

CVT components for powersplit commercial vehicle transmissions

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1 Introduction

The rise in fuel costs and the necessity for further reductions in emissions require new technical solutions allowing further optimisation of the entire drive train. As a result, the use of chain variators is also expanding in the passenger vehicle sector. Other applications will be added to the Audi multitronic® VL300 and its newer evolution the VL380 with 420 Nm variator torque [1]. Optimisation of the power train is also an important target for buses, vans and commercial vehicles as they are responsible for a considerable proportion of emissions. A manufacturer of commercial vehicle transmissions came up with the idea to make the total ratio of the transmission so large and variable that the engine can be operated more or less steadily in a lowest consumption and emissions range. Engine optimisation in precisely this range then offers additional potential for improvement. The variator required for such a transmission was positively evaluated on the basis of the LuK production components and a concept study, not only on the test bench, but also in a vehicle. The LuK components are: the primary and secondary pulley, the chain with the guide rails and the hydraulic manifold.

Vehicle	Audi A6 3.2FSI / 2.7TDI	Commercial vehicle
Permitted weight [kg]	2200 ... 4000	10000 ... 40000
Max. engine torque [Nm]	330 / 380	1000 ... 3000
Max. engine power [kW]	188 / 132	200 ... 500
Required lifetime [km]	300000	> 1000000
Max. speed [km/h]	250 / 225	120
Period of operation [h]	> 3000	> 15000
Exhaust standards/target	EU4 / EU5 draft	Euro 5 draft (through SCR)
CVT transmission	Audi VL300 / VL380	Power split CVT
Transmission structure	D & Reverse, unsplit	multi-range D & Reverse
Center distance [mm]	171	220
Max. variator slew speed [1/s]	1 ... 2	4 ... 5
Max. oil requirement [l/min]	5	19
Max. variator torque [Nm]	-60 ... +350 / -60 ... +420	-350 ... +600

Figure 1

Data and targets comparing passenger cars and commercial vehicle utilisation of a CVT

If the data of an Audi A6 is compared with the data of various commercial vehicles, as listed in figure 1, the task seems almost impossible.

The variator data shows, that more than one individual component must be optimised to fulfill these requirements. The following sections deal with all these aspects including the architecture of the powersplit transmission, the variator pulleys with the chain and the

hydraulic manifold. In the course of the development, it becomes clearer that the challenge of realising a CVT for commercial vehicles is acceptable.

2 Transmission architecture

One of the key technologies in this CVT application is the principle of power splitting as well as the use of several continuously variable driving ranges. The principle of a power split CVT was also described during the first CTI Symposium 2002 [2,3,4] for passenger car applications. Power splitting allows an increase of efficiency in combination with a reduction of variator load. The concepts for car transmission designs [5] introduced with this technology cover capacities up to more than 200 kW and corresponding torques, thus leading into the commercial vehicle segment.

This technological background makes it possible to develop customised transmission structures, as is also the case, for example, with hydrostat-based transmissions [6]. In comparison with the hydrostat, the chain variator offers efficiency and acoustic benefits which are of particular necessity for use in buses. In comparison to a wholly electrical power conversion, the benefit of the chain converter in cost, efficiency and power density is even greater.

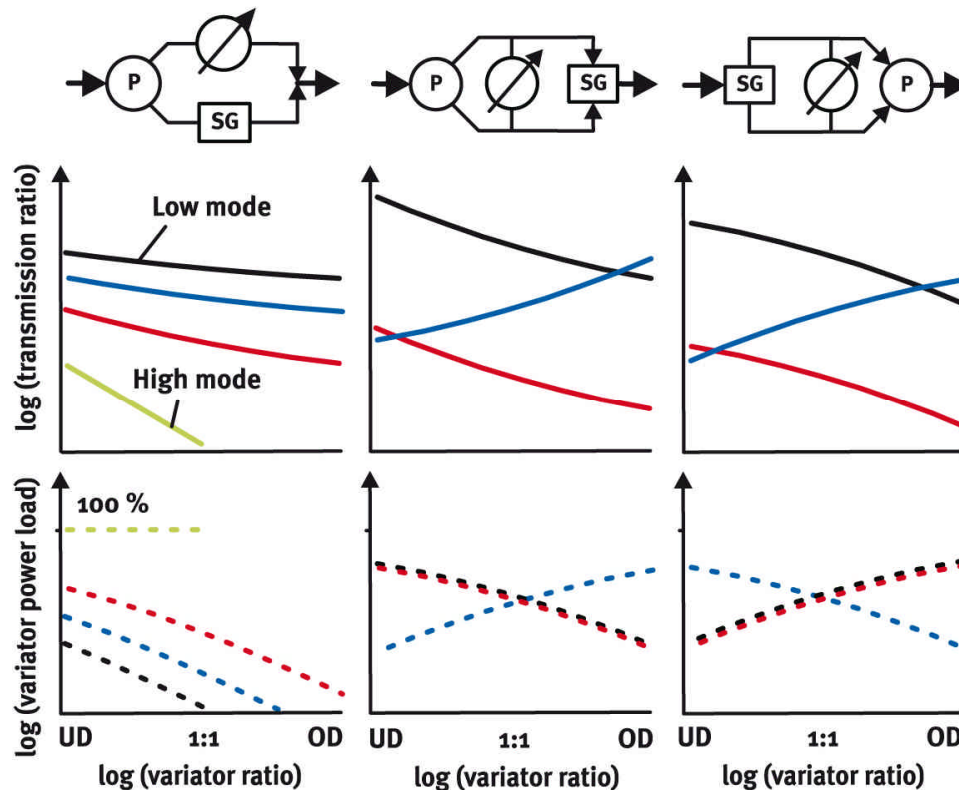


Figure 2 Three alternative transmission architectures with ratio and power load characteristics. Each color represents one of several operation ranges. P=planetary, SG = automated shift gearbox

A common prerequisite of applicable gearbox architectures is a speed-up of the variator, because commercial vehicle engines deploy their power at lower speeds. With some of the transmission architectures described, this function can take place directly in the planetary gear provided for input-side splitting. Also planned is a transmission for selection of multiple ranges that can work where necessary with the conventional commercial vehicle dog clutch if a speed synchronisation is achieved by suitable means.

The options to link a planetary gear and a chain variator together can only be divided into two classifications, namely with an input-side or output-side planetary gear. In each of these two classifications, there are alternative configurations for the multi-range manual transmission. Some options are described in figure 2.

In the architecture shown on the left, the planetary gear is configured on the input side and the multirange-transmission in the power path parallel to the variator. If the shaft leading to the transmission can be locked, a driving range without power splitting can also be presented. Below the drawing of the transmission architecture is an example of how, with appropriate gear ratios, several continuously variable driving ranges can be represented. In each case, the bottom diagram shows the power percentage to which the variator is subjected.

Depending on the application, the benefits of one architecture category or another may prevail. The transmission depicted on the left produces low driving ranges with very low variator load – beneficial for an almost steadily used unit. When changing from one driving range to the next, the variator resets.

The center column shows a transmission which has a manual transmission assembly with two non-coaxial input shafts. In contrast to the transmission shown on the left, it is possible to change between ranges without resetting. To do this, the variator must transfer an average of 50 % of engine power – more than with the transmission on the left, but this is sufficient for light commercial vehicles or vans.

With the transmission shown on the right, the cast of parts is simply switched between input and output. In the example at hand, this primarily influences the torque and speed ranges – but not the performance.

In order to depict a 'geared neutral' transmission, the planetary gear should fundamentally be placed on the output side. If the planet is configured on the input side, the opposite can be shown: very long ratios up to a 'geared zero' ratio which allows continuous start/stop of the engine.

In drive mode, the benefit of all the architectures shown in figure 2 is the low number of gears and therefore range changes compared to conventional commercial vehicle transmissions. For example, with four driving ranges after start-up only one shift is made and city centre driving can continue virtually without any range change. Another range change is necessary on leaving the city and a final one on the motorway.

3 Variator development beyond 500 Nm

3.1 The main dimension: Center distance

In order to increase the torque capacity, it is necessary to enlarge the main dimensions. With the center distance of 220 mm selected here compared to the 150 mm ... 190 mm feasible for passenger cars, a whole series of aspects are relaxed. In part, the effect is clearly even greater than the 25 % enlargement in the center distance, as the following list illustrates. The changes are indicated by arrows, the number of which reflects the relationship to the center distance enlargement:

Change	Benefit
Chain radii ↑	Chain tractive forces ↓ Clamping forces ↓
Number of chain links ↑	Force per rocker pin ↓↓
Chain rocking angle ↓	load distribution on chain link cross section ↓
Force per rocker pin ↓↓	Deflection of rocker pin ↓ Link load on chain edge ↓↓
Shaft diameter ↑	Bending stiffness ↑↑ Efficiency ↑
Crowning of surface ↓	Stress ↓, Wear ↓, Wear per rocker pin ↓↓

3.2 Efficiency measurement up to 600 Nm

To confirm the above considerations as well as the measurement results [7] and simulations [8] published by research institutions, efficiency measurements on the 220mm chain variator were carried out at LuK on a new high-performance test stand. As a complete characteristic diagram comprising several ratios, speeds and torques (and to some extent clamping forces) was to be used as the basis, this also posed considerable challenges for the test department. For example the electrical generator engine of this test stand has a nominal power rating of 720 kW.

The results confirm that chain variators achieve efficiency figures of over 97%. Figure 3 shows the 600 Nm efficiency characteristic diagram with a constant drive speed of 2500 rpm. Shown are the raw measurement data (except smoothing for noise suppression) from the test transmission including the losses of the pulley bearings. The center distance of 220 mm thus shows the expected positive effect.

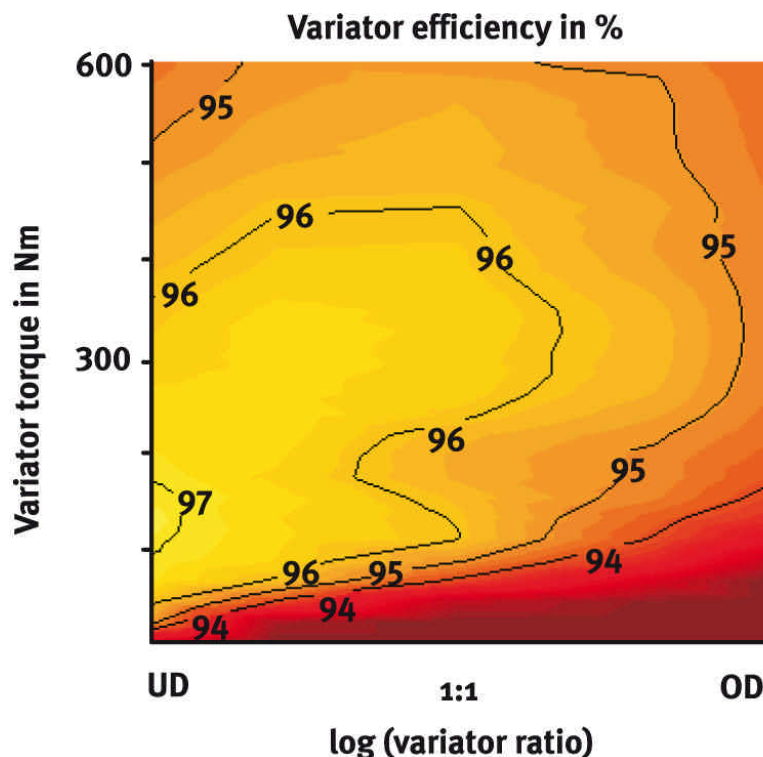


Figure 3 Measured efficiency map of the 220mm chain variator

In addition, efficiency measurements have been performed in the entire transmission including the hydraulic manifold with the pumps. A maximum value between 98% and 99% in direct-drive mode (3rd gear in the present concept) of the automated shift gearbox is achieved.

3.3 Continuously variable hydromechanical torque sensor

The pulley technology in use on the Audi VL380 is the basis for applications at even greater torques [9]. Of central importance is the space-saving continuously variable torque sensor [14] implemented inside the pressure chamber which has been further-developed for the commercial vehicle application, as shown in figure 4.

The variable torque sensor provides the indispensable properties for reliable continuous operation in commercial vehicles:

- Precise clamping force proportional to the actual torque for all ratios
- Prompt clamping to prevent damage even with step-like torque changes
- Easy adaption of the clamping strategy to different engine types

The function of the continuously variable sensor shown in figure 4 is to convert the torque introduced to the primary pulley via ball ramps into an axial force. However, the axial force produced by the ball ramps does not have a direct effect on the cone pulley but closes a hydraulic outflow orifice. This mechanism very dynamically adjusts a hydraulic pressure proportional to the torque. The pressure then generates the actual clamping force for the moveable sheave via large clamping pistons. The same pressure is also supplied to the driven shaft for clamping.

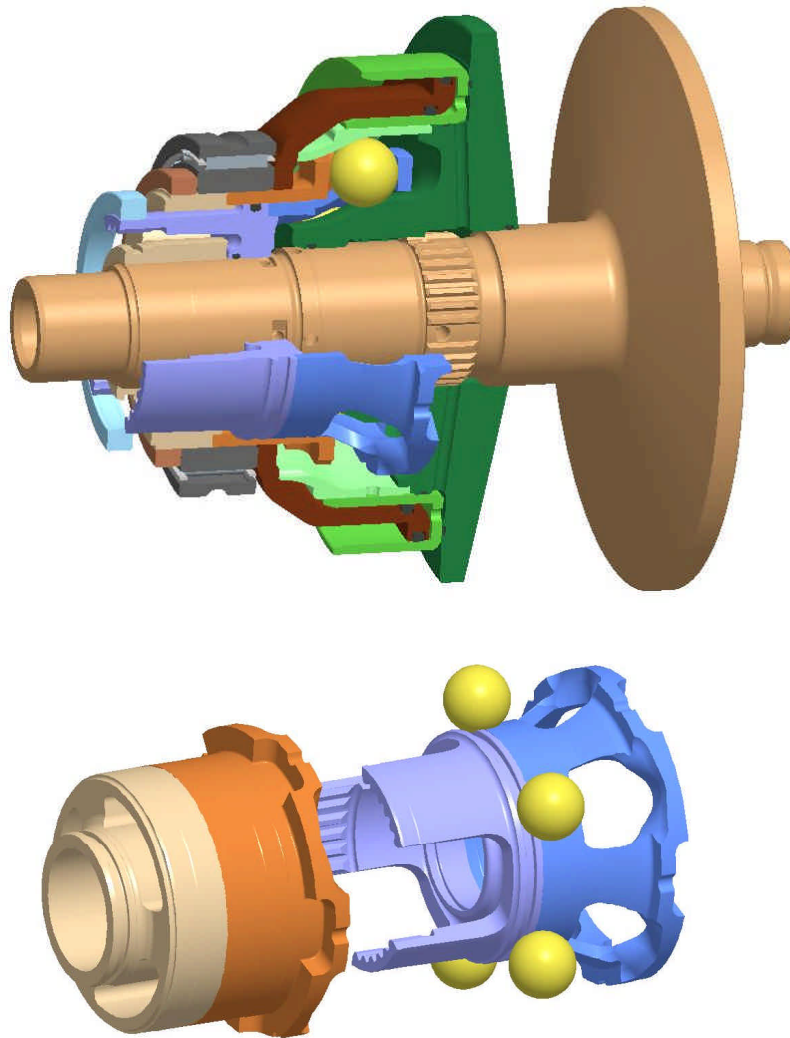


Figure 4 Primary pulley set (above) including the variable torque sensor (below)

The dependency of the ratio is thus achieved because the ball ramp mechanism has different ramp slopes at different radii. The ramp angle appropriate for the respective ratio is selected through the radial positioning of the balls by means of the guiding surfaces, which slide axially ratio-dependent with the moveable sheave.

In the result, the pressure related to the torque is highest in underdrive, decreasing continually until overdrive. The three-dimensional shape of the components is optimally adapted to the clamping requirement determined under many loads.

The promptness achieved through the direct hydromechanical principle is illustrated using the measurement in figure 5.

In the measurement shown, a step in the drive torque (upper curve) excites a decaying drive-train oscillation on the test stand. In a real vehicle this corresponds, for example, to a sudden, jerky acceleration. Even in this situation there is virtually no delay between the

measured torque and the measured pressure adjusted by the torque sensor (the two lower curves). Quantitatively, the adjusted pressure at each instance is also congruent with the target pressure calculated from the measured torque.

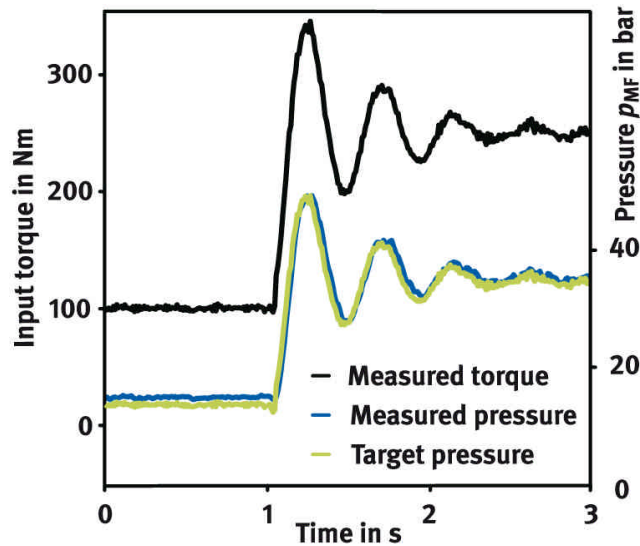


Figure 5 *Promptness of the variable torque sensor clamping system when exposed to a steplike change of the input torque.*

4 Hydraulics

The double piston principle already used in the multitronic© makes it possible to hydraulically operate all required quick changes of ratio even with small installed pump capacity. The resulting benefit in consumption is also to be strived for with commercial vehicles. Due to the extremely rapid adjustments required with the existing multi-range transmission structure, the double piston principle is actually indispensable for efficient hydraulics. This is the basis for the development of the hydraulic manifold and the pumps.

4.1 Independent actuation of the adjustment pressures

Despite upscaling the entire system and the piston surfaces, there is considerably higher coast torque due to engine brake assemblies, which causes an increase in the required peak system pressure up to 100 bar. This requires higher pressure amplification in the corresponding valves for the adjustment pressure chambers. Due to independent actuation of both valves, this high valve amplification is stable with volume flows up to 19 l/min and the hydraulics gain actuation precision.

The tasks of the clamping and adjustment system and the solution implemented by this hydraulics are clearly illustrated in figure 6. For steady operation, the variator requires two forces on the pulley sets which are in a particular proportion dependent on the ratio, known as the force-balance k_p/k_s . The task of the clamping pistons is to generate the basic clamping forces via the variable torque sensor on both pulley sets, whereupon high-pressure oil is exchanged between the pistons during variator adjustment. The task of the adjustment pistons is to generate the additional residual forces and adjustment forces required for equalisation.

This hydraulic system offers not only a high degree of stability and precision in the control of the pressures, but also permits other advanced functions due to the independence of the pressure control. Examples of these functions include an increase in clamping safety on poor road surfaces or a slight reduction in clamping force with appropriate design of the torque sensor, e.g. for compensation of residual centrifugal oil pressure forces. The result is optimised operating efficiency.

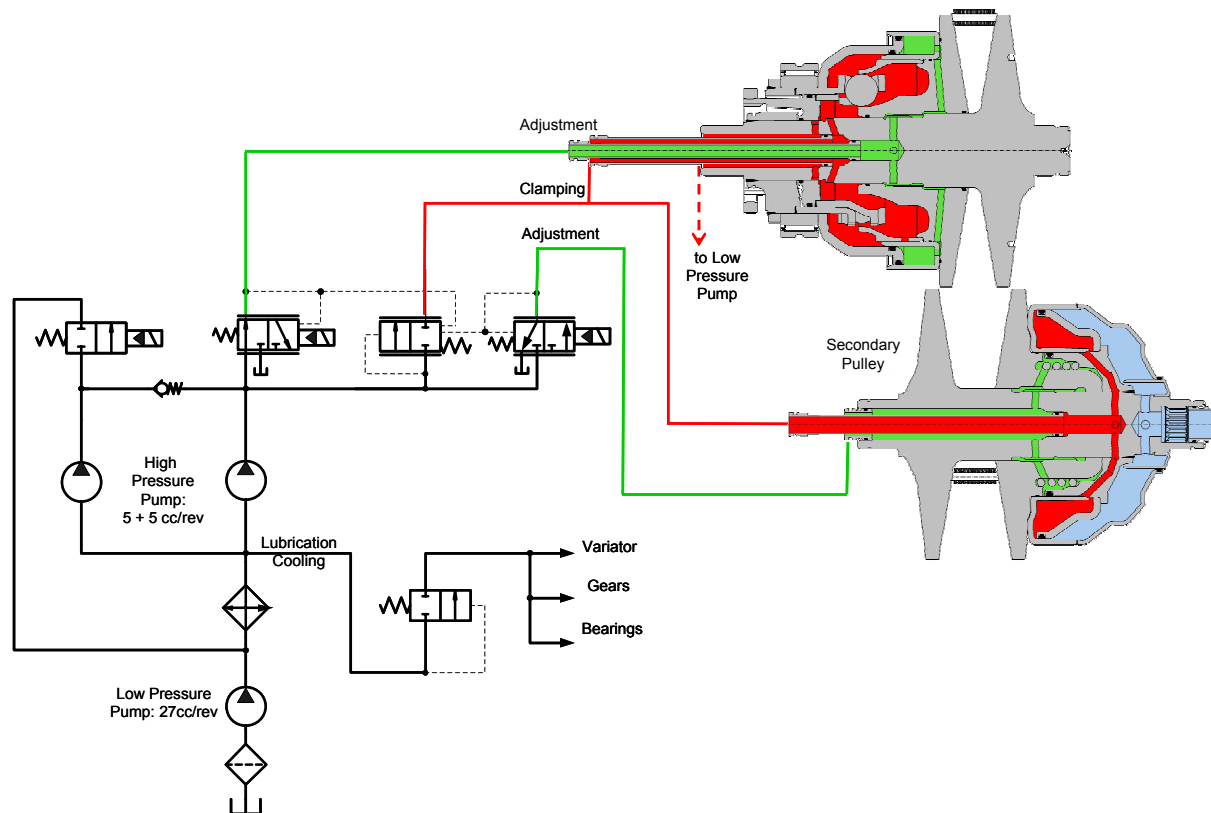


Figure 6

Schematic of the hydraulic including a low and high pressure pump as well as the double piston principle with torque sensor clamping and independent adjustment pressure actuation.

When adjusting the ratio, the force balance is changed in a controlled manner. Each adjustment pressure may be optionally reduced and/or the other adjustment pressure increased in combination. This degree of freedom is beneficial especially with the rapid adjustments of a multi-range transmission.

Thus the benefits of several systems are combined here: The continuously variable torque sensor contributes robustness and promptness. The independent pressure control of the adjustment chambers offers the freedom of software-controlled free clamping force. A combination with slip control of the clamping force [10] is also possible with this system, without losing the benefits of the torque sensor.

4.2 Pumps and cooling system

The lubricating oil and cooling requirements of the variator, gears, and bearings make use of a low-pressure pump which is advisable in transmissions with a power classification in excess of 400 kW. Following identification of the design-relevant operating point, the result for the chosen transmission with a dry clutch is a low-pressure gerotor-style pump with 27 ccm/rev delivery volume.

The low-volume high-pressure pump which is required for the clamping of the variator and the adjustment of the ratio is designed as a symmetrically divided, dual-flow, fully compensated vane pump with a delivery volume of 10 ccm/rev in total. A similar pump developed by *ixetic* (former *LuK FH*) is also in production in *DaimlerChrysler's* Autotronic© [11].

In combination with the low-pressure pump, a high pressure pump forms a tandem pump as a compact unit, figure 7, on a shaft which is overdriven by the engine. The delivery of low-pressure oil guarantees a cavitation-free supply to the high-pressure pump, permitting the compact design of the intake system as well as an efficient filter concept.

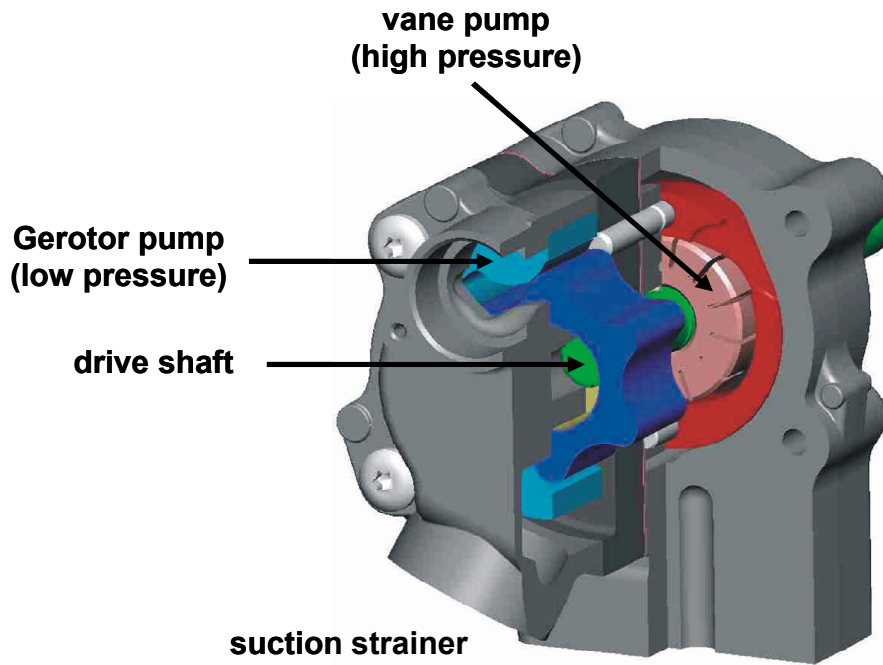


Figure 7

Compact design of the hydraulic tandem pump including Gerotor (low pressure) and Vane pump (high pressure)

4.3 Pump efficiency due to intelligent control of the pump flows

In comparison with a single-flow high-pressure part, the hydraulic power requirement is considerably reduced through intelligent control of the second pump flow. For this purpose, an electronic control valve is included in the hydraulic control unit.

By means of this valve, the second flow of the vane pump is connected to the input. The pump's drive torque is thus drastically reduced in the majority of driving situations where there is no high demand of oil volume. Note that this oil is not lost but used for the low-pressure consumers (lubrication and cooling).

On the other hand, in driving situations with high demand of oil volume the flow control valve is electrically controlled (disconnected) such that the second flow is united with the first flow via a one-way valve. This ensures that there is also enough high-pressure oil for fast adjustment of the variator, e.g. when starting up or when changing the driving range.

Based on the optimised design, the pump losses of this transmission are low, as is also the case for the Autotronic®. An alternative concept with only one pump for all consumers would have caused a threefold power requirement with no cost benefit because the costs of a high-pressure pump are scaled unfavourably to those of a low-pressure pump.

Consistent use of the surface technologies for wear protection positively tested in the car application, e.g. hard anodizing of the pistons and the valve body bores, ensures smooth hydraulic function even with the high life expectancy of a commercial vehicle.

5 CVT chain

The increasing experience and process optimisation with the strength-optimised light-link geometry have increased the torque capacity of the 37 mm wide LK3708 chain to such an extent that, according to the first trials with the 220 mm center distance, no enlargement of the chain seems necessary for 600 Nm variator torque. All the results described in this article have been achieved with this 37 mm wide chain. The connection between the center distance, chain strength and torque capacity described in the first CTI Symposium [3,4,12] is even exceeded with large center distance.

5.1 Chain strength

The range of chains in figure 8 is completed by narrower chains and chains with reduced pitch in the lower torque range. To allow for torques greater than even 600 Nm, a chain with an expanded pitch, i.e. LK10 links with stronger link cross section is also under development. Thanks to stronger rocker pins, the forces from the chain edge are evenly distributed on the adjacent links.

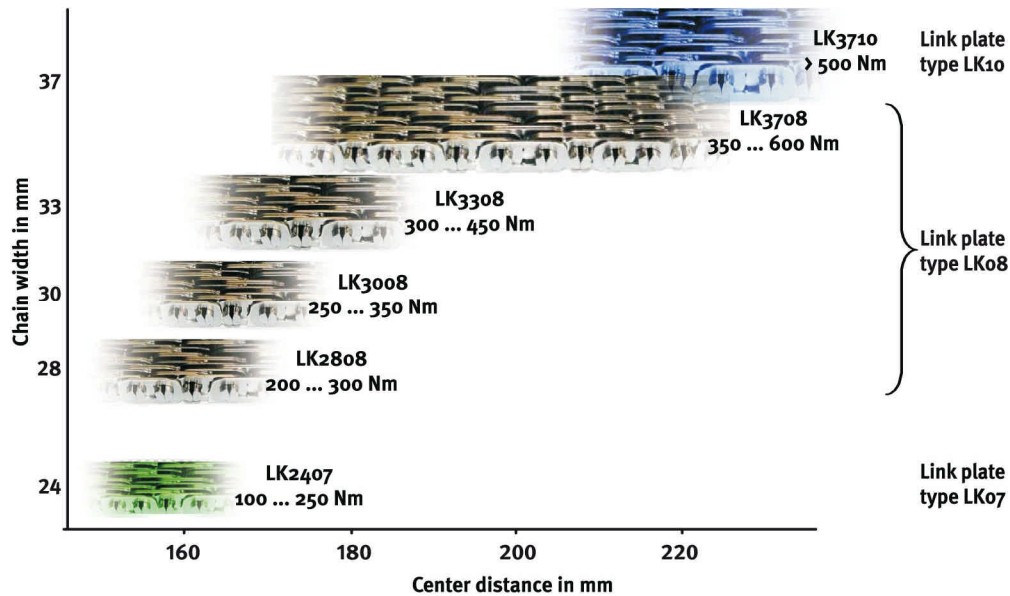


Figure 8 Torque capacity of the variator with different chain types

The suitability in terms of strength for the intended application was proven with commercial vehicle load cycles using damage calculations. A comparison of the force strokes for passenger cars and commercial vehicles (including the reduction achieved through power splitting) is shown in figure 9. The more extensive quantity of force strokes due to the mileage is at a similar force level for both commercial and passenger vehicles. The maximum force strokes of both groups which are only slightly increased despite the significant rise in maximum torque, occur comparably seldom.

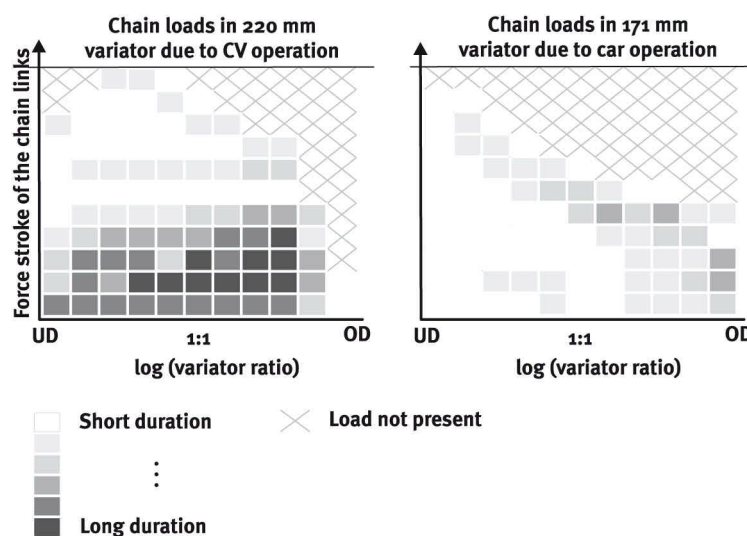


Figure 9 Summary of link force strokes resulting from a load cycle for a commercial vehicle (left) and a passenger car (right)

5.2 Durability of the pulley surfaces

In consideration of an enormous expected lifetime of more than one million km, particular attention must be paid to the subject of wear. However the high efficiency values already indicate low wear values: As a rule, wear requires energy loss. For a CVT as a friction transmission, it is not only the material wear (quantifiable in weight per friction energy) that counts, but also the wear of the friction surfaces (quantifiable for example as a change in friction value or change in roughness).

Based on the materials and test experiences [13] compiled, both aspects are provided in intensive and successful testing not only on high-performance durability test stands but also in the vehicle. Figure 10 shows the results related to the long-term stability of the metallic tribological system. The change in the force-balance value k_p/k_s over several thousand (!) hours is depicted here. k_p/k_s is a very sensitive indicator for changes in the friction values and hence for changes of the tribological contact between the pulley surface and the chain rocker pins.

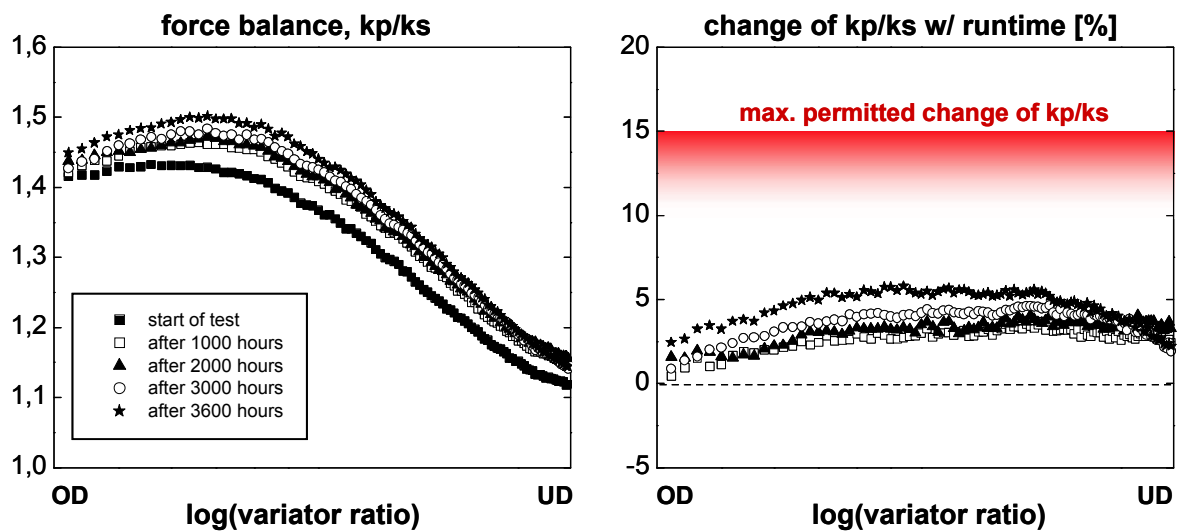


Figure 10

Left: Change of the ratio-dependent force-balance throughout a long-duration test

Right: Relative change of k_p/k_s in % throughout the same test with reference to k_p/k_s at the beginning of the load cycle

Figure 10 shows the evolution of k_p/k_s of the 220 mm chain variator which is exposed to a commercial vehicle load cycle with a maximum variator input torque of 600Nm and an input power of approx 300kW. Shown are the results which have been obtained with the best variant of the tribological system. This means not only an improvement of the chain pitch sequence, of the pulley surface geometry, and of the clamping system, but also an optimisation of the oil performance and the hardening process.

The discernibly low change in the force-balance value shows that the friction coefficient μ as well as the physical properties of the tribological contact have changed only minimally throughout the test. A maximum change of k_p/k_s beyond 10% still would have been acceptable, as known from other projects, hence, the tribological system is very stable and offers noticeably high reserves. It has to be mentioned, that this several thousand hour test consisted of a mixture of different load cycle parts, e.g. full load launches, high temperature cycle, and steady state in overdrive which altogether sum up to a total runtime of approx. 4000h. The total energy which passed through the variator was beyond 600 000 kWh and approx. 120 000 shifts of the automated shift gearbox with fast variator slewing have been performed. The total number of full load launches was approx. 400 000, which were distributed throughout the entire mixed cycle test.

5.3 Durability of the rocker pin end faces

The design of the rocker pin end faces ensures that stresses relevant to the wear, such as Hertzian stress, do not exceed the permitted level. Figure 11 shows as an example the analysis of the local velocity vectors of the rocker pins on the pulley surfaces using LuK's three-dimensional chain simulation tool 'CHAIN'. This program not only takes into account all torque-dependent elastic deformations of the pulley shafts and the rocker pins but also dynamic and kinematic effects in the contact zone of the pins and the pulley surface. Thus, load details are traceable and can be taken into account and minimized already in the design phase of the entire variator.

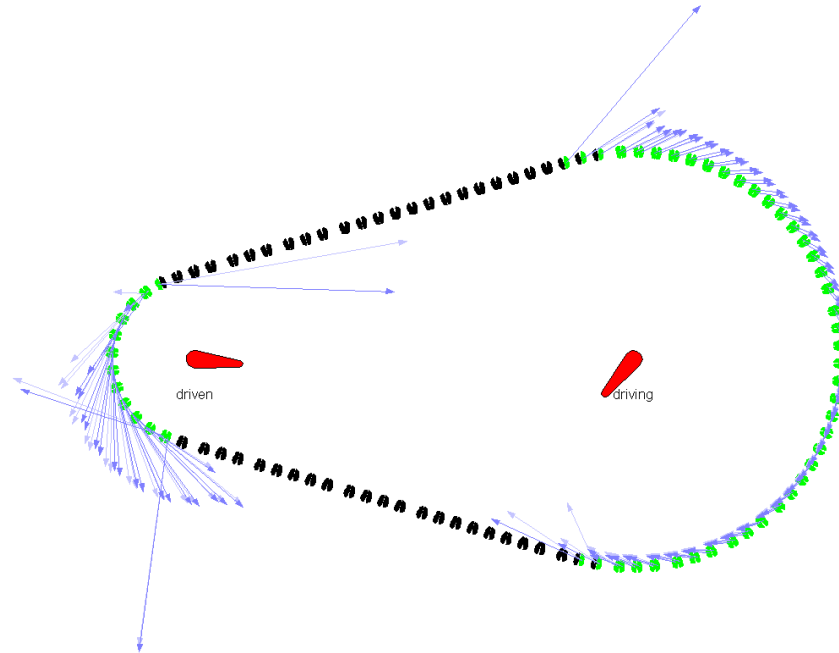


Figure 11 Local velocity vectors of the chain rocker pins determined using the multi-body simulation tool CHAIN. Driving pulley on the right, driven pulley on the left

With regard to any further increase in the running time, optimisation of the chain pitch sequence also gives a contribution, see figure 12. Pitch sequences of long and short links are favourable for acoustic priority, with the direct sequence of two long links specifically excluded. The reason for this is that the greatest end face loads occur experimentally and in calculations at precisely these locations. Chains optimised in such a way do not just show lower wear rates but can also withstand more overall wear because the width reduction in the chain occurs uniformly.

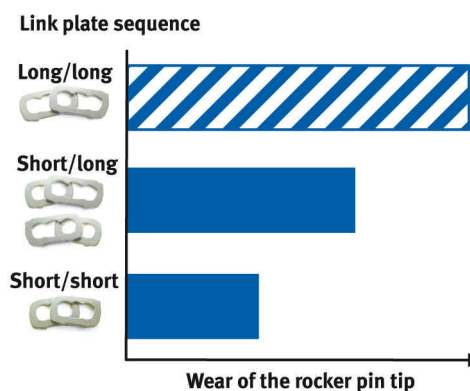


Figure 12 Correlation of the local width wear with the sequence of long and short links. The optimized chain avoids the direct sequence of long/long links

From the experimental point of view, besides k_p/k_s the second endurance aspect is the wear of the end faces of the rocker pins, which leads to a decrease in the chain width, hence reducing the thickness of the hardening zone. In several long-term tests, this end face wear was determined through periodic inspections, the results are shown in figure 13. Besides the results obtained with the best variant, i.e. the lowest curve in the diagram, the evolution of the chain width wear throughout the project is depicted. Taking into account all the measures for the improvement of the tribological contact mentioned above and based on a 4000h mixed cycle test, this leads to a nearly constant chain width wear which is only 10 to 15 % of the maximum permitted width wear value (lowest curve in figure 13). The tribological system of chain, pulley set surface, and appropriate oil used here thus demonstrates overall stable behaviour of the friction values and confirms the achievability of the ambitious service life targets.

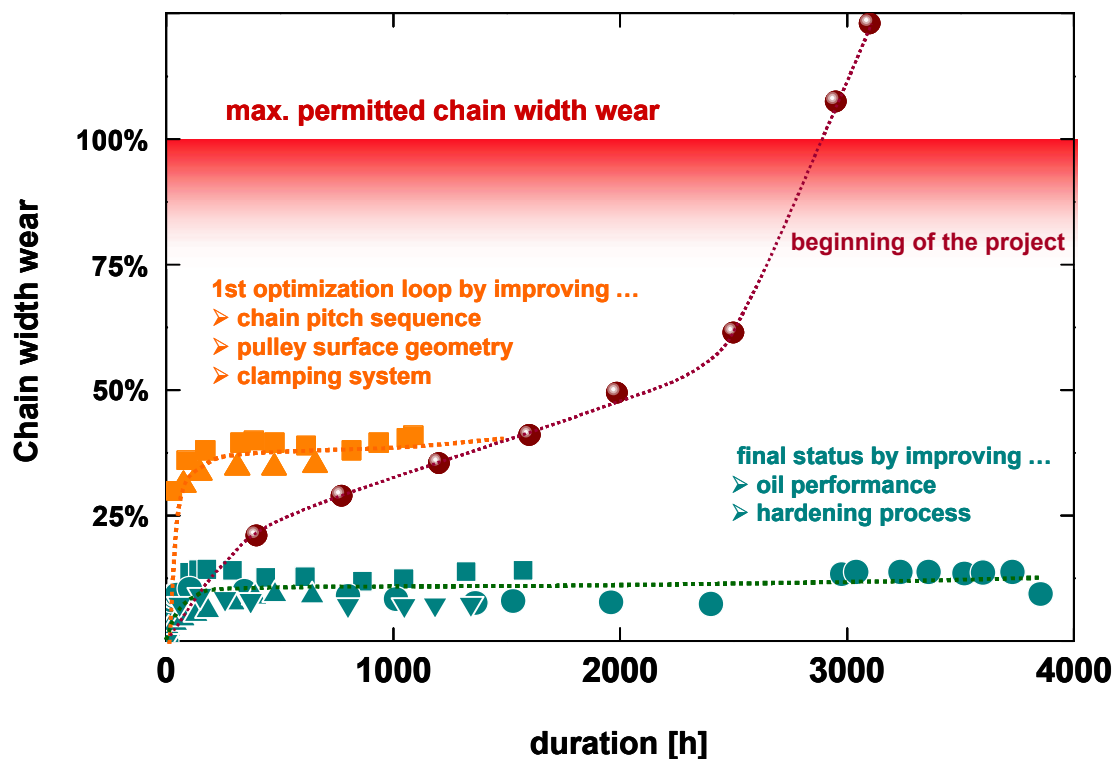


Figure 13 Evolution of the chain width wear throughout the project.

6 Summary

Following the successful production launch of the new Audi VL380 with 420 Nm variator torque which was achieved with the LuK CVT components (pulley sets with the variable torque sensor, optimised chain and hydraulic control with dual-flow vane cell pump), it has been shown that this technology can also be utilised for commercial vehicle applications. Based on a power split architecture and a maximum variator torque of 600 Nm, positive results on both the test bench and in a commercial vehicle are achieved. The challenges with respect to variator and chain design as well as pump and hydraulic development require careful preliminary investigations as well as the use of advanced simulation tools.

The interplay of a great deal of detailed work has now resulted in a promising situation for the application of power split CVT in high torque applications of both commercial vehicles and passenger cars.

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