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Foreword

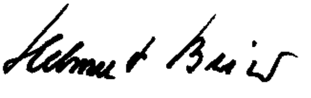
Innovations are shaping our future. Experts predict that there will be more changes in the fields of transmission, electronics and safety of vehicles over the next 15 years than there have been throughout the past 50 years. This drive for innovation is continually providing manufacturers and suppliers with new challenges and is set to significantly alter our world of mobility.

LuK is embracing these challenges. With a wealth of vision and engineering performance, our engineers are once again proving their innovative power.

This volume comprises papers from the 7th LuK Symposium and illustrates our view of technical developments.

We look forward to some interesting discussions with you.

Bühl, in April 2002



Helmut Beier

President
of the LuK Group

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The Parallel Shift Gearbox PSG

Twin Clutch Gearbox with Dry Clutches

Reinhard Berger
Rolf Meinhard
Carsten Bünder

Introduction

The Parallel Shift Gearbox (PSG) is a variant of the twin clutch gearbox.

Such gearboxes have been around for a long time. For example, at the end of the 1980s a Porsche twin clutch gearbox (PDK) was being used in motor racing [1]. However, it never reached production. Thanks to the search for transmission systems which minimise fuel consumption, twin clutch gearboxes are now experiencing a renaissance.

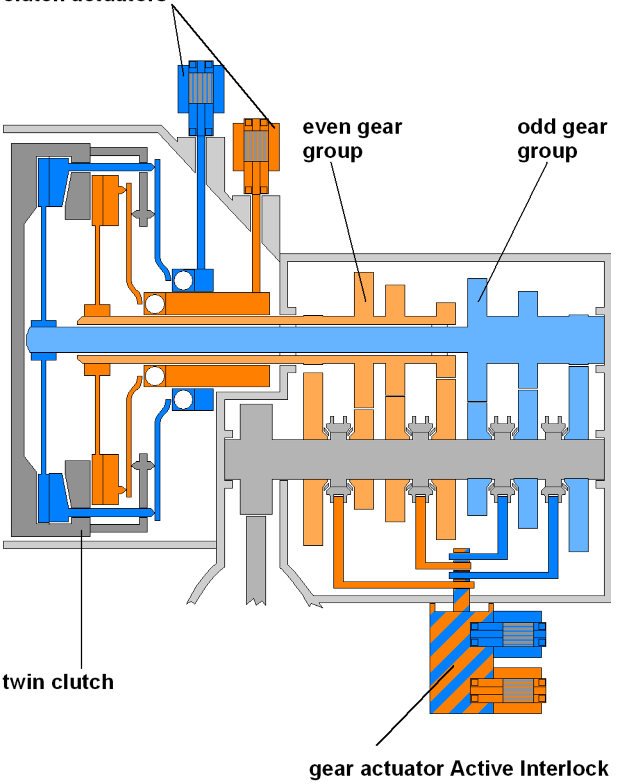


Fig. 1: PSG - Principle

As a member of the XSG family [2] the PSG has the following features:

- Dry clutches and electro-mechanical clutch actuators,
- Parallel shaft design and constant mesh gears
- synchronised dog clutches with electro-mechanical actuator and Active Interlock [4],
- system strategy elements in common with the ASG.

It is this combination of features that makes the PSG stand out from other twin clutch gearboxes.

In terms of functionality and comfort, the requirements of a PSG and its components correspond to those also placed on the automatic transmission designs of today.

The logical integration of the PSG in the XSG family is therefore particularly important, since a whole string of gearbox manufacturers are working on modular gearbox families, where, in addition to the manual gearbox, various automated versions are also produced using the same manufacturing equipment. LuK extends this modular approach to its components, clutches and actuators. Hence, for example, the same gear actuator can be used for an ASG or a PSG in conjunction with the Active Interlock.

This paper shows the opportunities available if dry clutches are used in a twin clutch gearbox, but also highlights the issues still to be resolved. It further demonstrates the integration of the individual components presented in other papers [2], [3], [4], [5] into the entire system. The subject of vibration isolation using slip control also plays an important part here.

Opportunities and Key Issues for the Dry Clutch in the PSG

Overview

LuK manufactures about 12 million dry clutches and about 3 million lock-up clutches for torque converters annually. Hence it possesses a wealth of know-how of both variants – dry and wet. This also applies to automation, since, in addition to ASG applications with dry clutches, LuK also played quite a crucial part in the development of the wet starting clutch for Audi’s multitronic®.

The following explains why the use of a dry twin clutch is preferred in the PSG:

- Lower fuel consumption,
- Option for modular gearbox families (MT, ASG, PSG),
- More frequently used as start-up element.

This has raised various concerns which crop up repeatedly whenever there is talk of replacing conventional automatic transmissions with gearboxes with dry clutches. More specifically, these are:

- Lining wear,
- Thermal capacity and robustness against misuse
- Controllability,
- Behaviour in the event of a system failure.

LuK will today demonstrate that all these issues can be satisfactorily resolved. In some cases, there are even advantages of using the dry clutch in comparison to the wet clutch -see figure 2. The most important criteria are now covered in detail.

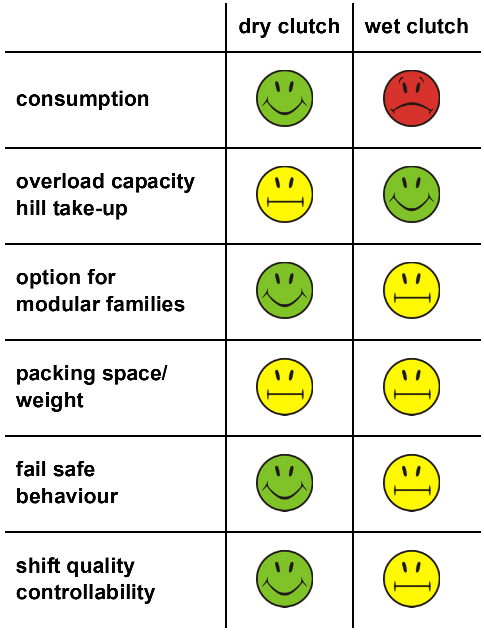


Fig. 2: Dry or Wet Clutch

Gearbox Power Loss and Fuel Consumption

With more than 96%, the manual gearbox (MT) has an outstanding full-load efficiency. However, in actual driving cycles with a high part-load content, friction losses in gears and bearings as well as lubrication losses, have already reached double figures [6]. Figure 3 illustrates the additional losses that occur in the clutches and for the gearbox control (whilst the base loss figures mentioned above are considered equal across all variants and consequently not shown).

These values describe the rates of power loss in relation to the energy induced in the combustion engine. Convertion into consumption values can result in slight discrepancies due to the engine consumption mapping.

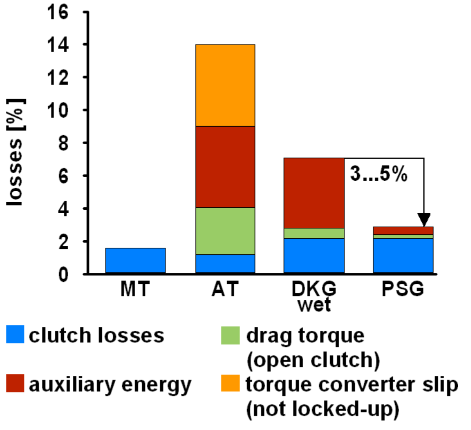


Fig. 3: *Clutch Losses and Control Auxiliary Energy*

Friction losses occur in the clutches when starting up and changing gear. This only affects the gear shifts for automatic transmissions, since start up is via the torque converter and not the clutches (start-up energy AT is included in the orange bar). As a result of the power shifts on the PSG and wet twin clutch gearbox (hereafter, 'DKG'), the clutch losses prove to be approximately 0.6% higher compared with the manual gearbox in the NEDC cycle.

Further losses occur as a result of drag torque in the non-active clutches. This represents a distinct advantage for the dry clutch. The largest slip losses occur in the automatic transmission (AT) of conventional design, since three or more load-free clutches or brakes with several discs are dragged along constantly [7]. On the PSG and wet DKG the load-free clutch runs without drag losses during driving, if no gear is preselected.

A dry clutch system with an electro-mechanical actuator is also the optimum solution when auxiliary power is required for the actuation of clutches and gear shifting. The average electrical power requirement is estimated at 20 watts; on the current ASG in production it is 12 watts. The oil pump, which runs permanently, the leakage of pressurised oil, the oil flow through the radiator and the filter all cause considerably greater losses on the AT or wet DKG. Since there is a constant engagement downforce on the clutch(es) during driving, an electrical oil pump would not improve the situation.

Given the automotive industry's voluntary commitment to reduce fleet consumption by 2008 to 140 g CO_2 per kilometre, the alternative concept with the dry clutches is the best possible solution.

Service Life of Dry Clutches

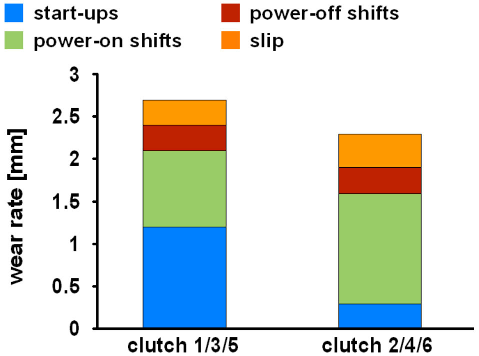
Since the introduction of asbestos-free clutch linings and self-adjusting clutches with increased wear life, the replacement of the clutch within the normal life of a car is practically never wear-related. With an automated clutch the average service life is even further increased. This is not only due to the greater consistency in clutch actuation, but also to the strategies implemented against misuse.

In contrast to that, one question is justifiably raised regarding the PSG: how does a dry clutch sustain the additional energy input following the power shifts? After all, a clutch must firstly take over the full engine torque and then reduce the speed difference. This creates three times more fictional energy than in ASG gearshifts.

A mixed driving cycle has been developed at LuK to estimate the service life of dry clutches, which has gained approval from our customers with field results and endurance tests. This cycle contains 50% urban driving, 30% rural driving, 18% motorway driving and 2% hill driving. In total, this cycle includes 240 000 journeys and 1.5 million gear shifts at a required vehicle endurance of 240 000 km.

Figure 4 shows the results of the wear extrapolation for a standard-size vehicle. It is clear from the figure that part of the start-up energy is also absorbed in the clutch of the second gear. Depending on the ratio configuration, it is possible on the one hand to start up directly in second gear (e.g. part-load operation) or to apply both clutches during start-up.

The results shown in figure 4 are based on an average wear rate that is 30% higher than today's manual gearbox. This is in order to take account of the properties of future lead-free linings and also to take into consideration the higher temperature of the component (higher energy input due to power shift).



vehicle	2000 kg
engine	400 Nm
clutches	diameter 250/220
life time	240 000 km

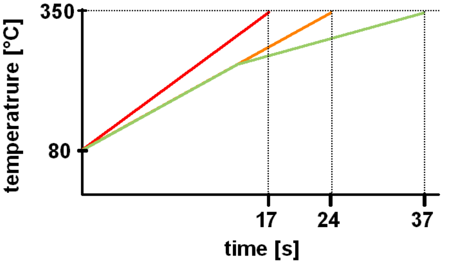
Fig. 4: *Service Life of Dry Twin Clutch*

Situations with a Very High Energy Input

Constant creeping, crawling, hill starts with and without a trailer, hillholding vehicle – most of the concerns in these situations are linked with the use of a dry clutch in an automatic transmission. Every vehicle manufacturer has his own tests for this, which, to some extent, focus on very different things.

By automating the dry clutch, situations with a very high energy input are better controlled than with a manual gearbox. This is reflected in the experiences with the Mercedes A-Class (ECM) and the Opel Corsa (ASG). The PSG again offers even more potential than an ASG, more specifically, for the following reasons:

- A gearbox that is 100% fully automated offers more freedom in the ratio configuration of the first gear than an add-on ASG. A higher ratio (first and possibly second gear) does not automatically lead to worse cycle consumption, but drastically reduces frictional power during start-up or creeping.
- On the PSG the frictional heat can be distributed over two clutches and their cast masses respectively.



case 1: 1st gear as per manual gearbox
case 2: 1st gear 20% shorter
case 3: start-ups with both clutches

Fig. 5: *Stopping on a Hill – Temperature Simulation*

Figure 5 compares temperature simulations for 'stopping on a hill' for different options.

It can be seen that the time for which a PSG vehicle can be held on a hill without destroying the clutch (Case 3), is almost twice as long as with a manual gearbox (Case 1). This is important, for example, in situations such as exiting a car park, when it is necessary to wait on a slope and give way to other vehicles.

From a certain point, however, the self-protecting mechanisms of the automated clutch have to intervene, even on the PSG [5]. Possible solutions for that are shown in figure 6.

A decision is therefore required as to whether the driver depresses the accelerator and thus explicitly expresses a desire to accelerate, or whether he merely lets go of the brake pedal and a small creeping torque builds up in the clutch. In the first case the clutch could be gradually engaged after some time (a few seconds, see figure 5), in the other case the clutch would have to open, see figure 6. During continuous creeping however, the period prior to the intervention of self-protection lasts several minutes due to the low torque.

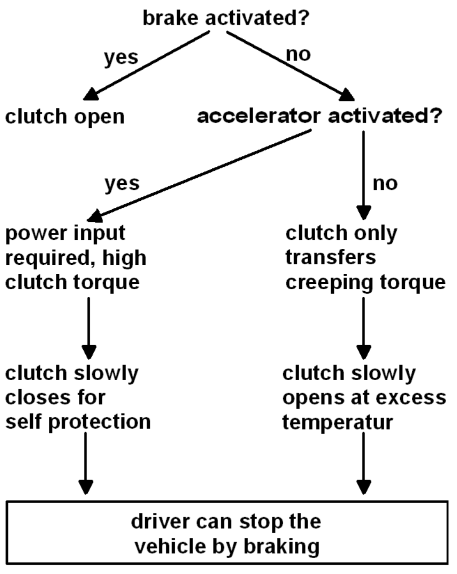


Fig. 6: *Hillholding – Possible Clutch Protection Strategies*

Measurements of clutch self-protection strategies are also shown in [5]. These strategies have been reviewed very positively in several prototype cars.

Nevertheless, even better solutions are envisaged for the future, namely when the gearbox control can request hill holding from an active braking system.

Limp-Home Behaviour

This section looks at the limp-home behaviour during a total failure of the gearbox control. From a gearbox control point of view, this is certainly the most serious scenario. Just as with every partial failure, it is essential that it should never reach a safety-critical condition, i.e. there is a direct risk of accident and danger to life and limb.

As the response to processor errors shows [5], it also applies here that maintaining the current status of both clutches at the moment of the failure is the best possible direct response, i.e. the clutches should neither open nor close, see figure 7.

With such limp-home behaviour, the risk of an accident at red traffic lights or on a pedestrian crossing, for example, can be avoided, because the clutches in these situations are open and remain open. Even when driving this limp home behaviour would take account of the current driver command and thus avoid dangerous situations, because the clutches can not disengage suddenly. For example, this is important during close overtaking or hill driving. On the PSG the driver of the vehicle could still accelerate, if the complete failure of the gearbox control occurred during an overlapping gear shift.

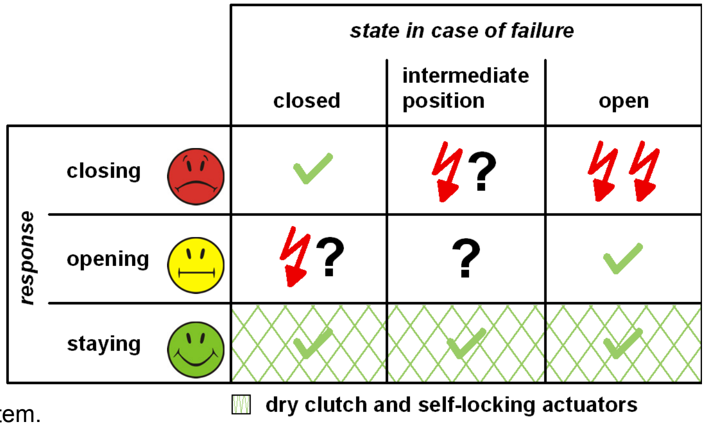


Fig. 7: *Safe Condition during Total Failure*

Dry clutches and self-locking electro-motorical actuators demonstrate such behaviour after the power output stages are switched off. On the other hand, additional mechanical effort would have to be used on wet clutches to freeze the condition of the clutches.

If the customer requested, LuK could also offer a system with opening clutches as the failure mode. An electromotorical actuator, hydrostatic release system incorporating a valve which opened in case of failure, together with normally open clutches, would give such a system.

If gearbox control and actuation fail in the middle of an overlapping gear shift, the question is raised as to whether this could lead to the gearbox locking. After all, a gear is engaged in each section and both clutches are transferring torque. It must firstly be stressed here that both clutches in total only transfer the engine torque, i.e. a clutch is simultaneously closed and the other opened during an overlapping gear shift. Thanks to the known adaptations for the clutch characteristic curve (touchpoint, gradient) [9] and due to the torque tracking or slip control, this can be very precisely adjusted.

In the event of limp-home operation, a clutch would now quite simply slip, but the gearbox wouldn't lock at all. The output torque can be calculated using the following formula:

$$M_{ab} = i_1 \cdot \left[M_{mot} + \frac{i_2 - i_1}{i_1} \cdot M_{K2} \cdot sign(n_{mot} - n_2) \right]$$

M_{ab} Output torque

M_{mot} Engine torque

M_{K2} Torque of slipping clutch

n_{mot} Engine speed

n_2 Speed – input shaft with slipping clutch

i_1 Ratio of gear with sticking clutch

i_2 Ratio of gear with slipping clutch

This equation shows that the output torque is determined by the engine torque, the torque of the slipping clutch and the gear step. Even during a failure of gearbox control the driver can still influence acceleration with the engine. Figure 8 shows the power fluxes for the scenario where the driver wants to accelerate further.

Only if the driver takes his foot off the accelerator would the deceleration be slightly higher than expected, figure 9.

The danger of the wheels losing their grip here only exists if everything occurs together (failure of both clutch actuators during an overlapping and driver taking his foot off the accelerator) on a smooth surface. With such road conditions, however, a power on/off load condition can lead to the wheels slipping on any gearbox system, ever without an error existing.

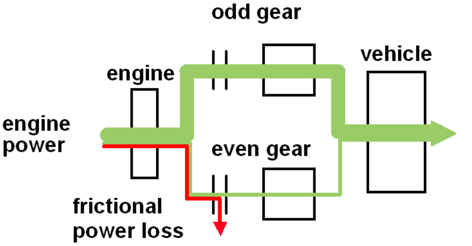


Fig. 8: *Power Flows during Acceleration after Gearbox Control Failure during Overlapping*

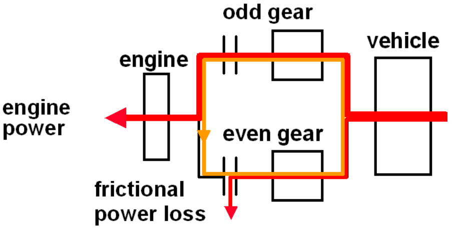


Fig. 9: *Power Flows when Coasting after Gearbox Control Failure during Overlapping*

This is clear from figure 10. In the acceleration time curves the largest deceleration occurs after the power on/off load condition in first gear. The deceleration of a PSG is only greater in the steady-state, if the gearbox has failed and the clutches were previously frozen during a gear shift.

An engine drag torque-control can prevent the wheels slipping in both cases. Figure 10 shows an example of this for a power on/off load condition on which the engine goes to zero torque.

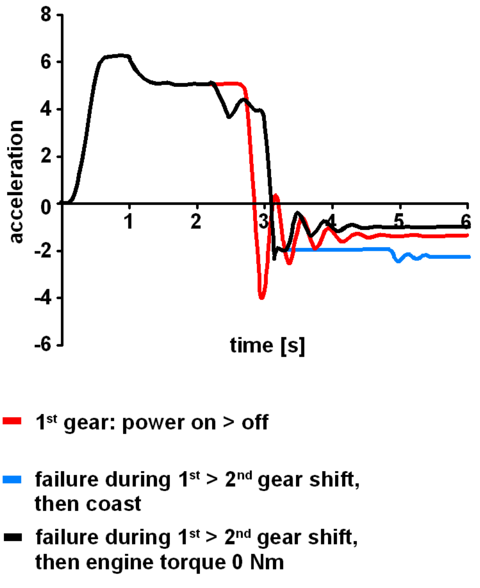


Fig. 10: Power On/Off Load Condition: Acceleration Time Curves

The reliability of gearbox control of a PSG compared with an ASG can be increased with marginal additional vehicle wiring.

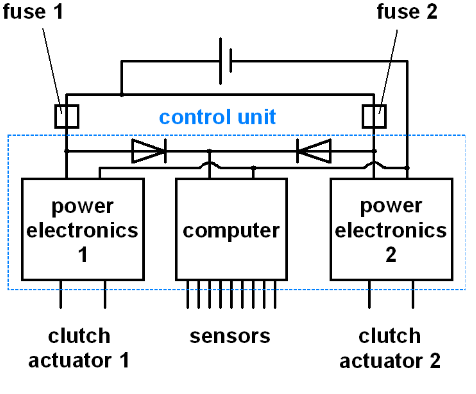


Fig. 11: Redundant Protection of the PSG-Gearbox Control

The individual clutch actuators are protected separately, so that during a short circuit in a clutch actuator, for example, the gearbox control and the other clutch actuator are still supported, figure 11.

The likelihood the failure scenario previously considered, occuring (i.e. that both clutches transfer torque and can no longer be modulated) is thus minimised and, in fact, requires a double failure to occur at all.

Controllability

A torque travel characteristic curve applies to the controllability of a dry clutch. On a wet clutch, a combination of torque-volume-characteristic curve and torque-pressure-characteristic curve describes the transfer behaviour, figure 12. This combination results from the fact that the wet clutch must firstly be pre-filled before an engagement force can be adjusted via the pressure.

Dry and wet clutches have one thing in common: the clutch torque transferred depends quite considerably on the current friction co-efficient and can effect both systems. The changes in friction co-efficient of both systems show the following peculiarities:

- The friction co-efficient of a dry clutch can change more drastically than on the wet clutch, but can however be successfully compensated for with intelligent adaptational algorithms [9].

- The dry clutch usually recovers over the service life, even if it has drastically altered its characteristic curve temporarily after a high input of energy. With a wet clutch it leads to irreversible oil ageing, which is expressed by a negative friction co-efficient gradient, figure 13. The original friction co-efficient characteristics can be restored by changing the oil.

These statements are backed up by experience with the ASG. For example, a programme exists at LuK for objectively evaluating shift quality using measured acceleration characteristic curves. This is used for the regular assessment of vehicles in an endurance test. The results illustrated in figure 14 show that the shift quality of an ASG with a dry clutch scarcely changes over the service life.

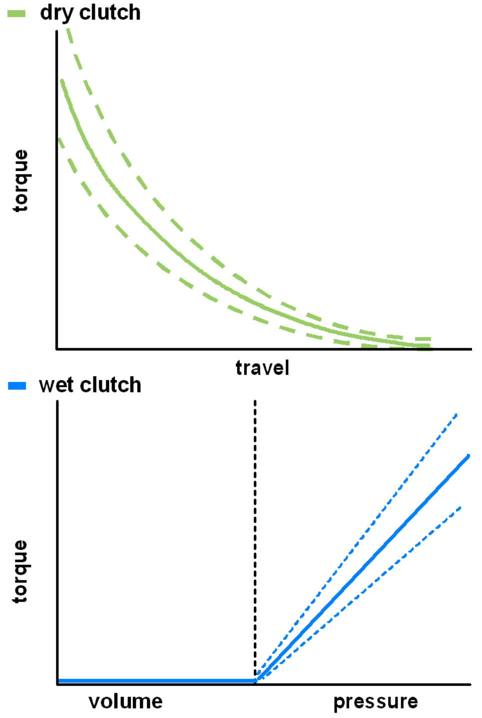


Fig. 12: *Characteristic Curves*

Fig. 13: *Wet Clutch: Oil Ageing*

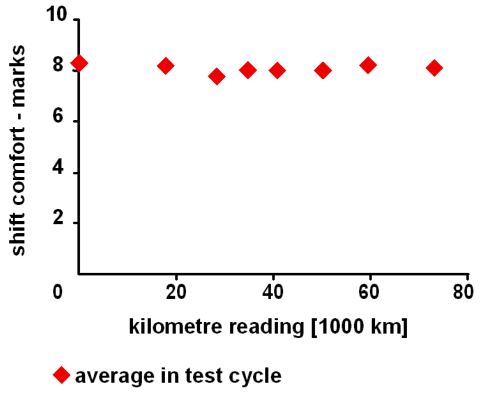


Fig. 14: *Dry Clutch: Constancy throughout Service Life*

Design Examples

Dry Twin Clutch and Actuation

Having dealt with the functional aspects of using a dry clutch in the PSG, design layouts are now considered. The twin clutch, flywheel, clutch release systems and clutch actuators must be configured in the assembly so that optimum use is made of the limited packaging space. All components should be located in the clutch bell housing or on the gearbox. This results in two particular challenges:

- The axial packaging space in the clutch bell housing (especially on transverse applications),
- the package around the gearbox for those actuator components which are not located in the clutch bell housing.

Figure 15 shows a twin clutch with DMFW on the left; on the right is a twin clutch with damper clutch discs and a flex plate. The latter variant has considerable advantages for an axial packaging space (about 20 mm) and inertia (about 0.1 kgm²). The following section shows how it should not result in any disadvantages with respect to vibrational isolation.

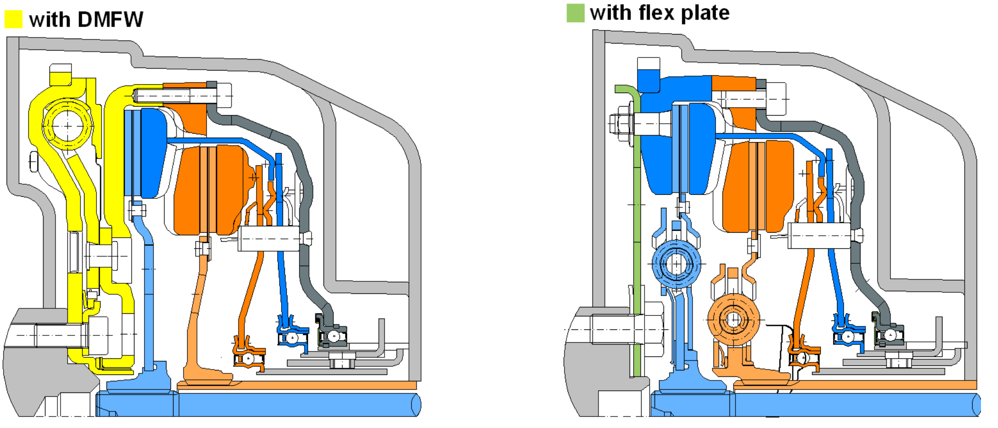


Fig. 15: Twin Clutch Design Examples

Various options for the release system and clutch actuator are the subject of [3] and [4]. The design examples in figure 15 contain a double ramp mechanism.

Figure 16 shows a design example for electro-mechanical actuators which actuate this double ramp mechanism.

The cover mounted connection of the release system on the clutch via a support bearing should also be emphasised. This has the following advantages:

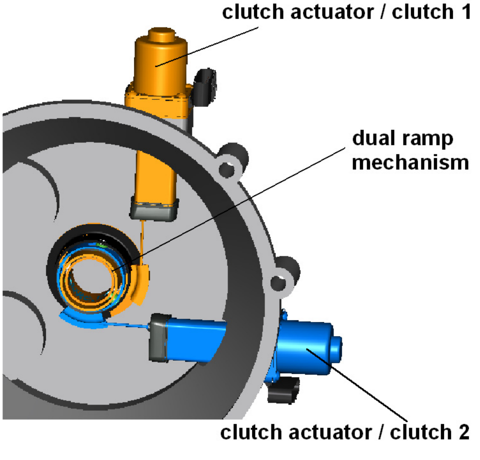


Fig. 16: Release System Twin Clutch

- Reduction of the tolerances to be considered in the release system,
- radial support of the clutch mass in the gearbox housing,
- actuating forces do not affect the crankshaft, but are supported internally.

The clutch and release system form a unit and belong to the gearbox. Hence the entire assembly and the self calibration of the complete PSG can be done in the gearbox plant.

Continuous Slip for Vibration Isolation

The usual low noise comfort factor from the DMFW in the vehicle can also be achieved with the more favourable, compact twin clutch variant with damper discs and flex plate. For this reason, the clutch in the vibration-sensitive operating ranges of the closed transmission system is operated with continuous slip. Information about the control strategy is presented in [5].

The torsional vibration dampers in the clutch discs guarantee that driving without slip in wide speed ranges is possible [8].

In the operating ranges in which there are noise problems with a closed transmission system, the specific slip offers partially better isolation than a DMFW.

With power on/off load conditions, both from tip-in to back out and vice-versa, vibration excitations in the transmission system can be avoided by using slip control [9].

From the dynamics point of view, the variant without DMFW offers equal advantages. All things considered, the inertia of the twin clutch is approximately as great as the entire inertia of a DMFW and a single clutch, i.e. in terms of the inertia there is scarcely any difference between the manual transmission or ASG and the PSG. Hence the accelerating performance and the engine dynamics are comparable. On a twin clutch with DMFW, the primary mass, which is necessary for stabilising the crankshaft, would still have to be added as an additional inertia. This lies within a range of 0.1 to 0.13 kgm^2, see figure 17.

	without DMFW	with DMFW
MT / ASG	0,2 kgm^2	0,25 kgm^2
PSG	0,25 kgm^2	0,35 kgm^2

Fig. 17: *Inertia Torques (inc. Crankshaft)*

The primary flywheel of the combustion engine on the solution with the flex plate is considerably greater than on the DMFW solution, hence the crankshaft runs more quietly anyhow. This means that the excitations on the engine side are reduced on a PSG with dry twin clutch, flex plate and damper clutch discs, compared with the other gearbox variants, also when compared with a manual transmission, see figure 18.

Simulation calculations and tests have shown that continuous slip in certain ranges for vibrational isolation does not lead to increased fuel consumption and only causes slight extra wear of the clutch linings.

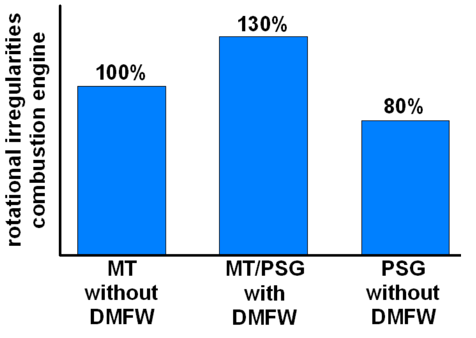


Fig. 18: *Rotational Irregularities*

At first sight, it may seem implausible that additional clutch slip does not increase fuel consumption. This is due to the lower inertia of the unit compared with the DMFW solution. The additional primary inertia of the DMFW must also be accelerated in all acceleration operations, so that non-usable kinetic energy is stored in it. With a primary mass of 0.1 kgm^2, it corresponds to additional consumption of about 0.5% in NEDC cycle. Losses were calculated in the same magnitude for continuous slip in certain ranges, hence the saving due to lower inertia and the loss due to slip compensate each other, see figure 19.

	increased consumption	
	manual mode	**automatic mode**
optimised engine operating point	± 0%	-5,0%
reduction of inertia	-0,5%	-0,5%
slip control	+0,5%	+1,0%
total	± 0%	-4,5%

***typical average value**

Fig. 19: *Consumption Results PSG with/without Slip Control*

The slight extra wear of up to 0.4 mm per clutch was previously shown in figure 4.

Active Interlock for Gearbox Actuation

With the Active Interlock system [4] developed by LuK, the effort for automated shift actuation in the gearbox is reduced to a minimum. The components of the internal shift mechanism (shift forks, shift rails, guide parts) correspond to that of a manual transmission or ASG, figure 20. No additional gear actuator is required for the second section of the PSG (figure 1). It is only neccessary to modify the ASG gear actuator locking elements in order to be able to use it for shifting both PSG sections.

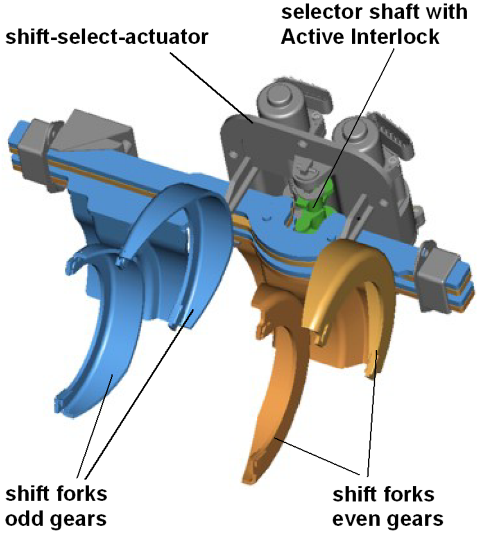


Fig. 20: Active Interlock Shift System for PSG

For gearbox manufacturers who derive the PSG modularly from a manual transmission family, the Active Interlock shift system is particularly advantageous due to its great similarity to the manual shift mechanism/ASG. Even the flange for the installation of the gear actuator can be designed exactly like the flange for the shift mechanism on the manual transmission module [4].

It is possible to do without the second actuator if the shift slot in the shiftrails is wider than the shift finger which operates the shift rails via this shift slot. Hence two gears can be simultaneously engaged in the PSG, i.e. an even and odd gear at the same time see figure 21.

The Active Interlock system also ensures that neither two odd, nor two even, gears can ever be engaged simultaneously [4].

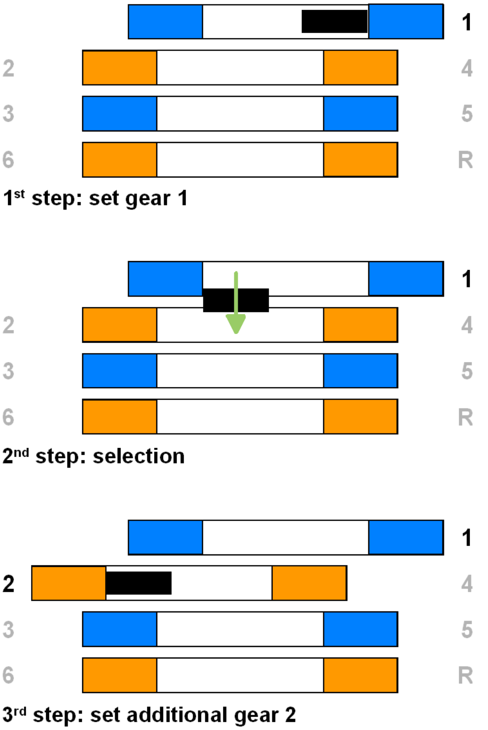


Fig. 21: Shifting Two Gears with Active Interlock

Integration of Parking Lock

The PSG should not only be a fully adequate replacement for today’s automatic transmission from the functionality point of view, but also in terms of operability. This now leads to the demand for an integrated parking lock. In the future this function can probably be achieved with electrical braking systems.

In current projects, LuK has investigated the possibilities of not only using the advantages of the shift-by-wire system (noise decoupling

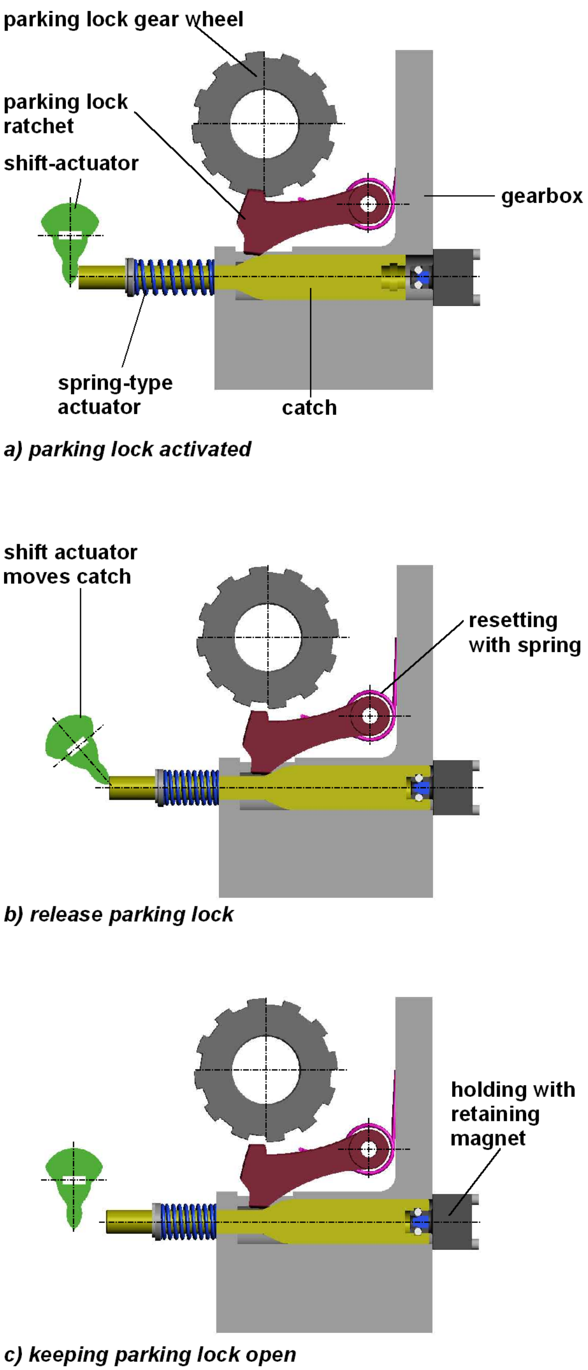


Fig. 22: Example of Design with Integrated Parking Lock

to the vehicle interior, freedom for interior design), but also developing parking lock solutions with minimum additional effort, without resorting back to the braking system.

The most simple solution appears to be to engage a gear in each section of the gearbox and close both clutches. However, the starting of the combustion engine with the selector lever in P position in this condition would not be possible, because the clutches would have to be open and hence the park lock lifted. This would contradict the state-of-the-art in terms of automatic transmissions with a parking lock.

As can be seen from today's automatic transmissions, the mechanical parking lock ratchet is a relatively simple component. The parking lock ratchet could also be operated by the electro-mechanical gear actuator of the PSG. This suggests the following ideas:

- The parking lock ratchet is actively engaged and disengaged through the gear actuator.
- The engagement of the parking lock ratchet is via a spring-type actuator, whilst the release is actively done by the gear actuator. During driving the parking lock is held open by a retaining magnet.

Both solutions are practical according to current state of the art technology. At the end of the day, the design must be adjusted in collaboration with the vehicle manufacturer to the preferred operating concept. LuK's concern is to save it's customers extra effort by considering the option of the parking lock operation when designing the gear actuator.

Figure 22 shows an example of a design for releasing the parking lock with the electro-mechanical gear actuator.

Summary and Outlook

The Parallel Shift Gearbox is an automatic transmission with a dry twin clutch. It is part of the XSG family and builds on the know-how that LuK has gained with the dry clutch and its automation, together with the automation of manual transmissions. The main argument for the application of dry clutches is the greatest possible fuel saving, without simultaneously having to tolerate loss in driving comfort compared with conventional automatic transmission systems.

A first PSG functional sample is currently being built at LuK. Further exciting projects are planned with different vehicle and gearbox manufacturers. As it stands, the first series application is envisaged for model year 2006.

Whilst the ASG is more suitable for smaller or sporty vehicles and engines, the PSG can be used in all vehicle classes and engines.

With its know-how and its components, LuK hopes to contribute to this by offering the customer innovative, economic and competitive transmission systems.

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