Double clutch –

Wet or dry, that is the question

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Introduction

Double clutch transmissions for passenger vehicles are currently occupying the attention of development departments in the automotive and supplier industries. The driving force is the improvement in fuel consumption available from current manual transmissions together with the comfort of automatic transmissions. In order that the new generation of transmissions can function with the highly effective gearshift and synchronisation devices of manual transmissions, two clutches that can be independently operated, are required [1]. Each of the two clutches links one subtransmission to the internal combustion engine, a function that can be fulfilled in principle by wet or dry clutches. Both clutches must be operated by automated means in order to control the gearshift operations without interruptions to starting or traction force.

Which clutch system (wet or dry) represents the better solution for a generation of vehicles is currently the subject of intense discussion in the technical arena (figure 1). LuK has experience with both dry and wet clutches and therefore wishes to address this subject without the oft stated prejudices against each system, but without making any claim to completeness.

The opinions commonly expressed in the technical arena are presented below. The dry clutch has only limited thermal capacity, so that under large energy inputs the system quickly reaches its limits, which are significantly below those of comparable automatic torque converters or wet clutches. Furthermore, wear of the dry friction lining is often a point of discussion where questions of service life are concerned.

The wet clutch in combination with a fully hydraulic control system for actuation and cooling is generally regarded as too demanding and expensive. Furthermore, the pump losses often lead to higher fuel consumption compared to dry solutions.

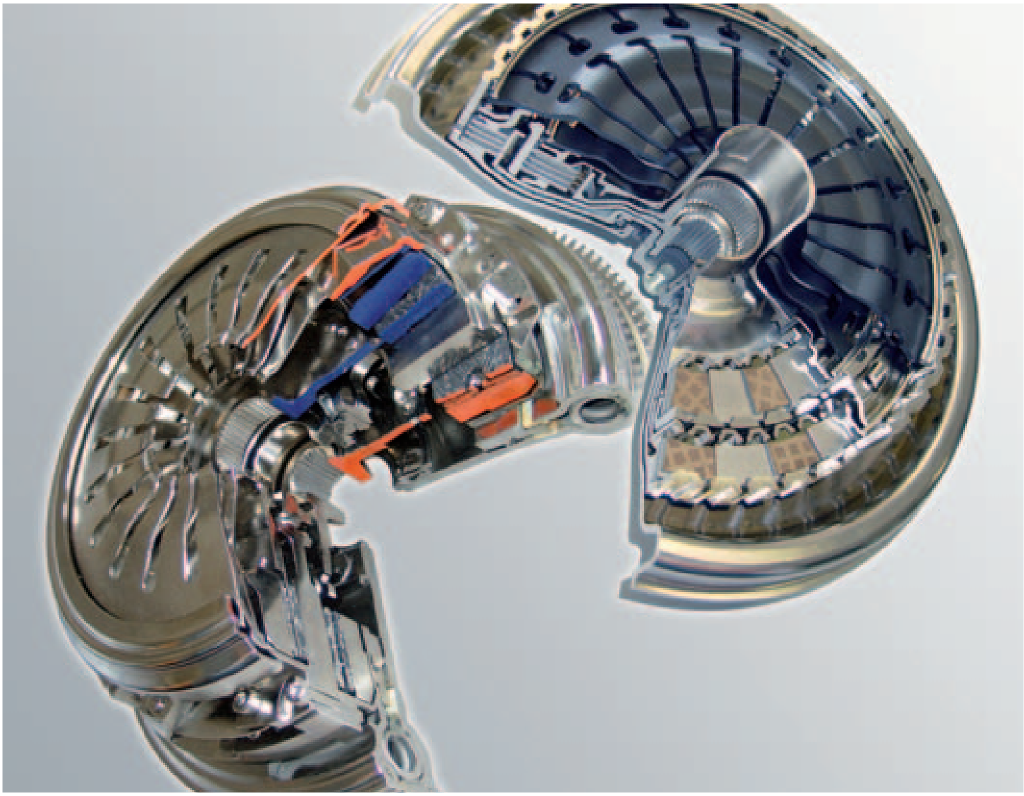

Figure 1 Dry and wet double clutch

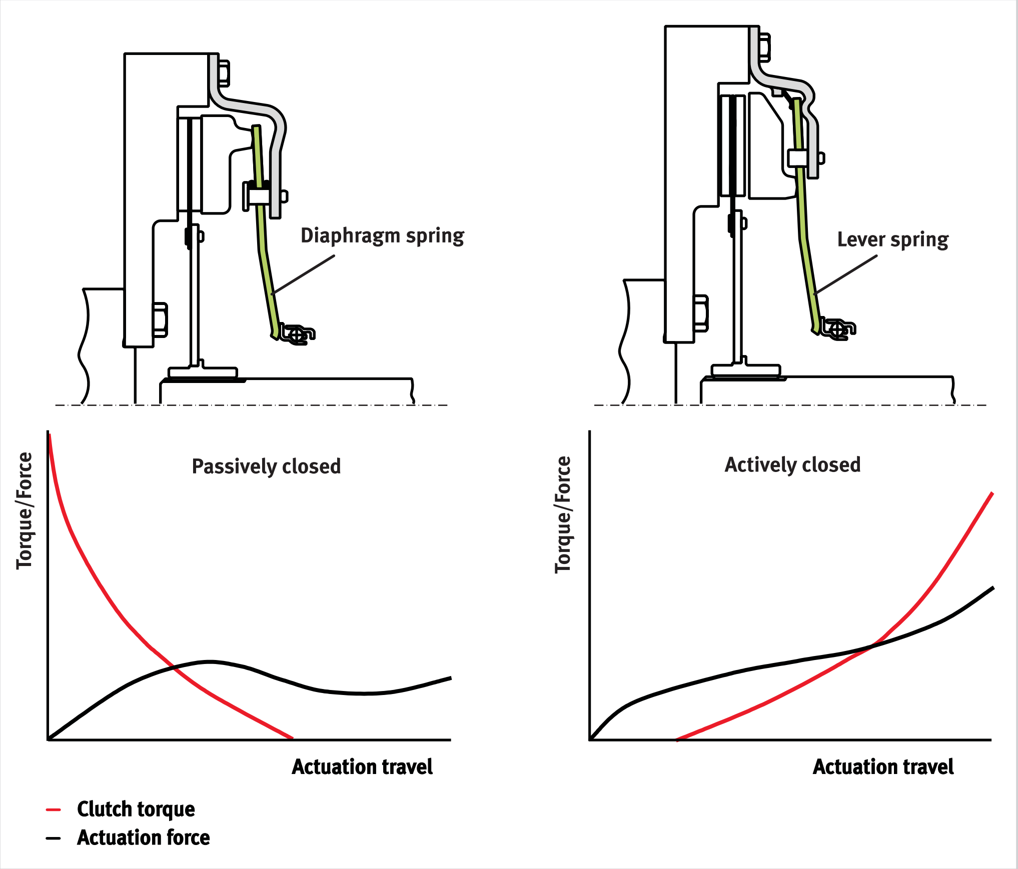

Figure 2 Comparison of passively closed / actively closed clutch

The dry double clutch

Due to the construction of powershift transmissions, safety reasons dictate that the clutches must open automatically if the clutch actuation system fails. This can be achieved very easily through the use of so-called "actively closed clutches". In actively closed clutches, the contact force is equal to zero if there is little or no force acting on the diaphragm spring fingers. In contrast, vehicles with manual transmissions have passively closed clutches in which the full contact force is present on the clutch linings if there is no force acting on the diaphragm spring fingers [2, 3]. In this condition, the maximum torque is transmitted. The left side of Figure 2 shows the cross-section of a passively closed clutch, while the right side shows the cross-section of an actively closed clutch. Since, in actively closed clutches, the diaphragm spring is used mainly as a lever to transmit the engage force to the contact plate, this is described as a lever spring. The particular requirement is that the lever spring fingers must be extremely rigid in an axial direction in order to minimise travel losses. The lever spring is also designed such that, throughout the working range of the engage bearing, there is always low return force and so secure opening of the clutch is ensured.

Arrangement of the clutches in the transmission

Clutches for manual transmissions are normally mounted directly via the flywheel on the crankshaft. The release force required for actuation is supported in most cases via the flywheel. Since double clutches require significantly more space in an axial direction and the actuation forces in certain travel conditions are higher than with manual clutches, direct linkage and bearing support on the crankshaft is not feasible in many cases due to the excessively high load. An alternative arrangement is to support the double clutch on one of the two transmission shafts, preferably on the hollow shaft. In this arrangement there are, in principle, two possibilities for linking the double clutch to the crankshaft.

In variant 1 with an "external damper", a damper system is mounted on the crankshaft. The torque is transferred from the secondary damper part to the double clutch via a drive gear preloaded in a circumferential direction, which also compensates for the axial tolerances between the engine and transmission shafts.

In variant 2 with a "Cardan joint", two torsional vibration dampers are integrated in the two clutch discs and torque is transferred between the engine and double clutch via a flywheel with a cardanic function. The Cardan joint is formed by elastic spring elements that can compensate for the axial and radial displacements between the crankshaft and transmission shafts.

Coping with high energy inputs

One of the key issues in the development of dry double clutches is securing an "adequate" clutch life. Current design briefs envisage that the clutches must achieve the same life as the vehicle itself. Furthermore, there must be sufficient overload capability to handle extreme situations. The overload capability of double clutch transmissions must correspond to that of stepped automatic transmissions or CVT.

The overload capability of clutches is currently measured in terms of how often and/or how long a clutch system can be subjected to frictional energy without permanently damaging the system. A comparative test would involve starting off repeatedly on an incline over a defined time interval. Figure 4 shows the example of the clutch temperature curve over time with repeated hill starts. In dry clutches with current linings, the critical temperature

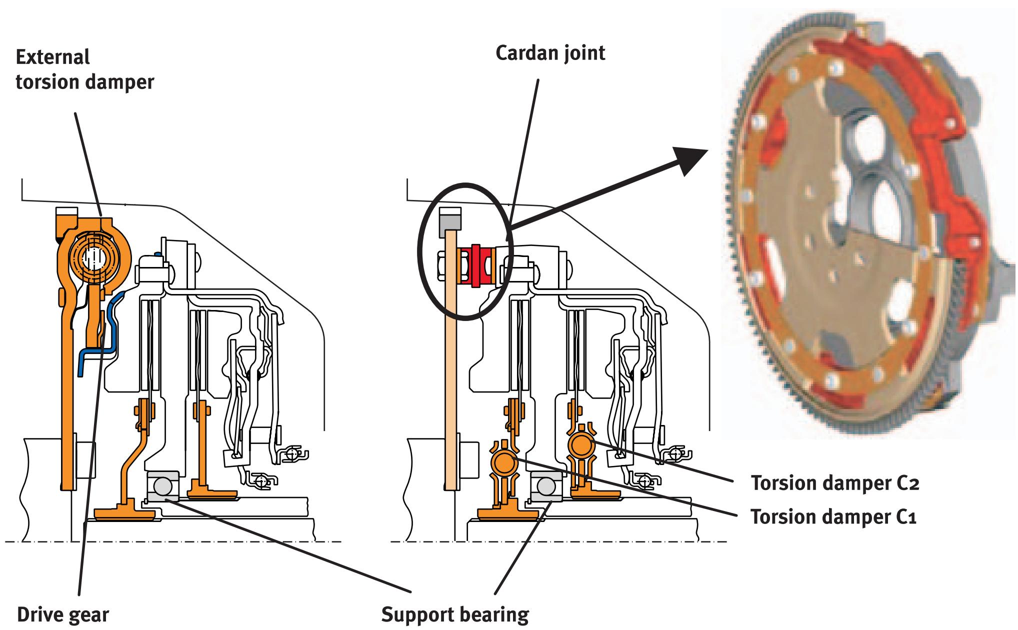

Figure 3 Arrangement of double clutches with torsion damper system in the power train

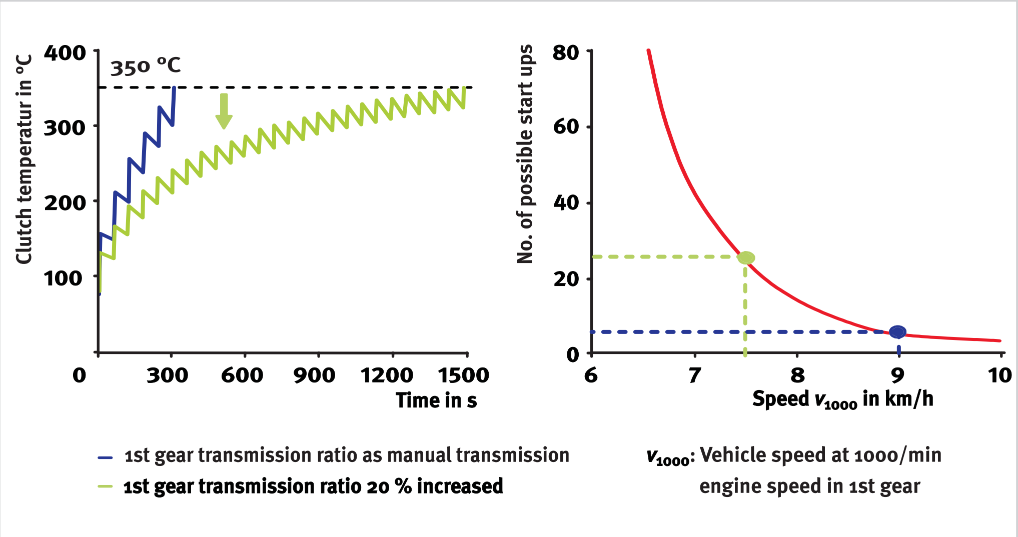

Figure 4 Clutch temperature curve and possible number of repeated hill starts

of the contact plate is approx. 350 to 400 °C. Above this temperature, the friction system starts to suffer permanent damage. In addition to the maximum temperature, however, the duration of the thermal load is also an important factor in damage to the friction system. The blue temperature curve in Figure 4 corresponds to the situation that exists currently in vehicles with manual transmissions when starting off repeatedly on a 12 % incline with a full load and trailer. The frequency of possible start offs can be considerably increased if the frictional energy per start-off is reduced by increasing the 1st gear ratio, as shown in green in Figure 4. A ratio change gives an approximately squared change in the start-off energy. This statement applies in general to all start-offs in 1st gear. Increasing the 1st gear ratio by 20 %, relative to a manual transmission, makes it possible to achieve an adequate overload capability of the double clutch for most applications.

There is additional potential for increasing the thermal robustness of dry clutches in providing the clutch housing with suitable openings to allow the heated air in the housing to be exchanged with cooler ambient air. The rotating clutch acts as a radial fan; in order to achieve high air throughput it must have an inlet near the centre of rotation and an outlet tangentially located on the outside diameter.

In order to prevent large quantities of contaminated air flowing continuously through the clutch area, it is advisable not to open the clutch housing until an air temperature of more than 100 °C is reached. This can be achieved simply and effectively by means of a thermostatically controlled flap (wax actuator) on the clutch housing. The effect on the clutch is indicated by the formula:

$$\dot{Q}_{out} = \alpha \cdot A\,(T_{pp} - T_{air})$$

$\dot{Q}_{out}$ = Thermal capacity
α = Heat transfer coefficient
A = Surface
T_{pp} = Pressure plate temperature
T_{air} = Air temperature

This means that the more heat that can be dissipated by the hot clutch components, the higher the temperature differential compared to the ambient air. In a simplified formulation, the housing air temperature is 50 K lower and the component temperature is approximately 50 K lower. Figure 5 shows a possible design for a thermostatically controlled clutch housing ventilation system and the resulting component temperatures.

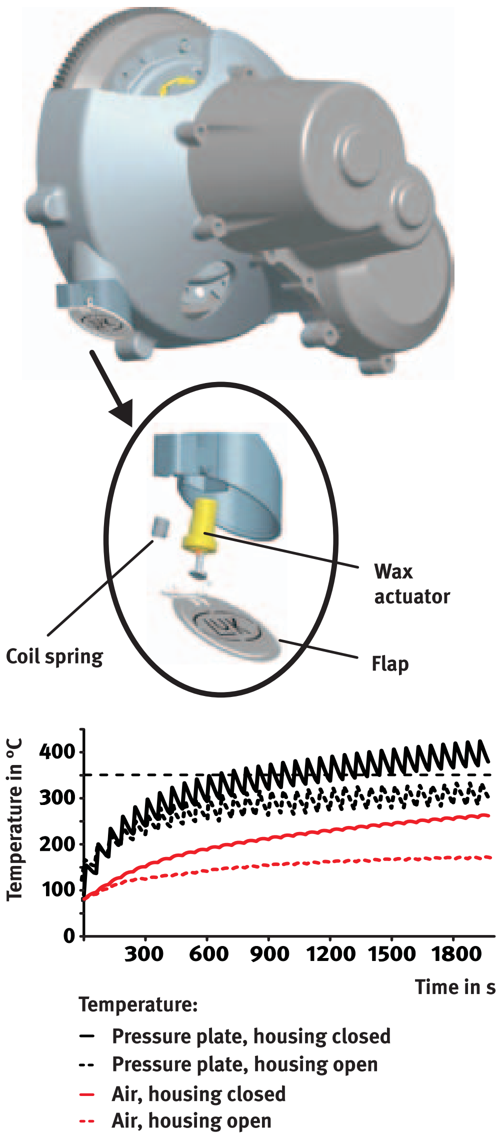

Figure 5 Schematic of open clutch housing

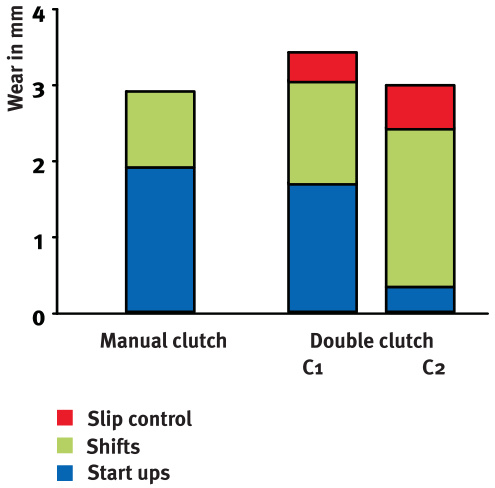

Figure 6 Clutch wear in dry double clutches

Service life

In addition to an adequate overload capability, the development of new double clutch transmissions also focusses on the expected service life. In comparison with manual transmissions, the clutches for double clutch transmissions are subjected to higher loads due to the overlapping gearshifts, slip control and more frequent gearshift operations. Slip control means that the clutches are deliberately set in the slippage limit range in certain situations in order to optimise comfort. Figure 6 shows the expected lining wear for a selected application.

The diagram shows that wear travel of approx. 3,5 mm is required on each clutch for a clutch service life of 240,000 km. This primarily influences the design envelope of the system. For dry double clutch systems with feasible space requirements, there are 3 basic approaches:

- Actively closed clutches with small internal ratio
- Actively closed clutches with wear adjustment by load sensor (LAC = Load Adjusted Clutch).
- Actively closed clutch with wear adjustment by travel sensor (TAC = Travel Adjusted Clutch)

Actively closed clutch with small internal ratio

A very simple and robust solution for a double clutch system is an arrangement comprising two actively closed clutches with an internal lever ratio of approx. 2:1, without measures on the clutch side for lining wear compensation (Figure 7).

In this solution, the zero point of the lever spring fingers changes by the wear travel multi-

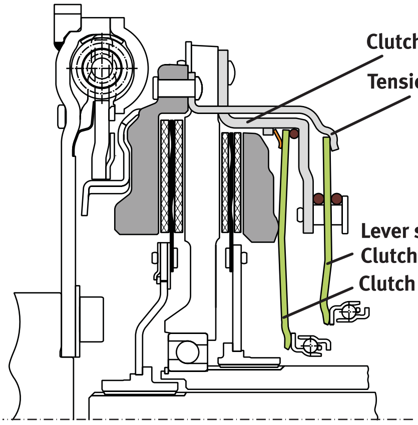

Figure 7 Double clutch with small internal ratio

plied by the clutch ratio. This change in travel must be taken into consideration in the design of the engage system. The disadvantage of this solution, however, is that the engage forces are higher due to the small clutch ratio, so that only engine torques of up to approx. 150 Nm can be handled while maintaining acceptable actuation forces.

Actively closed clutches with wear adjustment by load sensor LAC (Load Adjusted Clutch)

About 15 years ago, LuK started on the development of adjusters for dry vehicle clutches in order to mechanically compensate wear of clutch linings. A core system resulting from this development work is the self-adjusting clutch SAC [1-4], which has been in volume production for approx. 10 years and is fitted in many fully manual and automated manual transmissions. It seems logical to apply this proven technology to dry double clutches as well. In order to fulfil the specific requirements relating to double clutches, such as actively closed clutches and the highly restricted design envelope for the two clutches and two actuation systems, it was necessary to develop the system further. A double clutch system based on the load sensor principle has now been developed that has passed all the necessary function and endurance tests (Figure 8).

In order to achieve wear adjustment in the LAC, the lever spring of each subclutch is subjected to an axial abutment or sensor load by a sensor spring. In addition, both lever springs are supported on the clutch cover by means of a ramped ring. The clutch cover provides the opposing ramps on both sides, a particularly compact design. As in the self-adjusting clutch (SAC), wear is detected through the change in lever spring force by the sensor spring and compensated by the rotating ramp ring. Wear adjustment is completely automatic, free from overtravel and in very small increments, so no additional demands are made on the automated clutch control system. Both clutches can thus be matched so that the lever spring position remains almost unchanged despite the wear and the required axial travel for the complete system is minimised on the clutch side. However, there is a very long tolerance chain from the clutch lever spring fingers to the actuator system that requires compensation of tolerances through shims specific to the transmission. In order to avoid this time-consuming process in automotive assembly plants, intensive

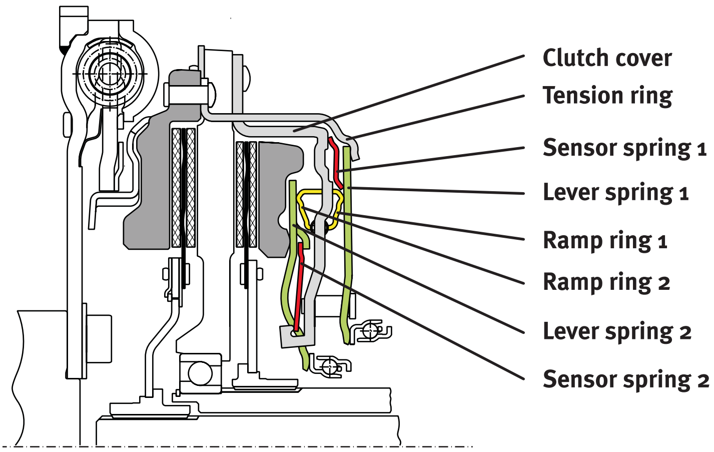

Figure 8 Double clutch with load-controlled wear adjustment LAC (Load Adjusted Clutch)

efforts are being made to develop a double clutch with a new wear adjustment. The most promising approach at present seems to be a wear compensation mechanism that senses and adjusts a constant engage stroke.

Actively closed clutch with wear adjustment by travel sensor TAC (Travel Adjusted Clutch)

In the development of dry double clutches, a central issue is implementation of the required functionality in the design envelope available in modern vehicles. The aim is to create systems in which functions are intelligently integrated in the components and system tolerances are largely eliminated. Of particular interest is the axial tolerance chain involving the two clutches and the corresponding actuation systems. In order to largely fulfil these requirements, intensive efforts are being made to develop a new double clutch system, Travel Adjusted Clutch (TAC). The TAC has two travel-controlled wear adjusters that can compensate for both lining wear and travel changes in the actuation elements as a result of mounting tolerances or wear. In this new wear adjuster, a slotted thin spring plate described as a “drive spring” is used not only to detect wear but also as a drive element for a ramp system. Figure 9 shows this drive spring with an outer tooth set in the two extreme positions “clutches open” and “clutches closed”. When wear occurs, the axial stroke of the lever spring increases such that the teeth on the right arm of the drive spring jump one stop further. The clutch then opens, exerting a torque on the drive spring which then rotates circumferentially and the teeth on the left arm jump one stop further. The lever spring is firmly connected to the rotating drive spring in a circumferential direction and therefore rotates as well. Wear compensation can be achieved if the lever spring fingers have formed circumferential ramps that are supported axially on ramps, for example in the preasure plate. Travel-sensing wear adjustment, however requires a special actuation system in which the maximum force can be restricted with relatively high accuracy.

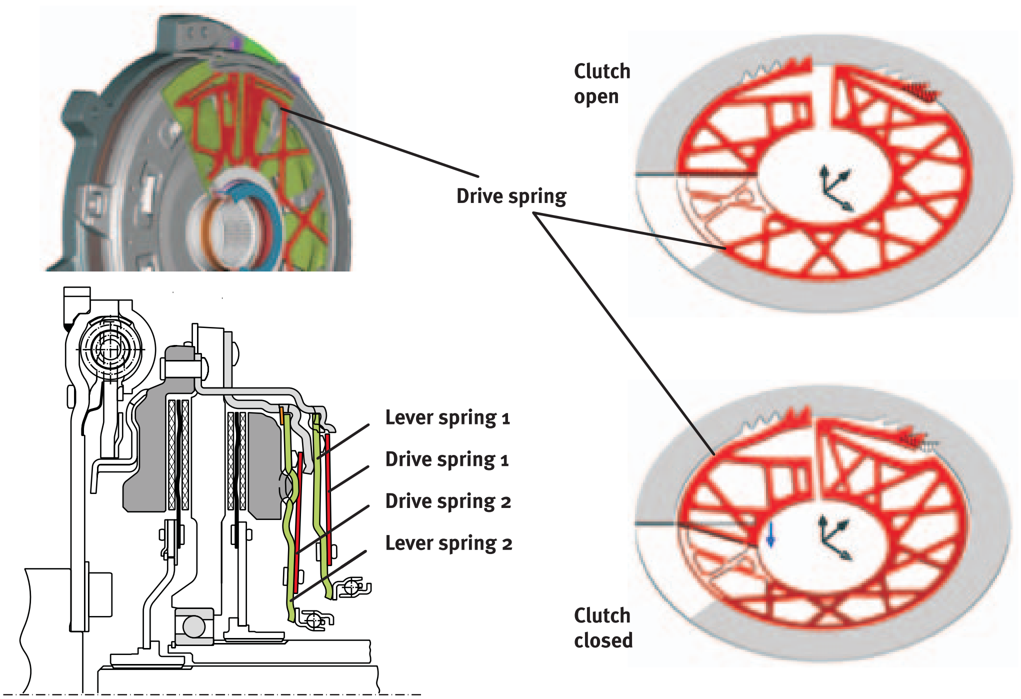

Figure 9 Double clutch with wear adjustment by travel control and stroke sensing (TAC)

The wet clutch

Complexity and requirements

The wet double clutches currently in the market or due to enter the market in the near future are actuated by hydraulic means. The pressure chambers rotate at the speed of the internal combustion engine (figure 10). This design with rotating actuator pistons is generally also used for the clutches in classical converter transmissions.

In order to move the hydraulic oil from the hydraulic unit to the pressure chambers, rotary passages are required that are sealed by slotted dynamic seals allowing leakage. This leakage is one reason why an additional hydraulic power pack is necessary to hold the clutch at the contact point while the engine is not rotating in order to achieve stop/start functionality. The clutch can then be closed easily and quickly when the engine restarts.

The pressure chambers are sealed by two seals each on the inside and outside diameter. In order to compensate for the influence of the centrifugal oil pressure that builds up under rota-

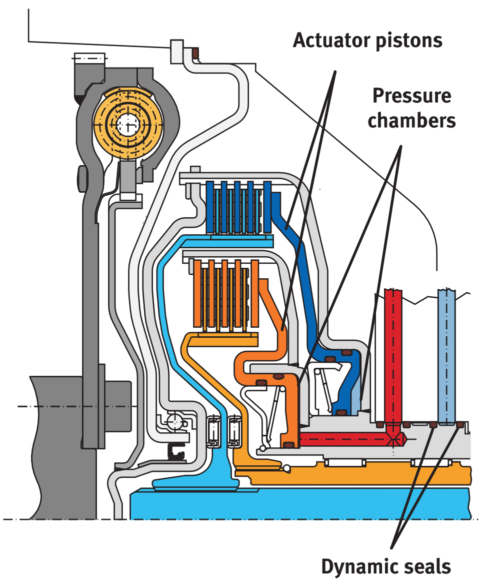

Figure 10 Wet double clutch with rotating actuator pistons

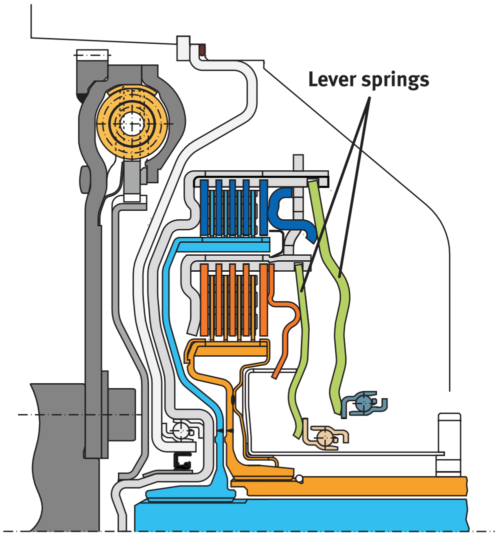

Figure 11 Wet lever-actuated double clutch

tion, additional oil chambers are included parallel to pressure chambers. At least one further seal is required per centrifugal oil chamber. The seals are largely responsible for the hysteresis in clutch actuation.

LuK, however, favours an actuation concept using lever springs similar to the state of the art used in dry, actively closed clutches (figure 11). The force is applied by the non-rotating, static actuation elements via engage bearings to the lever springs rotating at engine speed [5]. The engage bearings are thus the interface between stationary and rotating parts. The lever spring is suspended in the outer disc carrier and actuates a contact ring that presses the disc assembly together. If an actuator fails, the clutch opens automatically because of the force of lever spring. The external actuation forces are supported directly on a cover bearing, so the crankshaft is free from axial forces.

This system has the advantage that several actuation systems are suitable for use. If hydrostatic actuation elements or rotary or swivel levers are used in combination with slave cylinders, classic hydraulic control systems and electrohydraulic power packs can be utilised. An electromechanical actuator system can be simply adapted, too.

Actuation systems with an electrically-driven energy source independent of the internal com-

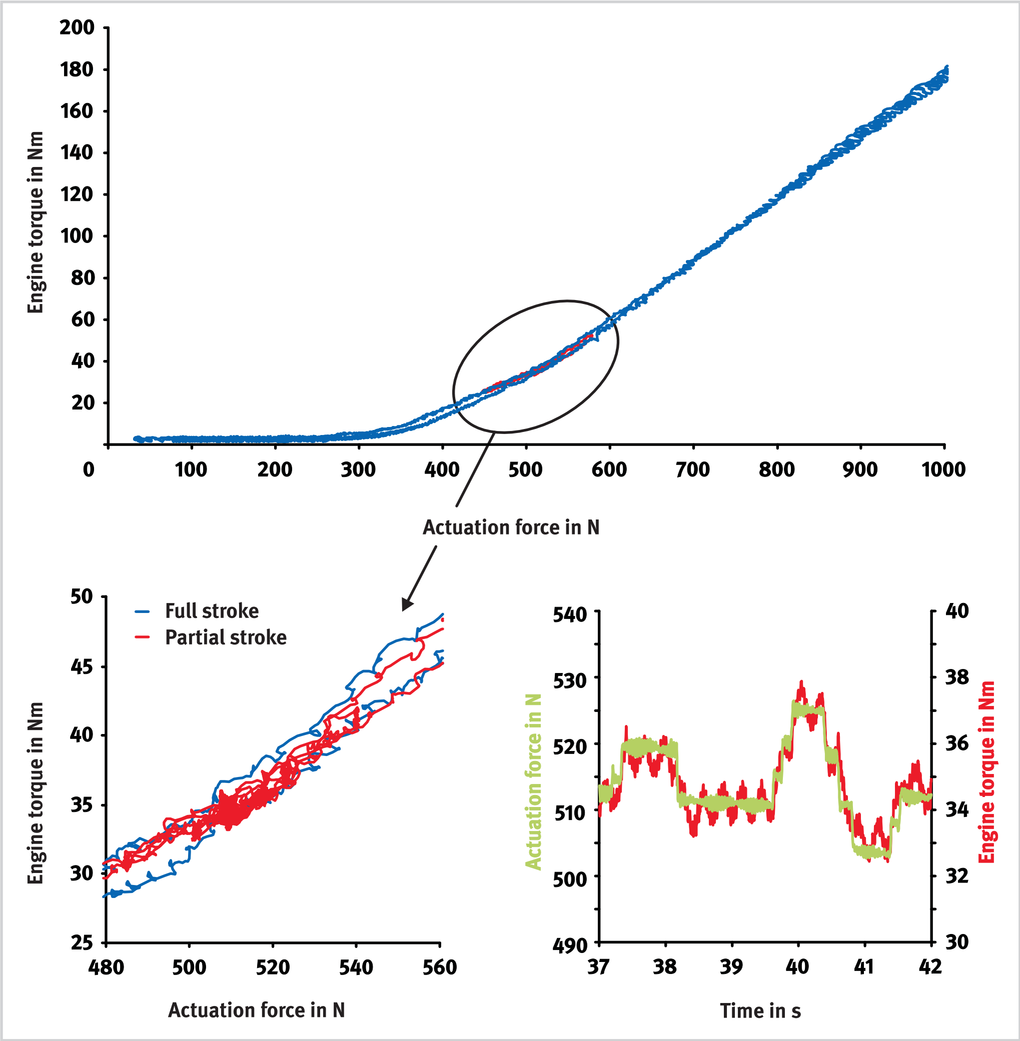

Figure 12 Hysteresis behaviour of a wet, lever-actuated double clutch

bustion engine benefit from the leak-free status of these lever-actuated clutches. This is an important precondition for hybridisation of the whole transmission, since not only can the clutches be actuated independent of engine running but there is also no need for permanent pumping of oil to hold the clutch at the operating point.

The lack of seals gives, in comparison with classic wet clutches, very good hysteresis values and sensitive modulation of the clutches (figure 12). Special shaping of the lever spring tongues allows compensation of centrifugal forces to be achieved simply.

When comparing the lever-actuated wet clutch with the dry clutch it can be quickly seen that, in terms of requirements and complexity, the wet clutch has no reason to fear comparison with the dry clutch (figure 13). It is important to know here that wet double clutch systems require no wear adjustment mechanism. Oil lubrication of the bearings and mechanical actuator parts saves on sealing work and special surface treatments can be omitted. Oil lubrication also ensures lower losses and hysteresis.

The relatively large mass of the cast materials in dry clutches is also significant. These provide

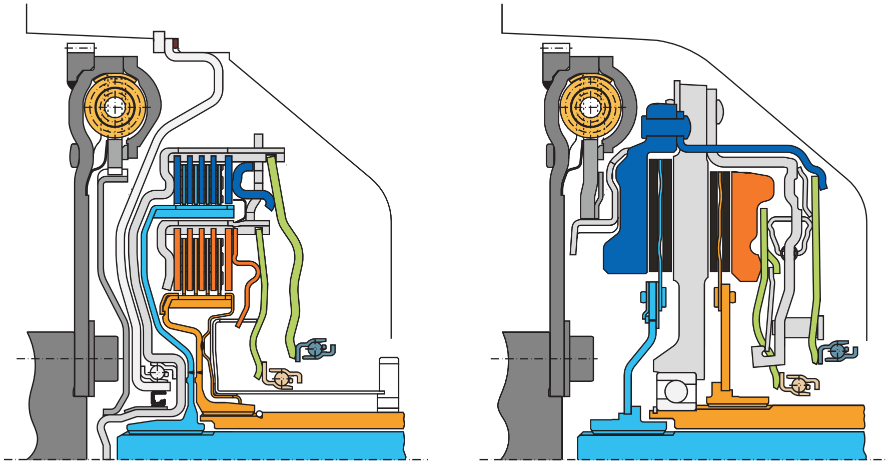

Figure 13 Comparison of lever actuation in the wet and dry clutch

the thermal capacity function, a task performed by the oil in wet clutches.

How is oil cooling of the clutches ensured with an electrically-driven actuation system? A suction-controlled circulation pump is used here [5] (figure 14).

This circulation pump is driven directly by the internal combustion engine. A cost-effective gerotor pump established in the automatic transmission sector and used as an engine oil pump is used. The outer gear of the pump is driven directly by a gear stage. This allows a radial arrangement that is neutral in design space in terms of transmission length and also saves on sealing work at the inner gear. In hybrid applications, the pump can also be driven by the hollow shaft via a double roller clutch in order to ensure cooling of the clutch even when the internal combustion engine is switched off. There is a suction control valve on the suction side that can match the flow rate to the coolant oil requirement during clutch operation and driving. This pump moves the oil

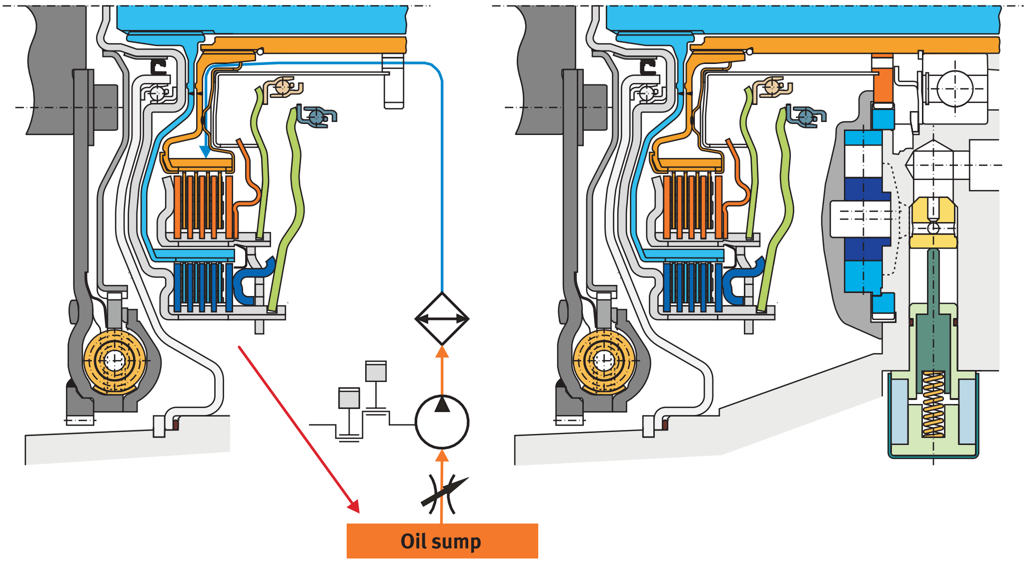

Figure 14 Cooling concept for an electro-mechanically or electro-hydraulically actuated wet double clutch

through an oil cooler before it is provided to the clutch as a cooled agent.

With the suction control valve in the "cool" position, the volume flow is determined by the theoretical displacement rate of the pump. In this state, adequate oil flows through the oil cooler maintains the thermal equilibrium of the complete transmission during "hill holding" and "uphill creeping".

In the "drive" position, the oil volume flow can be greatly reduced. It only prevents burning of the open clutch linings and ensures cooling of the actuated clutch under microslippage control. In this operating condition approx. 2...3 l/min of oil flow through the oil cooler. This oil quantity gives sufficient cooling for the small additional pump losses which compares to a manual gearshift of approx. 1 kW during rapid travel and hence maintains a stable thermal equilibrium in the complete transmission.

Since the pump is used only for cooling and lubrication of the clutch and only pressure losses in the oil cooler and ducts must be considered, pump pressures significantly below 1 bar occur in the range relevant to consumption and maximum pressures below 5 bar during clutch cooling and low oil temperatures.

These low pressures allow a technological leap in selecting the material for the pump, which can now be made almost entirely of plastic. The suction control valve can simply be integrated in the pump housing. Due to the use of plastic, no machining of parts is required and a significant cost benefit can be achieved (figure 15).

The combination of a lever-actuated double wet clutch and a simple, robust cooling concept significantly reduces the complexity of the wet clutch without compromises on functionality.

LuK is also working on cooling concepts using a ring-shaped oil cooler and the kinetic energy of the rotating oil emerging from the clutch to create coolant oil circulation (figure 16). A jet pump ensures permanent exchange between the circulating oil and the oil sump. This also allows complete utilisation of the oil sump as a heat sink in situations with high energy inputs.

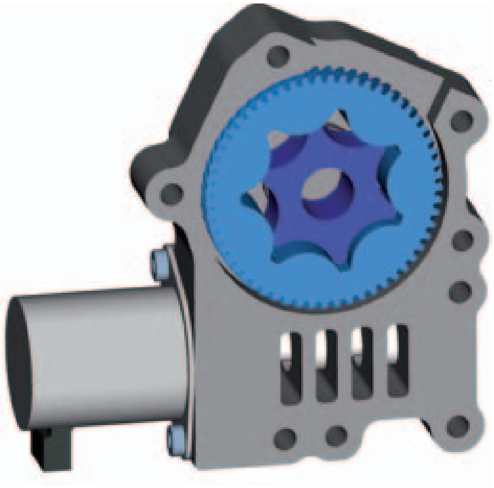

Figure 15 Plastic pump

These measures allow simplified feed layout, use of a more economical cooler and even elimination of the already economical plastic pump.

Fuel consumption

One of the most important issues in the development of double clutch transmissions is utilisation of the high transmission efficiency and the associated favourable fuel consumption of the vehicle.

If one analyses the fuel consumption data provided by automotive manufacturers, it is apparent that the high efficiency of the mechanical transmission side does not automatically give low consumption over the cycle (figure 17). In diesel vehicles with low and moderate power ratings in particular, the double clutch transmission still cannot use its advantages. Even the CVT transmissions, which are little disadvantaged in the partial load range in relation to variator efficiency, give some better or comparable results. At present the double clutch transmission is only convincing at high powered gas engines.

Why is this the case? The CVT transmission development engineers have compensated the small disadvantages at variator side by application of dual flow pumps and jet pumps as well as a hydraulic concept optimised in terms of low operating pressures and pump size.

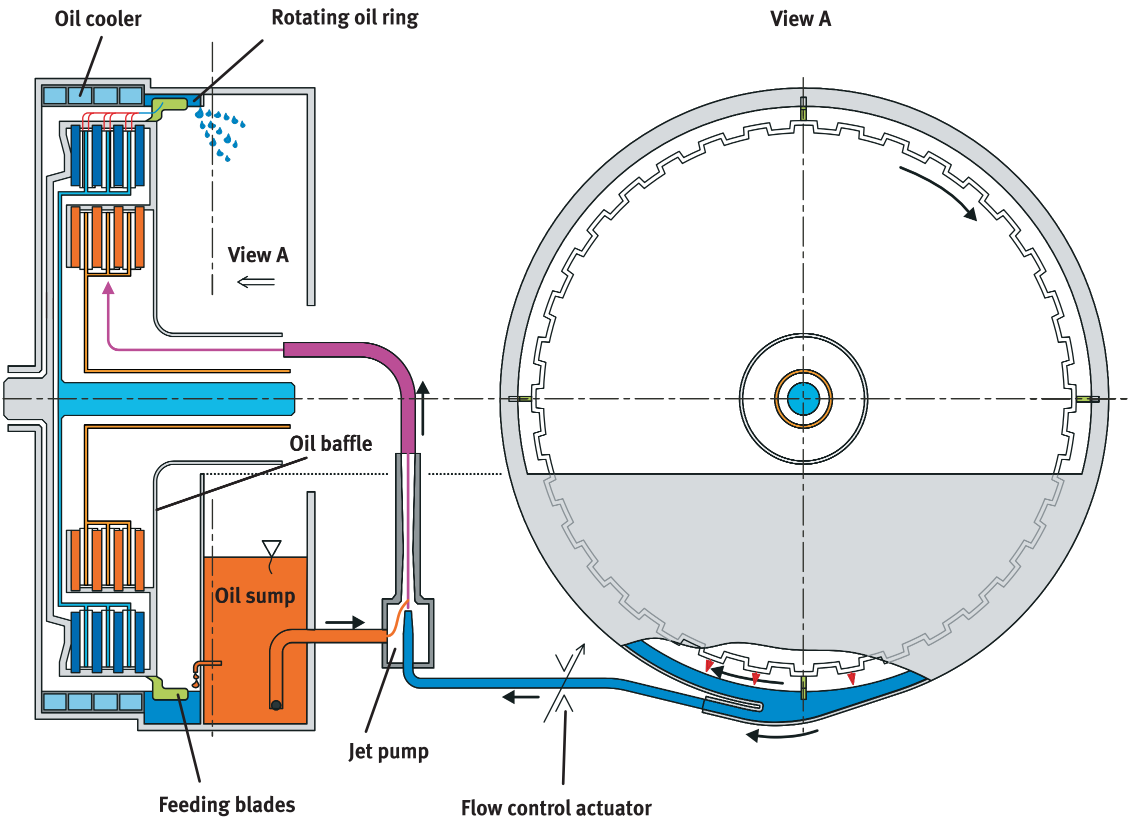

Figure 16 New cooling concept

The high efficiency of the mechanical transmission side of double clutch transmissions, the low mass moments of inertia of wet clutches, the feasibility of oil splash lubrication of the transmission and a suitable preselection strategy to eliminate drag torques in the disengaged clutch already give good conditions for exploiting further system advantages and eliminating disadvantages in comparison to other transmission concepts and also in comparison to double clutch transmission with dry clutches.

The greatest portion of losses are caused by the hydraulic pump which must on the one hand provide large quantities of coolant oil to cool the clutches and on the other hand create operating pressure to close the clutches. This conflict of interests leads to comparatively large pump sizes with correspondingly higher system pressures in the systems currently on the market.

Benchmarking studies have shown that when travelling at a constant 50 km/h, the pump losses influenced the fuel consumption of a 250 Nm diesel minivan by approx. 7 %. In the NEDC mixed cycle, this value was 7 ... 8 %.

There is enormous potential for improvement here. The lever-actuated clutch presented, in combination with an electromechanical or electrohydraulic actuation system and a circulation pump exclusively for coolant oil gives good conditions for such improvement.

The electrical losses in the electro-hydraulic power pack or the electric motors in the mechanical actuation system and electrically controlled suction control valve are of the same order as the losses in the numerous switching, PWM and proportional solenoids in the classical hydraulic control unit. It is thus permissible to concentrate on the comparison of the pure pump losses.

Pump losses in the almost pressureless circulation pump are approx. 1 % above in the 50 km/h case and in the NEDC mixed cycle. Fuel consumption benefits of approx. 6 ... 7 % can thus be shown compared with the state of the art (figure 18). Even when using optimised classical hydraulic concepts with a jet pump, consumption benefits of the order of 5 % or more can be achieved.

Based on these observations, it is clear that the wet clutch cannot be held responsible for

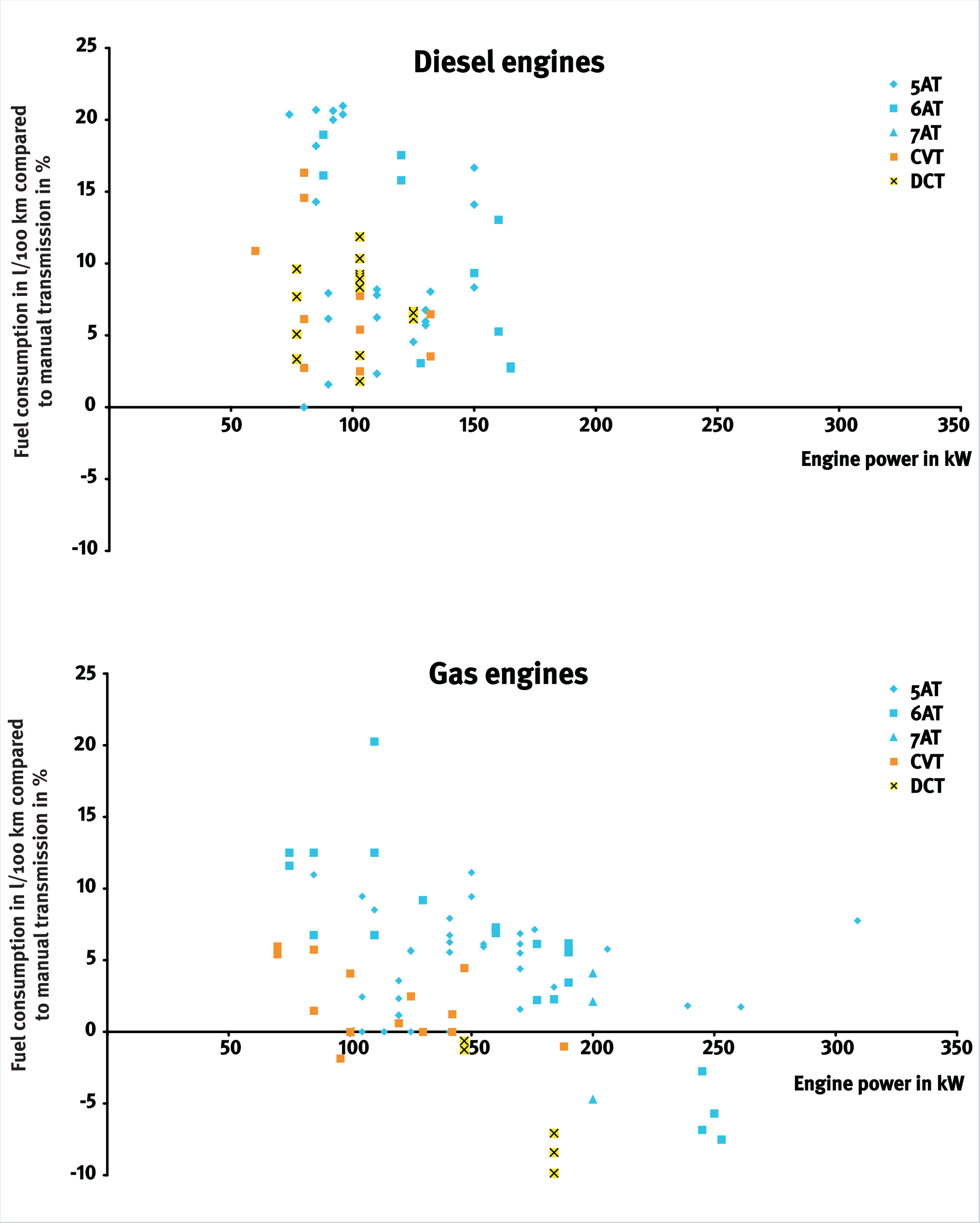

Figure 17 Differences in fuel consumption between various transmissions [NEDC mixed in l/min] compared with manual transmission with same engine power in same vehicle. Sources [6-9]

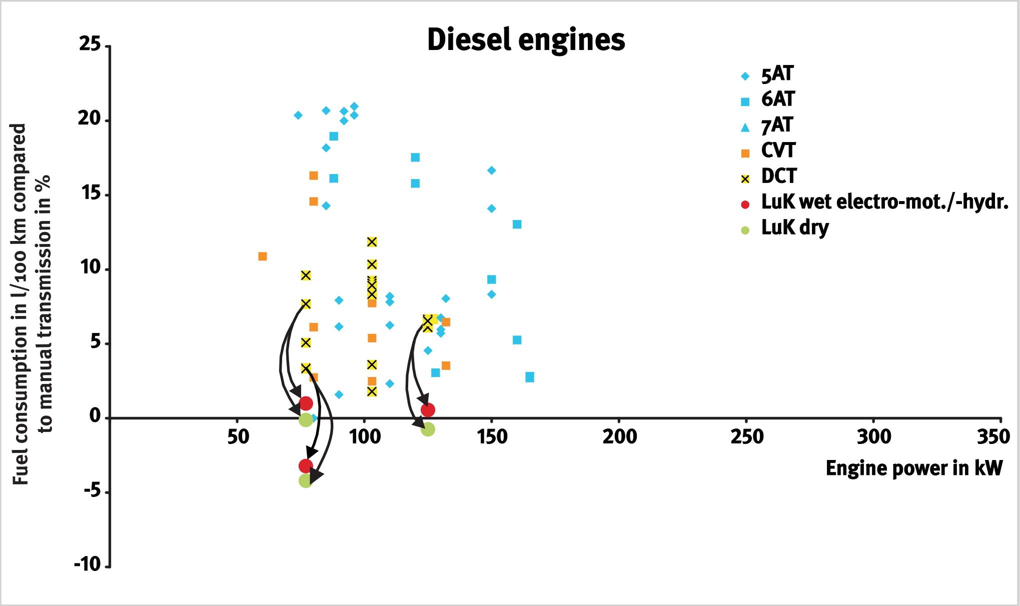

Figure 18 As Figure 17 for diesel applications, additional data for three selected vehicles for dry and wet double clutch transmissions with LuK electric actuators

higher fuel consumption; the actuation and cooling concept plays the decisive role. If losses here are actively reduced and advantages such as the lower inertia forces and splash oil lubrication are exploited, consumption values move very much closer to those of dry systems.

The use of electric actuators for clutch actuation and gearshift gives good conditions for the hybridisation capacity of double clutch transmissions. The extra cost of an additional power pack in classical hydraulic control systems to ensure prefilling of the clutches while the internal combustion engine is switched off can be saved by using electric actuators independent of the internal combustion engine. As a result, this actuation concept in combination with a lever-actuated wet start-up clutch is well equipped for the demands of the future.

Conclusion

Once boundary conditions such as torque capacity, axial design envelope, fuel consumption, life, ability to handle high energy situations and costs are taken into consideration, wet and dry double clutches have their specific advantages and disadvantages that, depending on the weighting and customer philosophy, may lead to different decisions.

Whether the double clutch transmission will make a major breakthrough or remain a niche product will be determined by market forces. Double clutch transmissions currently still have some potential in terms of fuel consumption and costs in comparison with CVTs and the best stepped automatic transmissions [6-10]. Attention must also be paid to the issue of hybridisation in all automatic transmission concepts.

Due to simple principles for wear adjustment and greater robustness in high energy situations, dry double clutches have also become a serious alternative for moderate to high engine torques.

Much of the dry clutches technology was carried over to the wet double clutch systems. Now it is possible to draw together the advantages of electrically driven actuators combined with a circulation pump, with the advantages in high energy situations and for the design envelope, especially at high power levels.

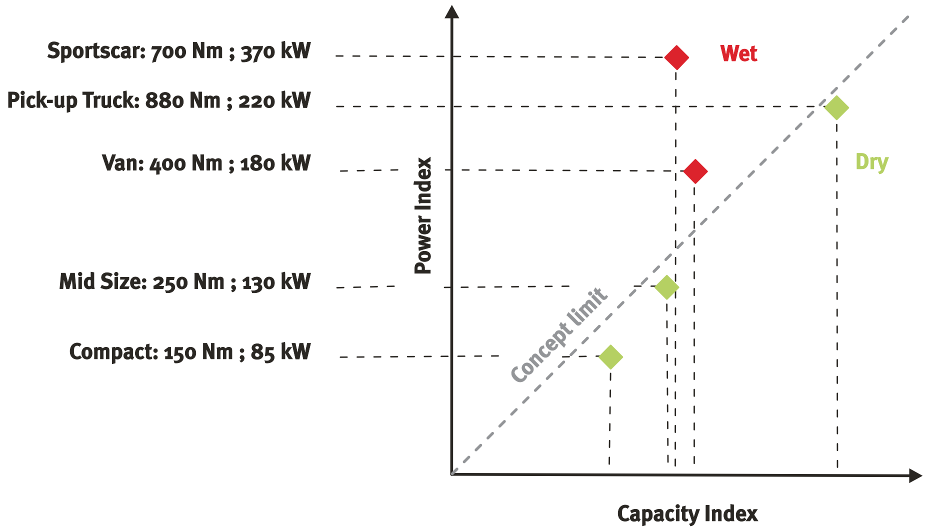

Figure 19 Dry or wet double clutch?

The decision in favour of a wet or dry double clutch depends on specific customer requirements, the vehicle and drive train parameters such as engine torque, startup ratio and vehicle weight as well as the design envelope and cooling conditions. The decision limits vary as a function of the vehicle category [11] (figure 19). An 880Nm pickup truck may still be fitted with a dry double clutch, while a van with a 400Nm engine may require wet clutches.

In both cases, an electro-mechanical or electro-hydraulic actuator system may be found to be advantageous where the aim is to achieve favourable fuel consumption compared with the state of the art or create the conditions for hybridisation. The solutions presented here show that this is not necessarily associated with complex solutions and additional costs.

Since lever-actuated dry and wet clutches have the same interface for introducing the actuation force, namely the engage bearings, a modular concept can be achieved [5, 11] (figure 20). In identical transmissions with electromechanical or electrohydraulic actuation of the gearshift, wet and dry clutches can be arranged in the clutch housing with an identical clutch actuation system. All the necessary adaptations, such as additional mounting of the circulation pump and suction control valve for the wet double clutches, can be restricted to the clutch housing, which must in most cases be adapted to the different engine flange mounting patterns anyway. As a result, the same basic transmission including the actuators can simply be matched to widely differing customer requirements and a future-proof solution offered that is matched to specific demands.

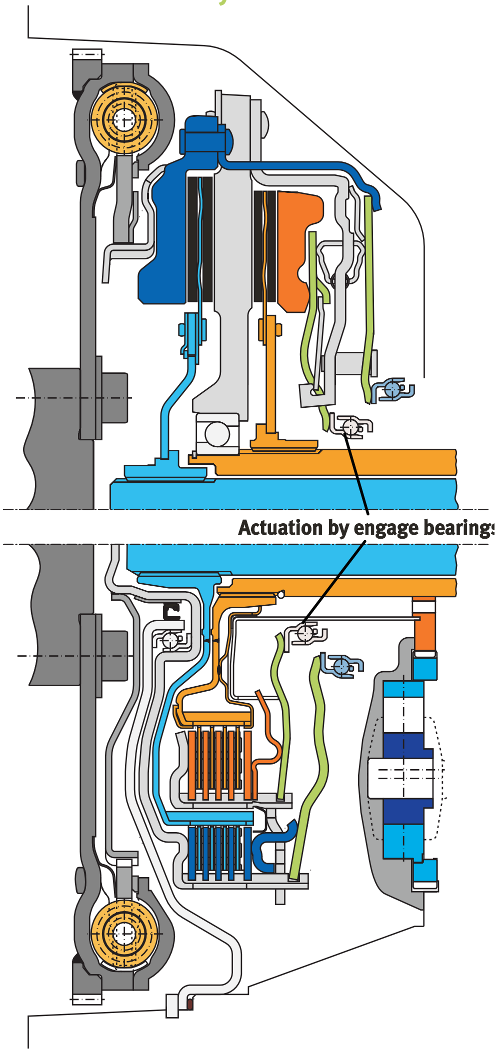

Figure 20 Modular clutch concept

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