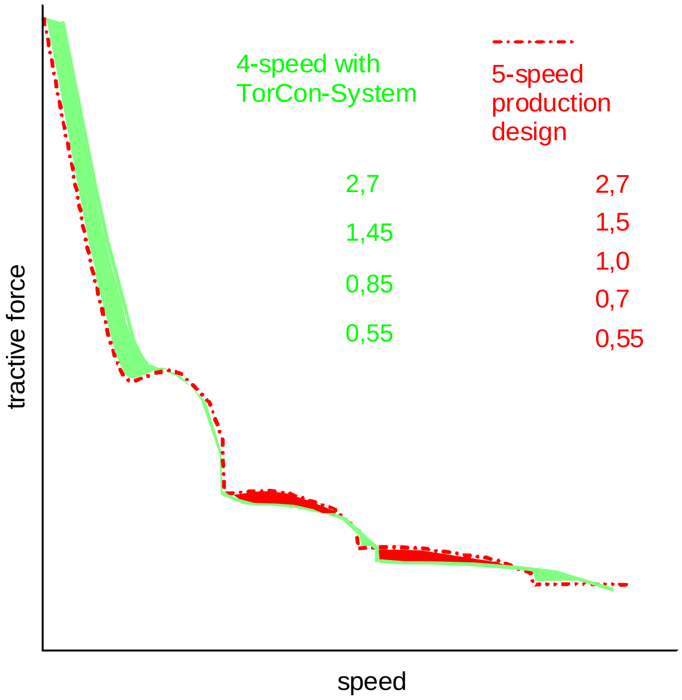


Figure 28: Full load tractive force curve for a 5-speed transmission with a conventional torque converter clutch and for a 4-speed transmission with the LuK TorCon System

The loose converter provides continuous compensation for the longer transmission gear ranges. The illustration also shows that the 5-speed transmission has to shift across two gear stages in the low-load range, whereas the 4-speed transmission with the TorCon System does not; this means that the system also reduces transmission shift frequency (Figure 29).

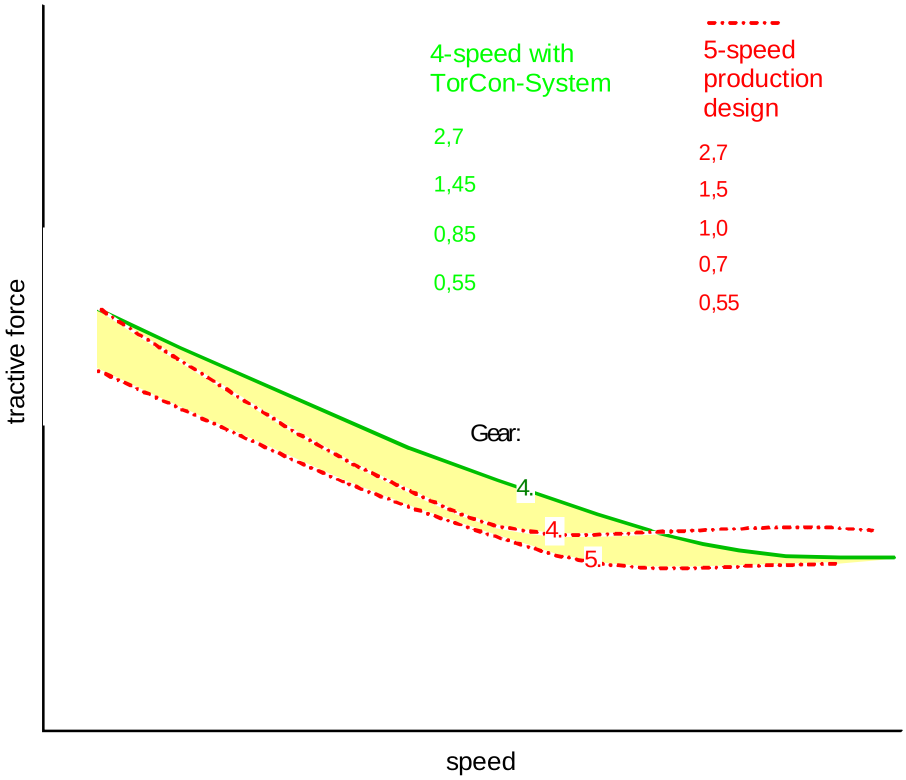
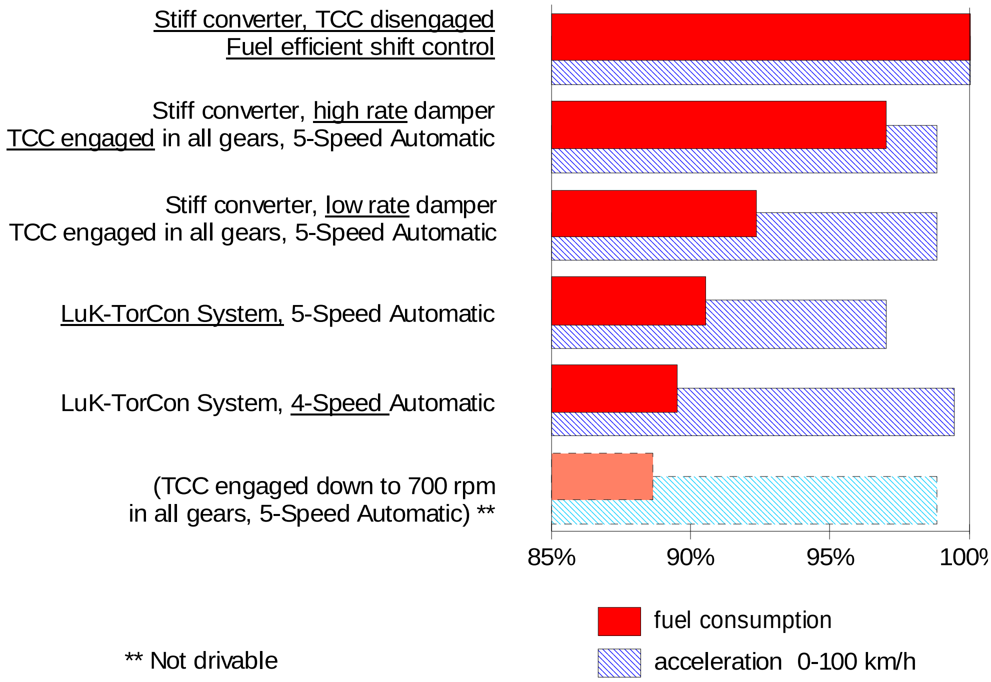


Figure 29: Partial load tractive force curve for a 5-speed transmission with a traditional torque converter clutch and for a 4-speed transmission with the LuK TorCon System

There is no great difference in acceleration from 0 to 100 km/h between conventional 4-speed and 5-speed transmissions because the transmission ratio in the lower gears is almost identical. Because of the loose converter design, the 4-speed transmission with the LuK TorCon System provide acceleration advantages in comparison to the conventional 5-speed transmission. A significant improvement in emissions is expected as well.

Summary:

As a system supplier, LuK provides a wide range of torque converter clutch solutions depending on the degree of integration the customer wants to achieve, whether these needs dictate a traditional torque converter clutch or a turbine damper, the TorCon System, or a TorCon with a 4-speed transmission. Figure 30 shows a final comparison of fuel consumption and acceleration criteria.



The high rate damper can be used down to 1600 rpm.

The low rate damper can be engaged starting with 1100 rpm. Option of turbine damper or conventional damper, depending on the drive train design.

The torque converter clutch is open during acceleration.

Figure 30: Comparison of various converter clutch systems

References

[1] VDI [Association of German Engineers] Report No. 1099 from the VDI-VW Joint Conference VDI-VW

[2] LuK Colloquium 1986, p. 5

[3] LuK internal report 047/94, H. Seebacher

Transmission Systems: A Comparative View

Dipl.-Ing. **Gunter Jürgens**

Introduction

Automatic transmissions have taken over in the USA and Japan, where they account for between 75 and 85% of the market. There are several reasons why this trend toward increased automation in the power train is to be expected in Europe. The automobile is becoming more and more just a means to an end - it is used to get from Point A to Point B comfortably and little operating effort possible. The "fun of driving" frequently disappears into the traffic gridlock, and more or less perfect clutch and shift lever operation becomes just another annoyance. Stringent exhaust and noise regulations require that vehicles be run at the optimum operating point - for instance during the warm-up phase. Without automatic gear selection, driver action could very well negate pollution control features.

Modern automatic transmission designs can compete with manual transmissions in fuel consumption and driving performance. The added cost is in the price range of a good car radio. The advantages of more relaxed driving and the world-wide statistics indicating fewer accidents with automatic transmissions should not be underrated.

This presentation will focus on several options for automating the power train, starting with the manual shift transmission equipped with an automated clutch and concluding with a look at continuously variable transmissions.

For purposes of comparison, these examples are all based on a vehicle with a 3 L engine because either production or prototype models of all the various automatic systems exist for this vehicle class. This comparison will include the following features:

- cost
- weight
- space required
- comfort
- fuel consumption
- driving performance

First, the transmission to be compared will be described.

5-speed manual transmission with an automated clutch (MT)

Figure 1 shows the outline of a manual transmission together with critical installation data.

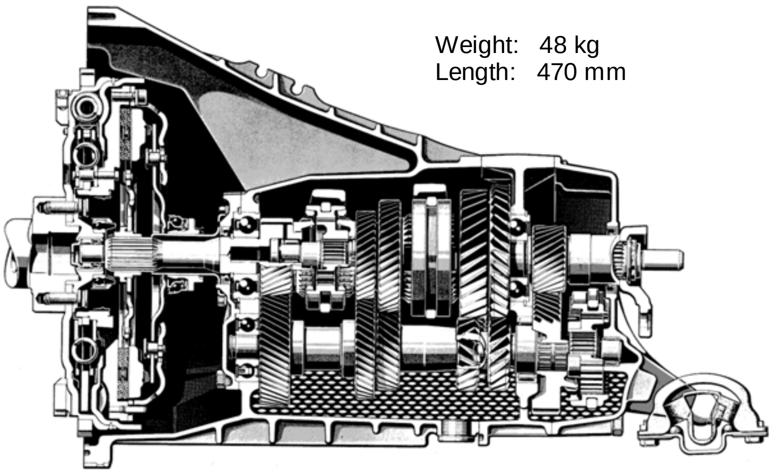


Figure 1: 5-speed manual transmission (MT)

The transmission is very compact and weighs only 48 kg, including the dual mass flywheel and the shift linkage. Figure 2 shows transmission losses in 1st and 5th gear as efficiency under street load [1].

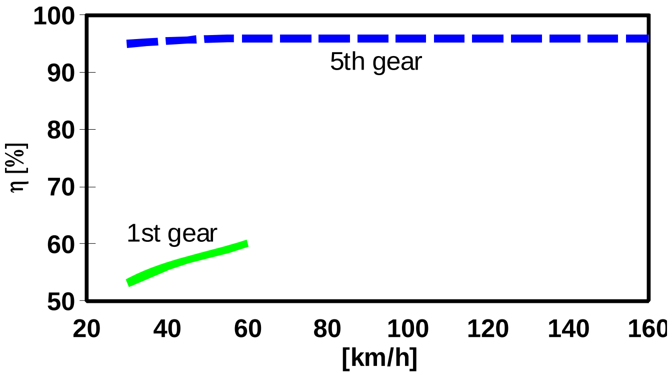


Figure 2: Efficiency under partial load

Losses for other gears range proportionately between these two values. Losses that are incurred as the result of electronic clutch management and slip strategies will be explained later when fuel consumption is compared.

These discussions will also account for the efficiency of the electrical drive and the battery.

The overall space required (including the clutch actuation system) is considerably less than for the automatic transmissions discussed later. Only the actuator - with the electonics incorporated - requires space in addition to the normally very compact manual transmission. The transmission has a total drive ratio range of 4.82.

This value is typical for the Power-to-weight ratio of the vehicle class treated in this study. It is not necessary to increase the transmission ratio for 1st gear (underdrive) because of the need to avoid exceeding the tire adhesion limit, and the ratio in 5th gear (overdrive) must not be too low because of acceptance problems with respect to acceleration capability in top gear. Consequently, there are logical limits to the drive ratio range [2, 3]. Even most 6-gear manual transmissions have drive ratio ranges of between 4 and 5.

Additional costs for automated clutch systems, including the flywheel and the gear-shift mechanism, currently lie in the range of 25 to 30% of base transmission costs.

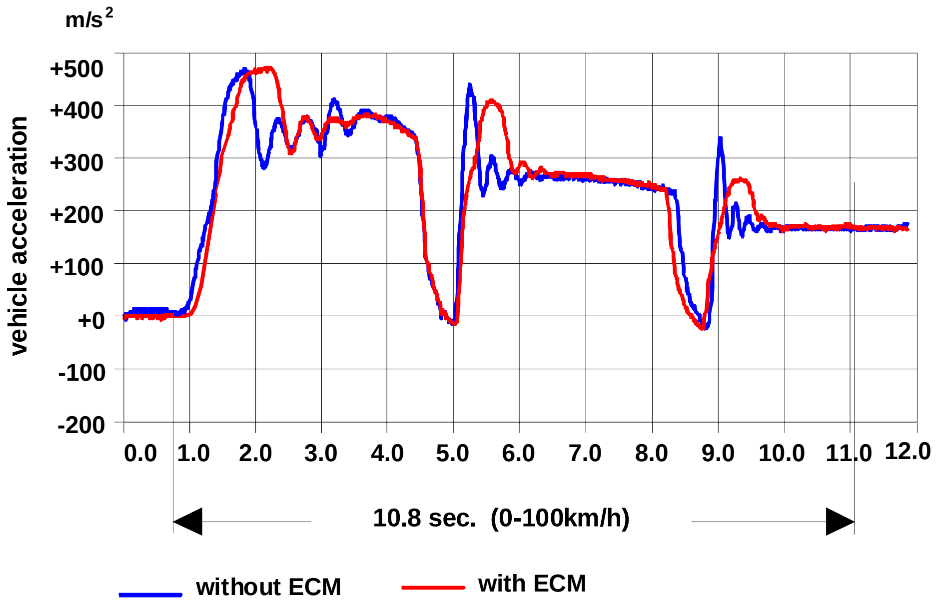


Figure 3: Comfort comparison for start-up (0 - 100 km/h)

Comfort during start-up and gear change is clearly improved for an average driver , as shown in Figure 3.

This figure compares acceleration under full load from 0 to 100 km/h for systems with and without ECM. The acceleration curve is a good indication of comfort. High acceleration peaks with resonant decay phases decrease comfort with non-automated clutches.

With automated clutch management, even inexperienced drivers shifting gears in the partial load range can achieve the same shift quality as with a modern multi-ratio automatic transmission.

Tip-in/back-out performance in engaged condition is often a critical point for power trains with manual transmissions. Figure 4 shows the potential that can be achieved with a good software strategy even without high clutch slip.

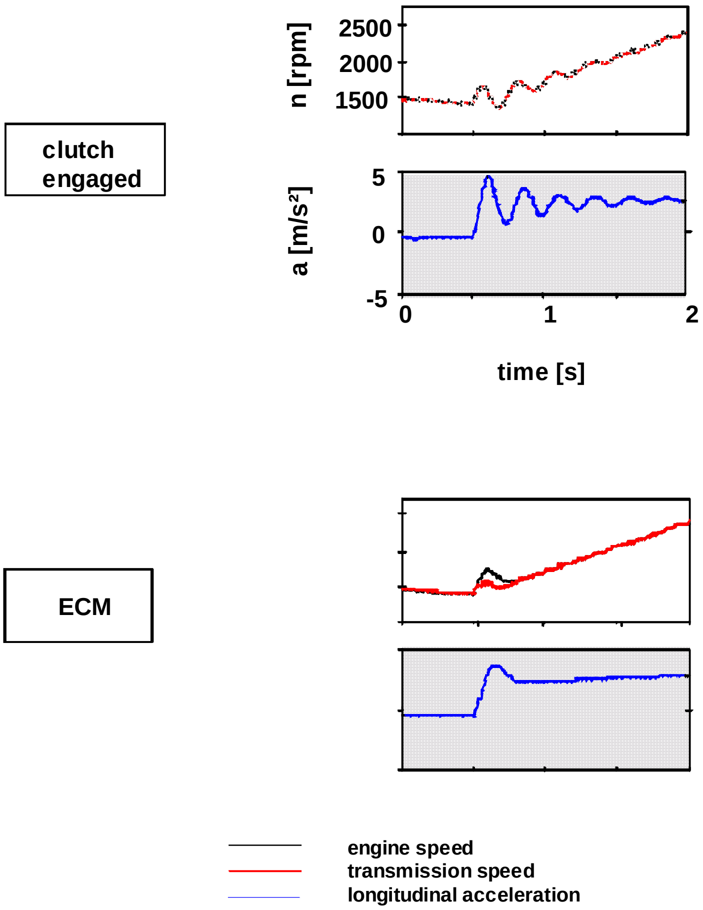


Figure 4: Tip-in/back-out performance with and without ECM

Automatic transmissions

When designers automate the clutch engagement process, it is obvious to think about automating gear shifting itself.

This solution is already in production for commercial vehicles, which often have more than 10 gears. Because conventional automatic transmissions with planetary gears would be very expensive and complex to build, designers have equipped the shift linkage in these systems with either semi- or fully automatic servo system operation. The additional expense of these systems, even for transmissions with up to 16 gears, is within an acceptable range when compared to what it would cost for a conventional fully automatic transmission. Shifting gears is, however, not fully automatic; the driver decides based on his own judgement or a shift indicator whether to up or downshift. The driver pushes a shift level in the desired direction to shift up or down; it isn't necessary to select the appropriate gear slot.

The interruption of tractive force - resulting from the clutch disengagement required to shift gears - occurs when the driver initiates the shift command and is prevented from occuring at an unwanted moment, which could occur with a fully automated system.

Although the additional expense for fully automatic as opposed to semi-automatic, demand-activated transmissions is quite minimal, the interruption of tractive force could, however, be one reason why no automated power-shift transmission has ever been introduced for production commercial vehicles.

Regardless of whether gear shifting is achieved using a servo cylinder or a stepped shifting mechanism, strategies need to be developed for engaging any gear under any circumstances. Because torque transfer in the synchromesh gearset is achieved using gear teeth, i.e. via positive contact, it is possible for the gear teeth to be touching at the moment the driver decides to engage gears. Under these conditions, it is impossible to complete the shifting operation without an additional adjustment to the system. This occurs sometimes with manual transmissions - particularly in first gear and reverse - and can, for instance, make it necessary to circumvent this problem by rotating the shafts another turn by reengaging the clutch in the neutral slot.

When only a few gears are involved, it costs almost as much to add an automatic gear selection feature as it does to introduce a fully automatic transmission, so this option has very little chance of establishing a market position. Although this design offers some slight fuel savings, in comparison to a fully automatic transmission, these savings are outweighed by the decreased shifting ease due to the interruption of tractive force.

Dual Clutch Transmission

Some of the problems cited above, such as the interruption of tractive force, can be circumvented with dual clutch transmissions. The main feature of these transmissions is that they actually consist of two intermeshed transmissions linked to a single output shaft. Each transmission has its own clutch.

The desired transmission ratio is selected by engaging the usual synchronizer in either sub-transmission 1 or subtransmission 2. It is possible to shift from one transmission to the other without interrupting the tractive force. If handled skillfully, controlled shift selection can be introduced virtually without disadvantages. For more than 5 gears, this transmission principle is equal to a planetary gear transmission. One of the two power shift clutches or perhaps an upstream torque converter with or without a bypass clutch can be used as the start-up component. Basic designs [4] demonstrating this principle already exist (Figure 5).

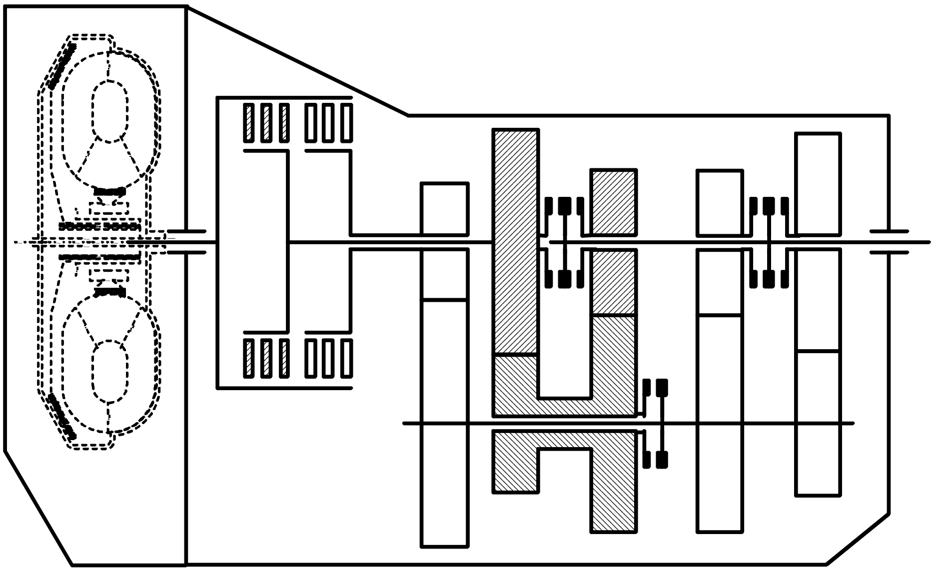


Figure 5: Dual clutch transmission

Figure 6 illustrates a shift mechanicsm that operates without interrupting tractive force. For purposes of simplicity, one sub-transmission is represented as a shaft with a single drive ratio and a second drive ratio is obtained by pairing with a spur gear. Despite the speed differentials involved, torque can be transmitted via both clutches, but the sum of the torque values from both clutches must be accounted for. For instance, if one clutch transmits the full engine torque, the 2nd partially activated clutch only generates losses as a result of its slip. If the transmittable torque from clutch 1 is reduced the torque from clutch 2 synchronously increases, the engine will be accelerated or decelerated to the speed of the other transmission train. Fine-tuning these procedures is easy with a fully electronic control system. Both theoretical and practical experience indicate that this design can be used to achieve the same shifting ease as with a planetary gear system.

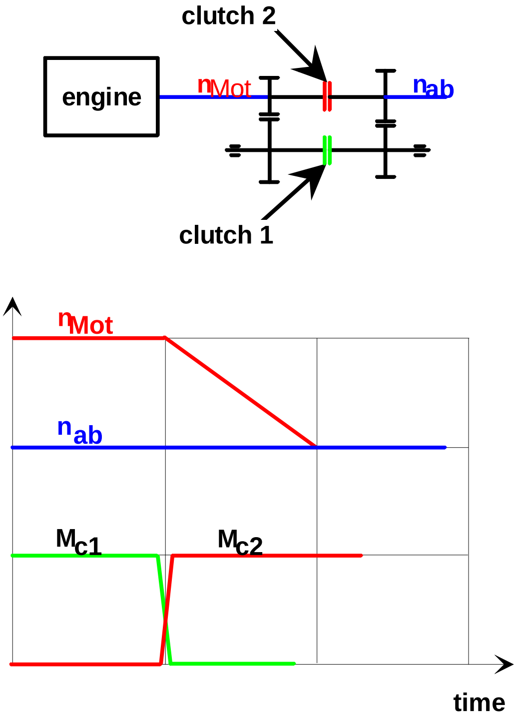


Figure 6: Shifting procedure without interrupting tractive force

In comparison to a manual transmission, additional costs include: the division of the transmission into two sub-transmissions, the additional clutch, the cost of the automatic controls for the two clutches, the operating systems for the synchromesh elements and the hydraulic oil supply in the event that a torque converter is used. Losses can be lower than for a planetary gear system.

Depending on the number of gears involved, Dual clutch transmissions - as the name would indicate - operate with two friction clutches and, depending on the number of gears, several positive clutches, which are usually combined with synchromesh elements.

4-Speed automatic with hydraulic torque converter

Automatic transmissions with a friction clutch for virtually each gear have been around for a long time. Whether this transmission is designed with a layshaft or as a planetary gear transmission, at least each of the forward gears is switched by means of a friction clutch.

4-Speed automatic transmission with layshaft

Figure 7 shows an example of a 4-speed automatic transmission like those that have been built in the USA and Japan for many years.

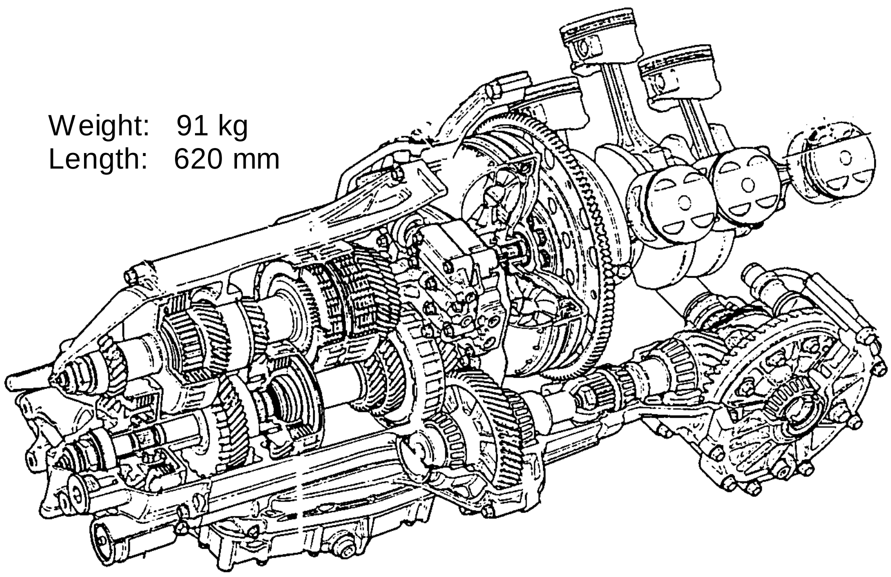


Figure 7: Automatic transmission with layshaft

The advantage of this transmission design is that it provides a relatively free ratio selection because each transmission ratio uses its own set of gears. A disadvantage is that the clutch diameter is limited by the distance between the shafts. State-of-the-art gear design and manufacturing procedures make it possible to suppress gear noise to the same level achieved by a planetary gear transmission - in other words, these transmissions are virtually noiseless. With or without one-way-clutches, they provide the same shift quality as planetary gear transmissions both in theory and in practice.

Planetary gear transmissions

The first automatic transmissions were manufactured in the United States for high-torque engines. Planetary gear transmissions were used at the time because of the power density involved and remain the standard.

As was the case with the dual clutch transmission, two power-shift clutches are used to shift gears without interrupting the flow of tractive force. If the torque converter has been retained as a start-up element, the total costs for this system are comparable to the cost for planetary gear transmissions with 5 shift elements, for layshaft transmissions with the same number of elements or for dual clutch transmissions (Figure 8).

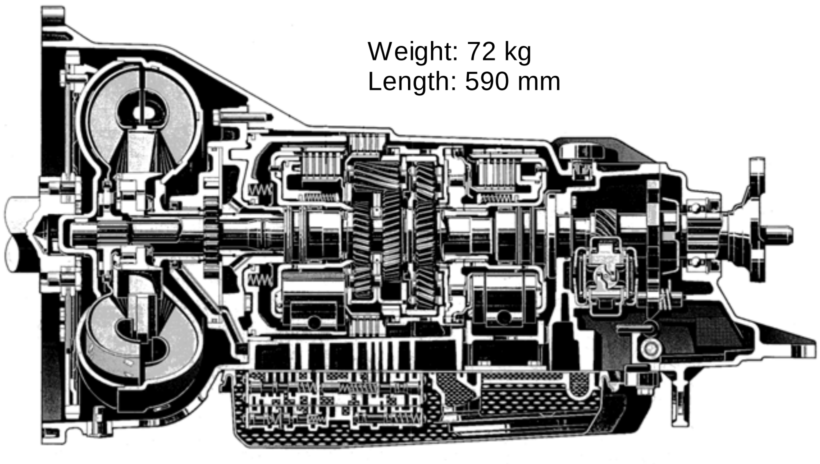


Figure 8: 4-speed automatic

All these transmissions require an oil pump for the transmission fluid supply to the torque converter and for the hydraulic control system. The hydraulic control system is basically in effect an analog, partially digital hydraulic "computer". Even when equipped with an electronic control unit, some of the control functions are still assigned to the hydraulic control system. Because oil is used as the operating medium and because wet clutches - in contrast to the clutch used in manual transmissions - cannot transmit torque without any pressure or load, the pump must run constantly. Figure 9 shows the total resulting efficiency rating for a 5-speed automatic transmission with a hard locked torque converter under street load. Efficiency ratings can vary depending on the design.

The total drive ratio for an automatic transmission in first gear can be somewhat "longer" than for a manual shift transmission if a torque converter is used to assist start-up. The drive ratio in top gear is usually somewhat "shorter" than for a manual shift transmission, which naturally leads to higher consumption at higher velocities, e.g. freeway driving.

Start-up comfort is similar to MT with ECM, but the ability to shift gears without interrupting tractive force is readily apparent. This elimination of the "shift interval" reduces the acceleration deficit that results from the lower number of transmission ratios.

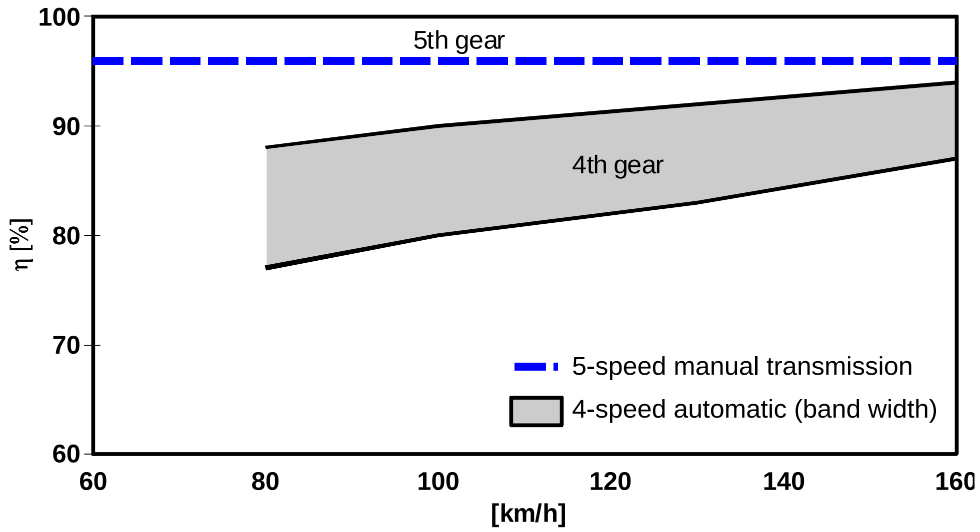


Figure 9: Partial load efficiency

The total weight of the 4-speed automatic with a torque converter is significantly higher than that for the 5-speed manual transmission. There are several reasons for this:

- The torque converter weighs about as much as a dual mass flywheel, which is most frequently used in the vehicle class in question.
- The friction elements used to transmit power flow - i.e., the full-load shift clutches - add weight to the design. These full-load shift clutches are usually oil-cooled, but a certain amount of shifting heat must be stored in an intermediate thermal mass, which of course adds weight.
- The required quantity of oil is significantly higher (6 - 8 l) than in a manual transmission.
- The hydraulic control system and the oil pump also weigh at least 5 kg. If the oil cooling system is added, then a transmission for this vehicle performance class has a total weight of between 80 and 90 kg, depending on the design used.

The design volume of the torque transmitting components (shift elements, planetary gear sets and shafts) can be about the same as for the corresponding components of a manual transmission, but the hydraulic control system and the oil sump provide additional volume. In addition to the cost, this is the main reason why, although 5-speed manual transmissions are used in all front-wheel drive vehicles, generally only 4-speed automatic transmissions are used in these vehicles.

In comparison to manual transmissions, the total costs (including the flywheel and the clutch, etc.) are considerably higher. The additional cost can vary considerably depending on the design. For instance, if one-way-clutches are used to simplify gear shift control, they have to be bypassed with additional clutches in coast.

5-speed automatic transmission with hydraulic torque converter

If one wants to retain start-up gradeability with an automatic transmission and at the same time achieve an overdrive ratio similar to that of a 5-speed manual transmission, it is usually necessary to add a gear. Until now it has been impossible for the torque converter to completely make up for the required gear ratio spread because power losses, for instance on a steep grade with a trailer, have been too great. In order to continue to take advantage of current investments in 4-speed transmissions, a supplemental planetary set gear with its shifting elements is frequently added in order to end up with a 5-speed transmission with a wider total drive ratio range. In terms of total range, these transmissions are comparable to 5-speed manual transmissions. In fact, with the start-up assist provided by the torque converter, they even operate with a more pronounced overdrive effect. The lower tractive force in the overdrive range can be made acceptable by shifting into the second highest gear. Unfortunately it has been very difficult to come up with a successful schedule. Several test reports criticize high shifting frequency in the top two gears.

Adding the extra planetary gear ratio with two shift elements increases costs by about 25% compared to the basic design.

The weight situation is significantly worsened as well - this type of 5-speed transmission can weigh up to 100 kg. In terms of design volume, this concept is only feasible for standard drive trains because the additional length required for the additional set of gears makes the system too long to install in a front-wheel drive vehicle. The additional fifth gear also leads to higher losses, as indicated in Figure 10 [5]. One should be aware that the input speed which is mainly responsible for the losses is lower than with a 4-speed transmission.

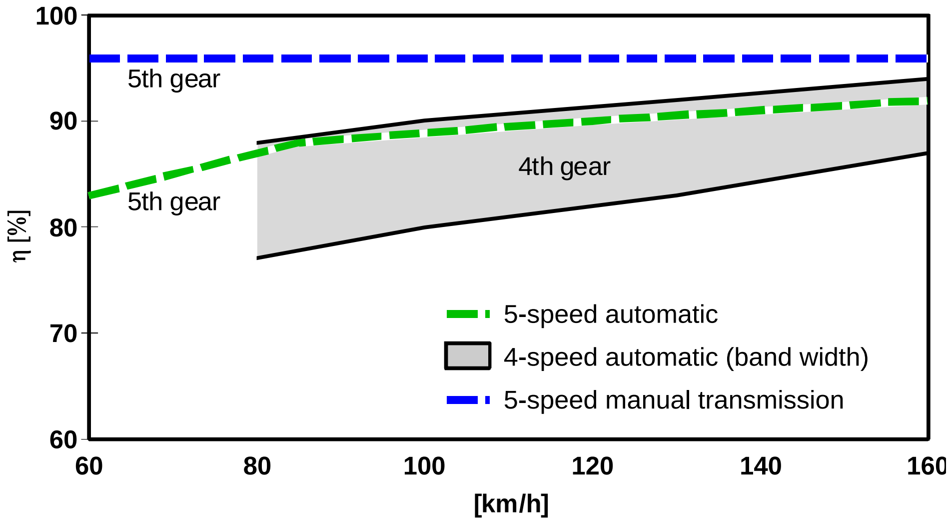


Figure 10: Performance losses for a 5-speed automatic transmission

Despite the increased weight and the higher losses, the significantly higher overdrive ratio of the fifth gear results in lower fuel consumption and improved driving performance over the 4-speed automatic. Figure 11 shows such a transmission.

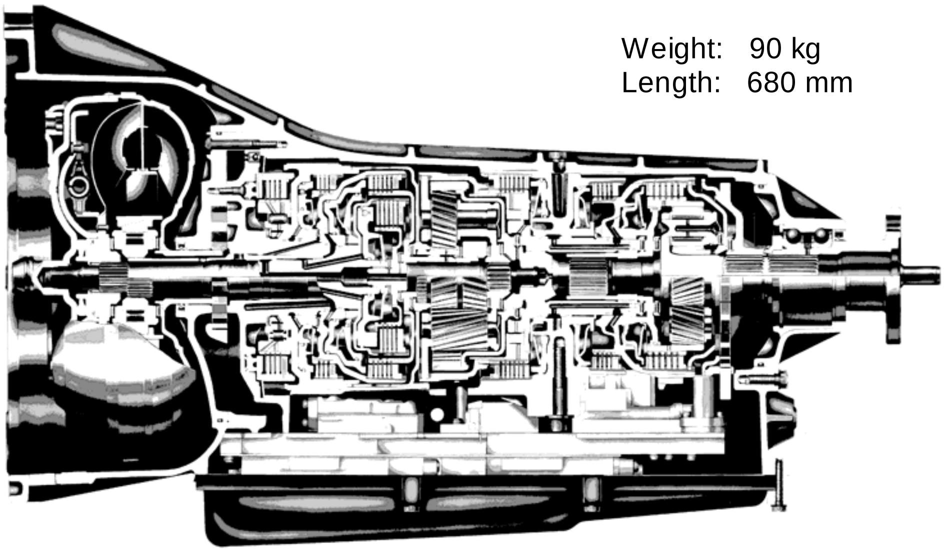


Figure 11: 5-speed automatic

Modified 4-Speed Automatic with the LuK TorCon System

The first automatic transmissions had only two, at most three, mechanical gears. They also featured hydraulic torque converters with large torque ratio and high stall speeds. These design features produce comfortable driving conditions, but they had large losses as well. Over the years losses were reduced by introducing stiffer converters. The large torque ratio was no longer possible and must be compensated for by increasing the drive ratio range with more gears.

At the beginning of the 70s, torque converter clutches were introduced to eliminate converter slip losses, at least in top gear, resulting in a transmission with improved fuel consumption, but offered less comfort. In an SAE paper from this period, one developer noted with regret, "when we locked the converter, we discovered the advantages that we had lost".

Efforts have been made to optimize the shifting strategies of the automatic transmission and the torque converter with the bypass clutch: For instance, with adaptive shifting programs or program selectors. It has proven difficult to find the ideal compromise between reduced shifting frequency and either fuel savings or performance related gear selection. Even if shifting is almost unnoticeable, drivers are still aware of acoustic changes associated with changes in engine speed. A multi-ratio transmission always provides an appropriate match between engine speed and driving speed. Only in high-slip ranges, such as can occur with loose converters, will the engine speed remain almost constant for acceleration and shifts. Even the introduction of an additional number of gear ratios - supposedly the route to an continuously variable transmission produces its own problems because of increased shifting frequency. Would more gear ratios - more then five - improve this?

LuK has found another solution to the problem.

One should take another look at the properties of the hydraulic torque converter as a comfortable, continuously variable transmission combined with a bypass clutch. Building on this model, other power train designs with transmissions having fewer speeds can be visualized. The total transmission ratio in first gear should still be very high in order to provide high tractive force for trailer operation and similar tasks, but with low power losses of the torque converter in partially or fully bypassed condition. This design features a mechanical overdrive and relatively few intermediate gears, each of which covers a wider speed range and is comfortably bridged by the continuously variable transmission "torque converter". If the torque converter only comes into play briefly during acceleration phases, short-term loss in efficiency is relatively unimportant. In many cases, the often critisized shift frequency, specially out of overdrive, can be eliminated by using a torque converter with wide conversion range; made continuously variable by partially or fully opening the bypass clutch. Figure 12 shows that the fourth gear of a 4-speed automatic equipped with the increased conversion range of the LuK TorCon System successfully covers the same range as the fourth and fifth gears in a 5-speed automatic for the speed range between 60 and 140 km/h.

Costs, power loss, and weight for this kind of 4-speed transmission are comparable to the advantages of a 5-speed automatic. Whether we can design the wider drive ratio range depends on the type of planetary gear sets used and the available free space for bearings and other elements. The additional development is less; maybe it is only required for a redesign of the hydraulic control system or something similar.

Especially in the case of automatic transmissions with layshaft design, the cost of making the changes in the ratio of the transmission is very low. The additional investment for the design change is usually no higher than would be the case with an upgrade to a 5-speed transmission. It is not necessary to incur the cost of a total reinvestment to supply an optimum 5-speed transmission design.

Figure 12 shows drive performance figures for a 4-speed vehicle with LuK TorCon System compared to a 5-speed automatic transmission. The two vehicles feature the same transmission ratios for first gear and for overdrive. Acceleration in the 4-speed was even improved because of the higher torque ratio, as was fuel economy because of the overdrive ratio and the lower losses compared to the 5-speed transmission.

Naturally, setting aside weight, cost and space a 5-speed automatic with the same drive ratio range would be better in a few operating ranges. Theoretically, every point in the engine operating curve has an ideal transmission ratio that results in the optimum fuel economy. However, one must keep in mind the higher losses and the very difficult shift philosophy optimization. It appears that, in any case for front wheel drive vehicles, the new "old" concept with few gears and an optimized torque converter remains the best. If one uses this concept with a 5-speed transmission with a wider total drive ratio, one can achieve the advantages of a 6-speed transmission of the current type.

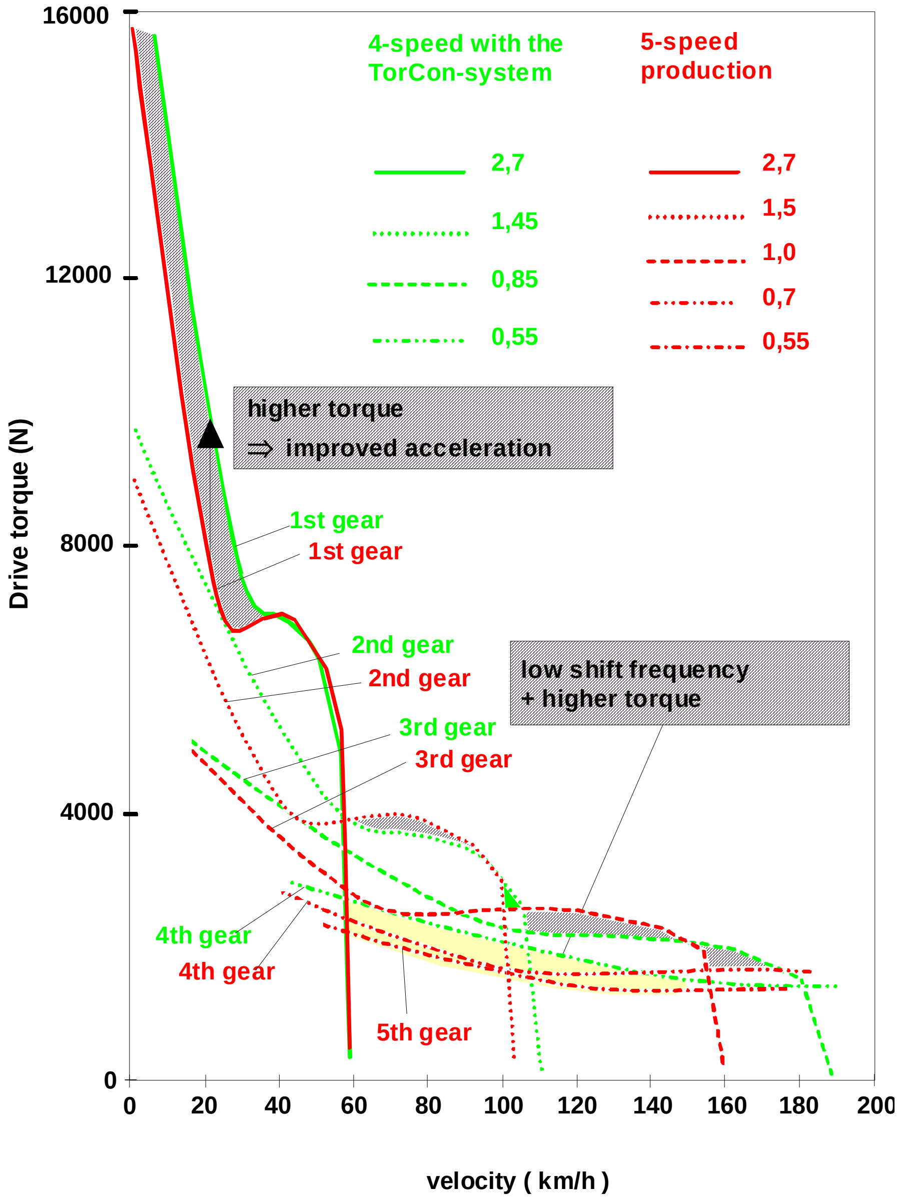


Figure 12: Comparison of tractive force for the LuK TorCon-4-speed System and a 5-speed transmission

CVT

The demand for an continuously variable transmission has been around for a long time. The hydraulic torque converter represents a compromise in this direction. However, its efficiency is not particularly satisfactory, and the selection of transmission ratios is not free - it depends on the limitations of the characteristic curves. The only way to modify the ratio is with a parallel bypass clutch like the LuK TorCon design that can modify the system within certain limits.

In mechanical continuously variable transmissions, the drive ratio is varied by modifying the friction radii of the load transmitting elements. There are several approaches to this solution; one of them will be demonstrated here.

LuK has been working together with other partners to develop a prototype continuously variable transmission designed for a torque of approximately 250 Nm (Figure 13).

The core of the design is the variator - a variable speed mechanism consisting a belt drive between tapered pulleys. In the first continuously variable transmissions introduced at DAF in 1959 this belt drive was made of rubber. Over a million transmissions were built based on this principle. Since then the belt drives have been made of metal in order to achieve higher output and to meet demands for higher product life. The most widely used design is the Van Doorne belt. Many prototypes and even production applications also operate with chains.

Because these transmissions utilize the friction between the tapered pulleys and the belt element to transmit power, high clamp loads are necessary because of the low coefficient of friction associated with steel-to-steel pairing. The transmission elements that are subjected to these loads (e.g., the pulley sets, shafts, etc.) must be extremely sturdy in design.

Currently familiar designs for continuously variable transmissions provides drive ratio ranges between five and six. This means that continuously variable transmissions make it possible to achieve transmission spreads that otherwise are only possible with five or size mechanical gears in an automatic transmission.

This wide range of transmission ratios can be used to take some of the load off the start-up element; the load exerted in this range is proportional to the square of the total transmission ratio in underdrive. It is also possible to use some of this wide range to develop an overdrive characteristic.

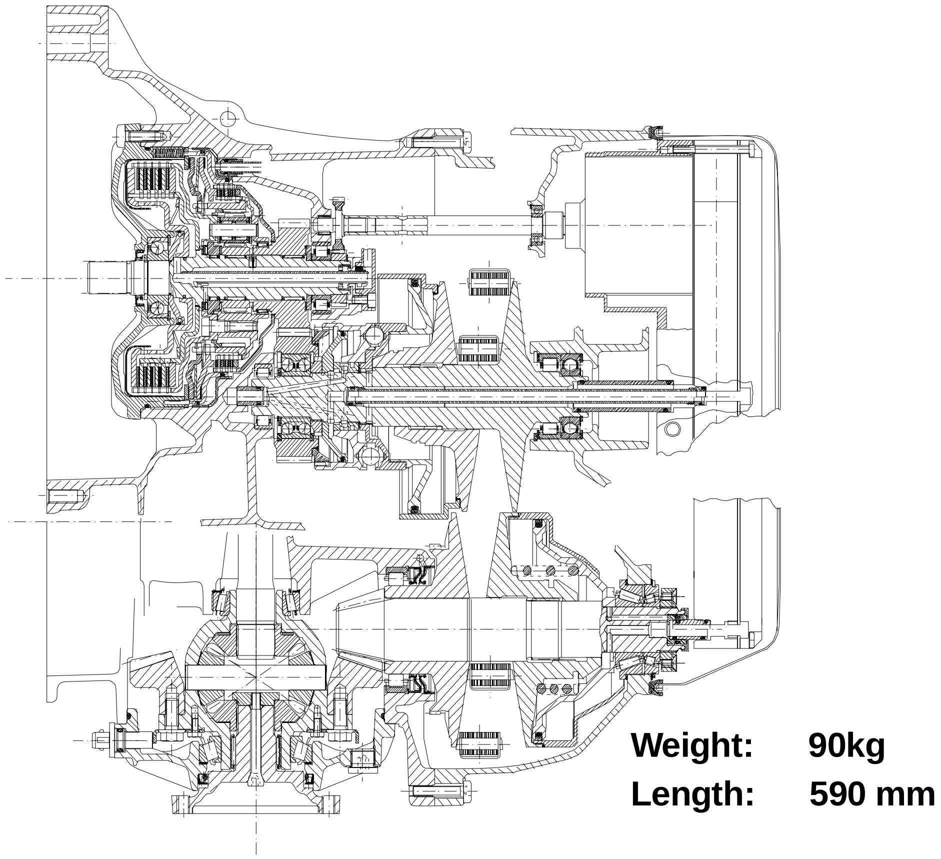


Figure 13: CVT-prototype (AUDI-LuK)

CVTs introduced in the past had relatively high losses. The high clamp loads exerted in the pulley sets required high oil pressures and large quantities of oil to facilitate the rapid adjustment of the tapered pulleys. This means that oil pump output must be considerably higher than for a multi-step automatic. The selection of the appropriate pump design is consequently very important.

A further source of loss is the so-called spiral circulation. The chain or belt is drawn increasingly toward the inside on the clamping points as it moves from the engagement to the output point, which results in additional losses. This can be prevented by designing stiffer pulleys and shafts [6].

The losses resulting from seal friction are actually greater than by conventional automatic transmissions because of the high pressures involved. However, there are fewer elements to service, and the seals can be reduced to very small diameters and positioned at the ends of the shafts, which reduces these losses.

Furthermore, the continuously variable transmission has fewer elements than the multi-ratio automatic, which means that drag losses due to open shifting elements and similar components are lower.

It is extremely important to maintain the tension of the pulley sets against the chain or belt. It is easy to visualize how excessive clamping can lead to friction losses between the pulley and the belt element, and to understand that pump output must increase because of higher pressure. An optimum design for a continuously variable transmission could achieve loss performance in the most important partial load range similar to values for a good 4-speed automatic (Figure 14).

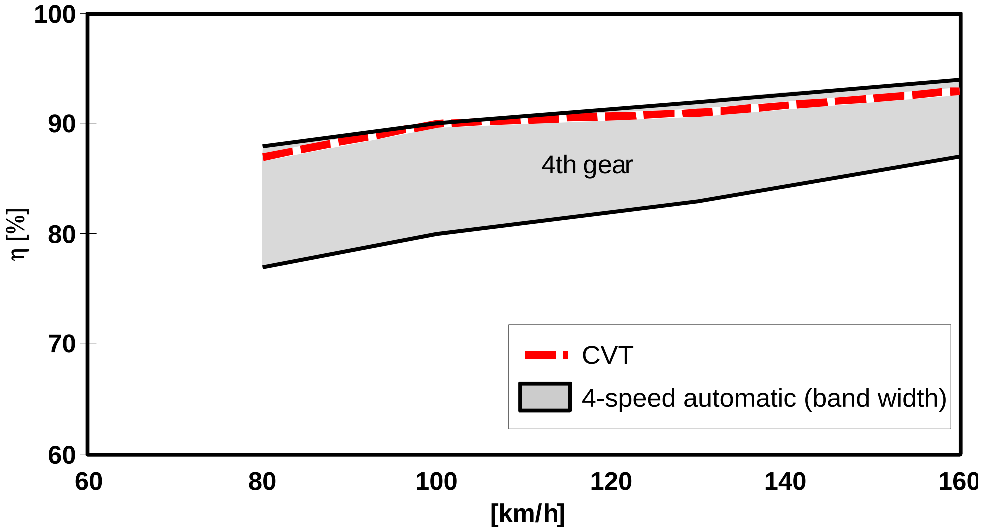


Figure 14: Comparison of partial load efficiency

Continuously variable transmissions have in the past exhibited relatively poor partial load efficiencies. In comparison to multi-ratio automatics, however, the larger total drive ratio range can be exploited to achived greater overdrive effect, which produces a more favorable engine operating point so that overall engine and transmission losses are reduced.

There have, however, been many complaints about operation in extreme overdrive because the vehicle makes a flaccid impression.

If the CVT has a very good partial load efficiency rating, designers can introduce the overdrive ratio wherever it is likely to be accepted. Because this CVT design achieves optimized drive ratio selection, it is possible to improve fuel consumption compared to manual transmissions that do not feature such precise fuel consumption control. Nevertheless, there is a tendency to overestimate CVT's advantages with respect to operation at optimum engine fuel economy. The actual fuel economy advantage lies in CVT's wide range of drive ratios and thus in its overdrive characteristic.

There are contradictions with respect to desirable objectives so far as the adjustment performance of the transmission is concerned. Uncompromising adherence to engine operation in the most advantageous range has often been criticized because of the unfamiliar acoustics of the engine speed curve in comparison to multi-ratio transmissions. On the other hand, there are complaints about changes in engine speed when the ratio changes in multi-ratio transmissions, even if the actual shifting process is imperceptible and in no way impairs driver comfort. Either a process of getting used to this shifting behaviour on a clever design compromise is required here. In the USA CVT-type shifting behavior in vehicles with multi-step automatic transmissions or "soft" torque converters has been completely accepted.

Because of the stiff pulley sets and shafts, total weight may be higher for CVTs than for comparable 4-speed automatic transmissions. There are, however, several options for optimizing the design. For instance, a start-up clutch instead of a torque converter ca be used if the CVT has been optimized for transmission losses and has a wide ratio range that can be exploited to increase the start-up ratio. This has significant weight advantages.

If one compares continuously variable transmissions with 5-speed automatic transmissions in the same performance class, it is conceivable that the weight advantage lies with the continuously variable transmission.

One also needs to compare design volume for multi-ratio automatics in the same performance range. In cases where the vehicle design dictates an axial displacement between the input and the output shaft, continuously variable belt or chain drive transmissions have design advantages.

When comparing manufacturing costs, it makes sense to limit the examination to parts that do not represent similar expenditures. If for instance, the design has an electronic control system, then the total cost for the oil pump and control system for the multi-ratio automatic and for the CVT will be similar. The transmission housing and oil pan can also be viewed as cost-neutral. In comparison to a 4-speed automatic, the forward/reverse shift element (1 planetary gearset and 2 clutches) for the CVT will amount to about 1/3 the costs of the counterpart unit for a 4 or 5-ratio automatic, assuming that the latter will use 5 clutches and 2 planetary gearsets. If the actual variator with its tapered pulleys and belt element will cost about as much to build as 3 clutches and 1 planetary gear set the total cost will be similar to a 4-speed automatic.

Whether a CVT requires more or fewer intermediate shafts between the transmission input and output shafts depends primarily on available space, specifically on the axle base. For instance, two shafts would cost less, while four would cost more.

As mentioned previously, it may be possible to elimimate a torque converter as the start-up element because of the wider ratio range.

Even if these calculations are relatively rough, nevertheless, it looks as if depending on the specifications the continuously variable transmission would be cheaper or would cost about the same to build as a 4-speed automatic, while it is highly probable that it would be less expensive than a 5-speed automatic.

At this point, the comfort of "gear changes" or changing transmission ratios probably don't need to be discussed. Especially with electronic controls, everything is possible within the physical parameters. Of course, when the engine speed increases, only part of the engine torque can be used for vehicle acceleration because part of the increase is lost for acceleration of the engine itself. As long as this limitation is accounted for, everything else is optional.

An across-the-board comparison

The total overdrive ratio is the primary determining factor for fuel consumption. On the other hand, a certain minimum first gear ratio or underdrive is necessary for good acceleration performance and trailer pulling capability. This minimum transmission ratio also contributes to a reduction of losses in the start-up element under difficult start-up conditions regardless of whether a clutch or a hydraulic torque converter is used.

	Manual trans.	4-speed AT	5-speed AT	4-speed LuK AT	CVT
Underdrive ratio	14.17	12.62	11.88	11.88	13.25
Overdrive ratio	2.94	3.27	2.30	2.30	2.21
Ratio spread	4.82	3.86	5.16	5.16	6.00

Table 1: Comparison of various drive concepts

In making this comparison neither the production design for the start-up ratio nor for the overdrive in particular has been changed, although in some cases a more highly developed overdrive design would be possible. In the case of manual transmissions, any ratio that would be readily identifiable as an overdrive gear would not be accepted by test drivers, and probably not by the market either.

In order to avoid compromises in tractive force in 1st gear of the 4-speed automatic with the ratio range shown here, the top gear had to be designed so that the maximum speed occurs approximately at maximum power. Shifting the overall design to lower total transmission ratios would result in decreased maximum speed.

For 5-speed automatic transmissions, the maximum vehicle speed is reached in 4th gear. 5th gear is used as overdrive, and first gear has an adequate tractive force.

In the 4-speed design with the LuK TorCon System, 1st gear and the top gear are the same as for the 5-speed transmission design, and the maximum vehicle speed is reached in 3rd gear.

For the CVT, part of the larger total drive ratio range is used for an even longer overdrive, and part of the ratio range is also used for a shorter underdrive ratio in order to be able to use a wet clutch as the start-up element, which has cost and weight advantages.

Test cycles

If a dual mass flywheel or DFC is used, the power train can be operated without booming or rattle noise all the way down to idle speed. This factor alone can reduce fuel consumption. Of course the driver will disengage the clutch at a certain point above idle speed in order to avoid killing the engine. This can be eliminated by installing an automatic clutch. If all the engagement and disengagement operations are automated, additional changes in the overall drive performance can be expected.

Changing driving performance will affect fuel consumption. Because official test cycles specify the shifting points, it is impossible to determine the affect of the automated clutch management system. For this reason, as noted in the previous presentation, a test route including city, overland and autobahn driving was established. This test route was driven by several persons in vehicles with and without automatic clutch management systems.

The cumulative frequency for the power used was similar to that for official cycles (Figure 15). The drivers with the highest and the lowest total output are shown as the extreme values. Because the actual power consumption figures are very low, the partial load efficiency ratings - for instance for street load (on a level road) are very important.

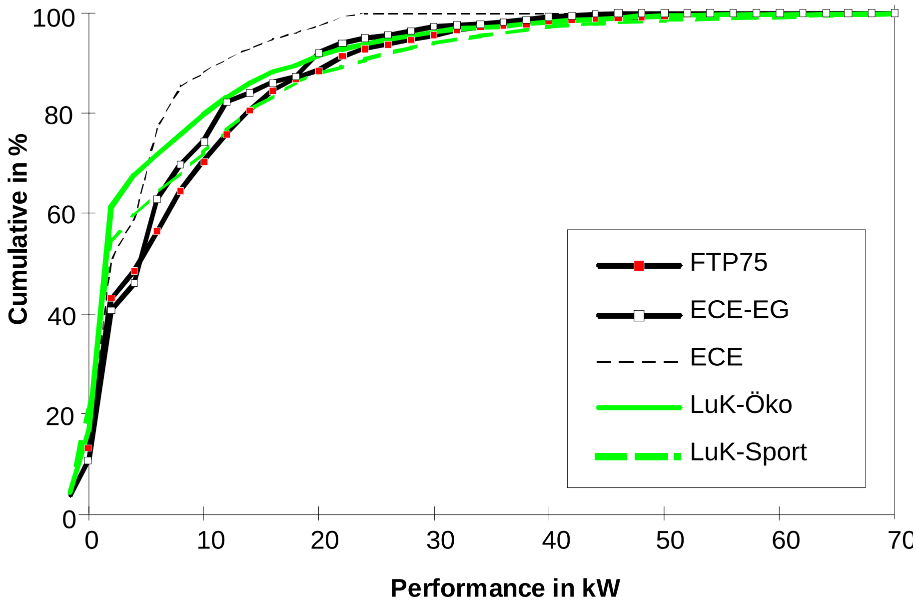


Figure 15: Cumulative frequency distribution for power

Evaluation of these figures produces some surprises. Almost all drivers achieved a higher average speed with the automatic clutch management system. On the one hand, this may be attributable to improved concentration on traffic; on the other hand, however, it may be attributable to frequent downshifting for acceleration. The inconvenience of downshifting with the accompanying loss in driver comfort, especially if the engagement of the clutch is not precisely carried out, frequently encourages many drivers to avoid using the higher engine performance in a lower gear. This may mean that with the automatic clutch management system it would be possible to take greater advantage of the overdrive characteristic in the top gear because the resistance to downshifting would be lower. This would produce greater fuel savings.

The differences in fuel consumption of the three test conditions are shown in figure 16. Because of the high average speed of the autobahn cycle, only transmissions with overdrive and low losses (e. g. 4-speed automatic with LuK TorCon System) have significant advantages compared to 5-speed manual. The reason for the similar fuel consumption of the CVT and the 5-speed manual lies in the strategy of the ratio control of the CVT. In country and city cycles all automated drive train versions are at least equal and sometimes considerably superior to the manual transmission.

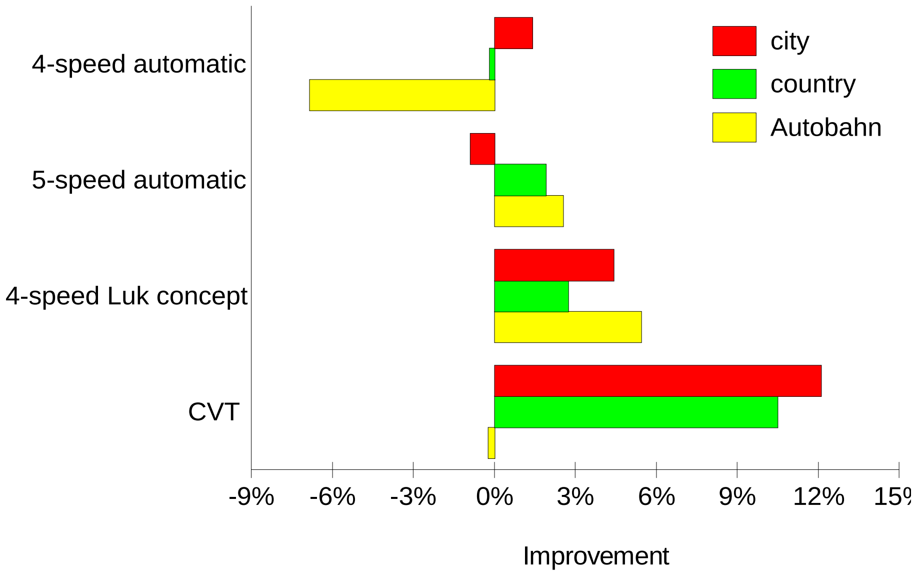


Figure 16: Comparison of fuel efficiency (estimate)

Driving performance

Acceleration from 0 to 100 km/h was used as a comparative bench mark for performance (figure 17). For acceleration under full load using foot-operated clutches, a typical start-up procedure (see figure 3) was chosen and the common "jack-rabbit start" avoided.

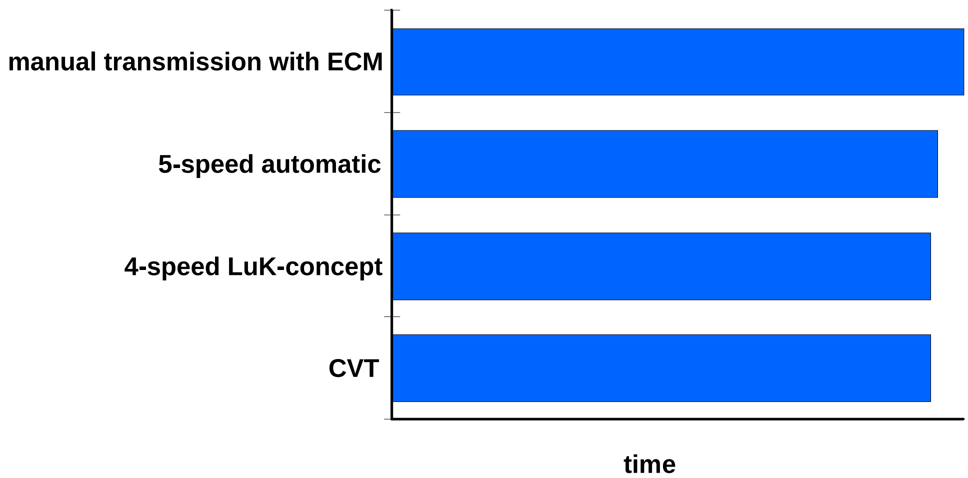


Figure 17: Comparision of acceleration

A so-called elasticity evaluation was not conducted because, in the case of automatics, this test is strongly dependent on the down-shift philosophy or, in the case of the continuously variable transmissions, on the variation philosophy.

All Consumption and performance data were determined using simulations. It was possible to compare most of the simulation calculations with measured vehicle data, so the values shown in the chart are reliable.

System size and weight

The various designs were compared based on the data cited in the previous chapters. Weight values were very carefully compared in an effort to select the most favorable designs currently on the market.

The 5-speed manual transmissions cannot be beat so far as size requirements and weight are concerned. The 4-speed automatic and the CVT come out about the same, but the 5-speed automatic takes its toll.

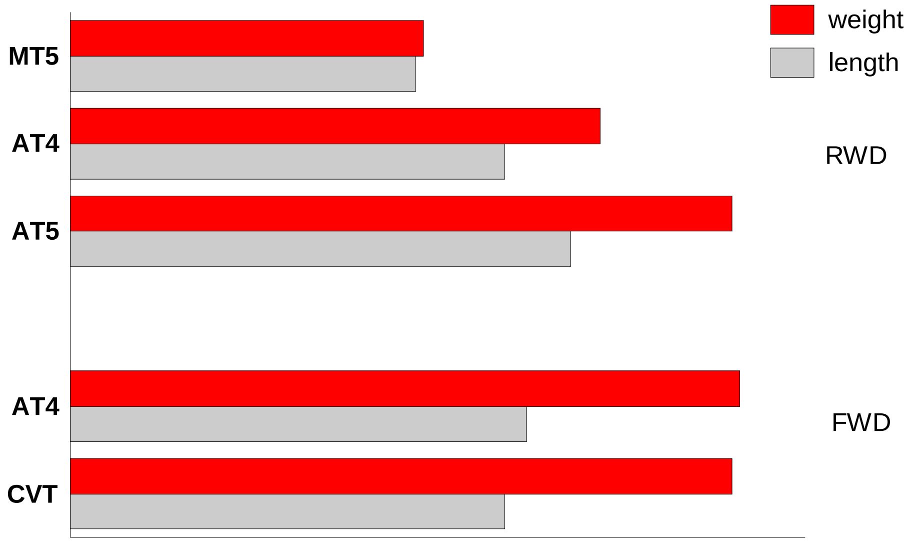


Figure 18: Comparison of length and weight

Of course, transmissions with similar features were compared, for instance, if the overrunning features were replaced with an electronic control system, space and weight were able to be saved, regardless of whether it was at a 4 or a 5-speed transmission. Nor was the so-called swapshift principle considered whereby a shiftpattern is used with the automatic to transform a 4-speed automatic to a 5 or 6-speed automatic with the same ratio range, but with a modified control system. A quantitative comparison between all transmissions must take into account that the CVT and the 4-speed layshaft automatic are designed for front-wheel drive and the integrated differential requires additional installation space and weight (figure 18).

Costs

Production quantity also plays a significant roll in manufacturing costs. It can even distort the picture. For instance, manual shift transmissions only make up 10% of the market in the US, so they can cost up to 100% more to manufacture than in Europe. Exactly the opposite is the case for automatic transmissions manufactured in Europe. Large American automotive plants produce as many units per day as some European automatic transmission plants do in a month!

Consequently, an automatic transmission manufactured in the USA can easily cost less than a manual transmission produced in small numbers. The continuously variable transmission in particular could be affected by manufacturing costs because current producion quantities are so low. An attempt to take these factors into consideration was made by postulating comparable piece numbers and development stages for all the transmission concepts. The use of electronic controls was assumed for all the designs. Figure 19 shows this comparison of manufacturing costs.

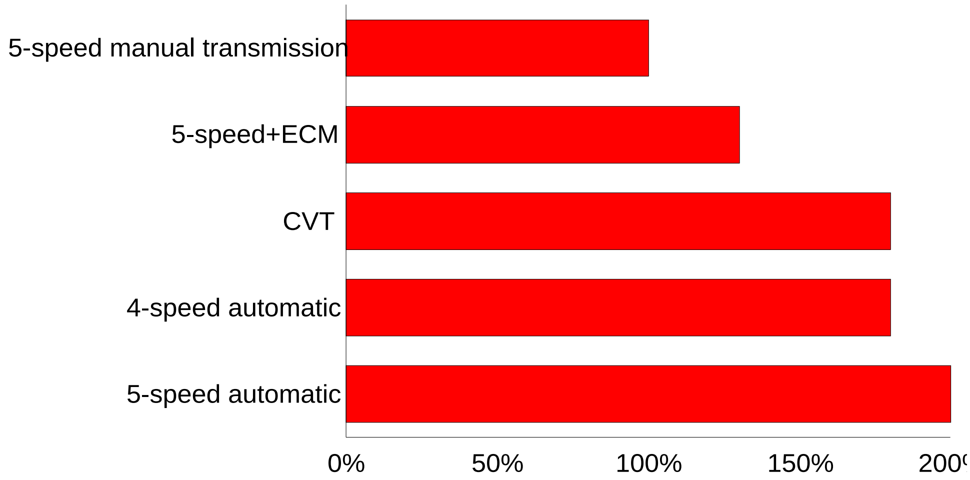


Figure 19: Manufacturing cost comparison (estimate)

Summary

Table 2 shows an attempt at a comparative evaluation. The 4-speed automatic transmission was viewed as a bench mark solution for an automatic power train.

	MT5	ECM	AT4	AT5	LuK AT4	CVT
Comfort	- -	-	0	-	+	+ +
Consumption	+	+	0	+	+ +	+ +
Emissions	-	-	0	0	+ +	+ +
Acceleration	+	+	0	+	+	+
Size	+ +	+ +	0	-	0	0
Weight	+ +	+ +	0	-	0	0
Cost	+ + +	+ +	0	-	0	0

Table 2: Comparison of various drive concepts

Comfort:

In the case of manual transmissions with foot-operated clutches, driving comfort during gear change and tip-in/back-out is almost entirely dependent on the driver. The electronic clutch management system can produce comfort improvements here if introduced with the appropriate strategies, but it cannot compete with the fully automatic transmission because of the interruption of tractive force during upshift operations.

The 5-speed automatic transmission was rated slightly less favorably for comfort in comparison to the 4-speed transmission because the higher shifting frequency very often leads to complaints. The 4-speed automatic design with the LuK TorCon System provides excellent comfort comparable to earlier automatic transmissions because of less frequent shift operations and the significantly looser torque converter. Of course, the continuously variable transmission provides the highest degree of comfort - anyone who drives for longer periods of time in a vehicle equipped with one of these transmissions will consider going back to a traditional transmission to be a step backwards.

Fuel Consumption:

Manual transmissions and automated clutches have an advantage in comparison to 4-speed automatics, especially if country roads and freeway driving make up a high percentage of overall driving. Because of the more advantageous gear selection, the 4-speed automatic may come out better in city traffic. Generally speaking, the 5-speed automatic has the advantage of an authentic overdrive. With the 4-speed automatic with the LuK TorCon System, losses can be reduced in comparison to the 5-speed automatic, which again results in improved consumption. A continuously variable transmission can achieve comparable consumption results.

Emissions:

Manual transmissions with automated clutches and all fully automatics provide the option of turning off the engine using a so-called start/stop function when the vehicle is stopped, which decreases emissions and noise. Many of today's four-speed automatics already utilize the option of engine/transmission management systems, particularly during the warm-up phase in order to force the catalytic converter to respond sooner and thus improve emission performance. The LuK TorCon System and the continuously variable transmission provide even better, more controlled operation, which makes it possible to achieve emission improvements of up to 30%.

Acceleration:

The electronic clutch management system allows even less experienced drivers to achieve good acceleration values. As shown in comparative testes more frequent downshifting, when necessary, utilizes existing engine performance. Based on the higher losses and lower ratio range, 4-speed automatics usually fair less well in this regard. 5-speed automatics, 4-speed automatics with LuK TorCon System, and the CVT can exhibit advantages even in comparison to manual transmissions. The higher losses involved are made up for by eliminating the interruption of tractive force during shifting or, in the case of the CVT, by the possibility of always utilizing maximum engine performance.

Size, weight, costs:

In this case, manual transmissions with electronic clutch management have the clear advantage. The 5-speed automatic has more design disadvantages than do the 4-speed automatic and the continuously variable transmission.

Based on the conditions analyzed above, in our opinion the primary trends in automated drive trains will include:

- 5-speed manual transmissions with automatic clutches
- 4-speed automatics with optimized torque converters
- Continuously variable transmissions

While the engineering principles for the first two options are well known and most of the existing investment can be applied to future designs without any problem, introduction of the continuously variable transmission will require investment changes in some areas. Nonetheless, the change will not be as great as is sometimes assumed because only the variator and the belt element differ significantly from conventional automatic transmissions. Electronic control systems in particular, have a great deal in common with automatics. It can be assumed that as demand grows to decrease vehicle fuel consumption and emission, a great future can be predicted for the continuously variable transmission.

References

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Theorie des Umschlingungsgetriebe mit keilförmigen Reibscheibenflanken