CVT without limits –

Components for commercial vehicle transmissions

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Introduction

The rise in fuel costs and the necessity for further reductions in emissions require new technical solutions allowing for 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© [1] VL300 and its newer evolution the VL380 with 420 Nm variator torque as well as the Ford/ZF CFT30 [2] which are already in production. Optimisation of the power train also makes sense for buses, vans and commercial vehicles as they are responsible for a considerable proportion of emissions.

A manufacturer of commercial vehicle transmissions applied itself to making 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 development was begun.

Vehicle	Audi A6 3.2FSI / 2.7TDI	Commercial vehicle
Permitted weight in kg	2 200 ... 4 000	10 000 ... 40 000
Max. engine torque in Nm	330 / 380	1 000 ... 3 000
Max. engine power in kW	188 / 132	200 ... 500
Required lifetime in km	300 000	> 1 000 000
Maximum speed in km/h	250 / 225	120
Period of operation in h	> 3 000	> 15 000
Exhaust standards/targets	EU4	Increasing requirement foreseeable
Transmission or variator	Audi multitronic© VL300 / VL380	Power split CVT
Transmission structure	D & Rev. without power split	Several ranges + D & Rev.
Fastest complete variator adjustment in s	1,2	0,65
Max. oil requirement for adjustment in l/min	5	19
Max. variator torque in Nm	-60 ... +350 / -60 ... +420	-350 ... +600
Number of chain rotations during the period of operation in 10^6	300	2 000
Center distance variator in mm	171	220

Figure 1 Data and targets comparing passenger and commercial vehicle utilisation of a CVT

The challenge

If the data of an Audi A6 is compared with the data of various commercial vehicles, the task seems almost impossible. With a whole range of targets, not only is a doubling required but also an increase in the size as shown in figure 1.

As the variator data shows, more than one individual component must be optimised to fulfil these requirements. The following sections deal with all these aspects of transmission structures from pulley sets and pump systems to the chain.

In the course of the development, it is becoming increasingly clear that the challenge of realising a CVT for commercial vehicles is acceptable.

Transmission architecture

One of the key technologies in this CVT application is the principle of power splitting which was also described in the last LuK Symposium 2002 [3, 4] as well as the use of several continuously variable driving ranges. 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. Compared with a wholly electrical power conversion, the benefit of the chain converter in cost, efficiency and power density is even greater.

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.

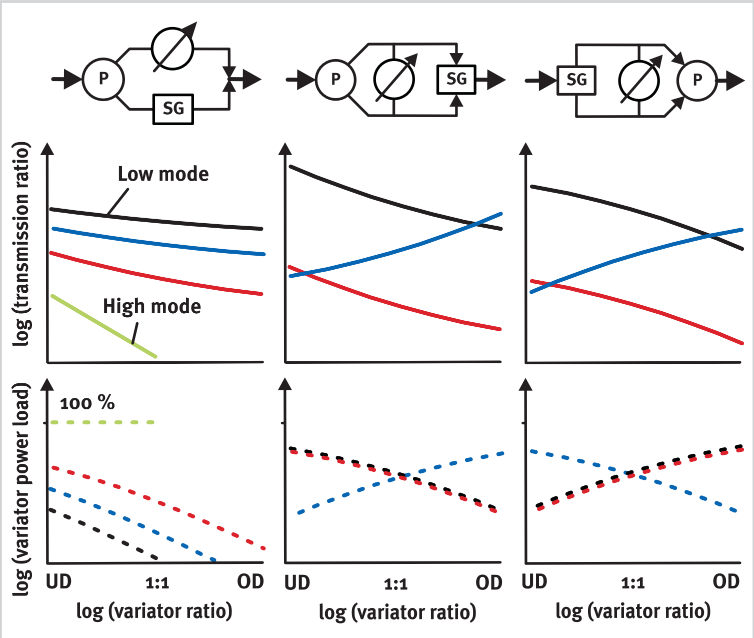

Figure 2 Three alternative transmission architectures with ratio and power load characteristics. Each color represents one of several operation ranges

The options to link a planetary gear and a chain variator together via adapter stages 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. Several options are described in figure 2.

In the architecture shown on the left, the planetary gear is configured on the input side and the multi-range-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.

When driving, the benefit of all the architectures shown lies in the, for commercial vehicles, comparatively low number of gears and therefore range changes. For example, with four driving ranges after start-up only one shift is made and city centre driving can continue virtually without any range changes. Another range change is necessary on leaving the city and a final one on the motorway.

Variator development beyond 500 Nm

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 ... 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 ↑	Forces per rocker pin (RP) or joint (RJ) ↓↓
Chain rocking angle ↓	Better load distribution on chain link cross section
Force per rocker joint ↓↓	Deflection of the RJ ↓, Link load on chain edge ↓↓
Shaft diameter ↑	Bending stiffness ↑↑, Efficiency ↑
Crowning of the surface ↓	Stress ↓, Wear ↓, Wear per RJ ↓↓

The combination strengthens several strain elimination effects so that the maximum 600 Nm starting from the 420 Nm already realised in the mass-produced VL380 is very possible with clean design of all such aspects.

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 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

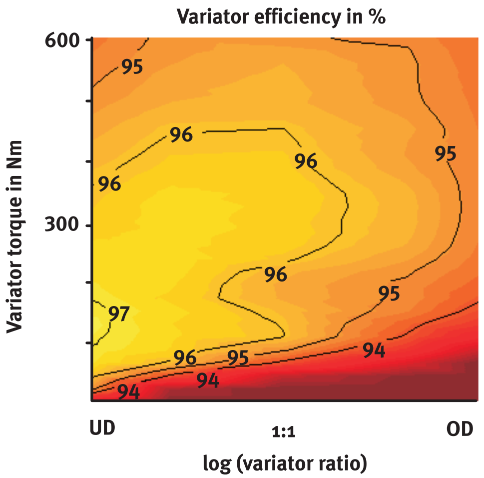

Figure 3 Chain variator efficiency characteristic diagram

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 as an example the 600 Nm efficiency characteristic diagram with a constant drive speed of 2500 min-1.

Shown are the raw measurements data (except smoothing for noise suppression) from the test transmission including the losses of the self aligning pulley bearings. Due to its design with replaceable pulley discs, the stiffness of the test transmission is below the target stiffness. In reality, the efficiency could therefore be even higher.

The center distance of 220 mm thus shows the expected positive effect.

Continuously variable hydromechanical torque sensor

The pulley technology in use on the Audi VL380 is the basis for use at even greater torques [9]. Of central importance is the space-saving continuously variable torque sensor (VTS) [14] implemented inside the pressure chamber, shown in figure 4 in production design with stamped sheet metal parts.

The VTS 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 jumps in the torque

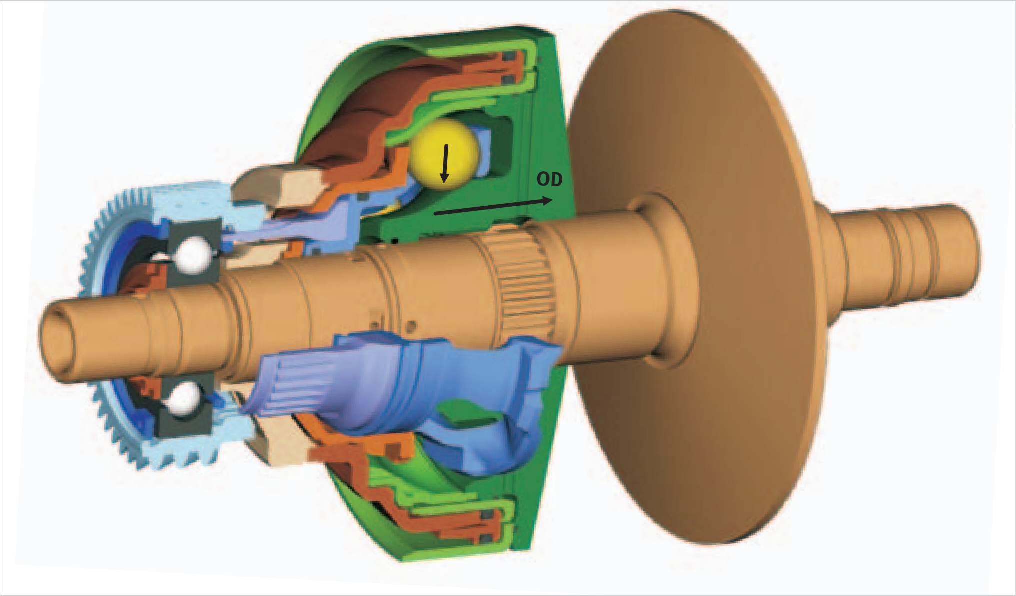

Figure 4 Driving pulley set with VTS continuously variable torque sensor for Audi VL380 with 420 Nm variator torque

The function of the continuously variable sensor is to convert the torque introduced along the components shown in blue in the figure via ball ramps into an axial force. These balls shown in yellow are arranged inside the mechanism. The torque then enters the brown component on the shaft via the opposite ramp and from there moves on to the fixed sheave or via the teeth to the moveable sheave components coloured in green.

However, the axial force produced by the balls does not have a direct effect on the cone pulley but, with the blue component, 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 green moveable sheave via large clamping areas. The same pressure is also supplied to the driven shaft for clamping.

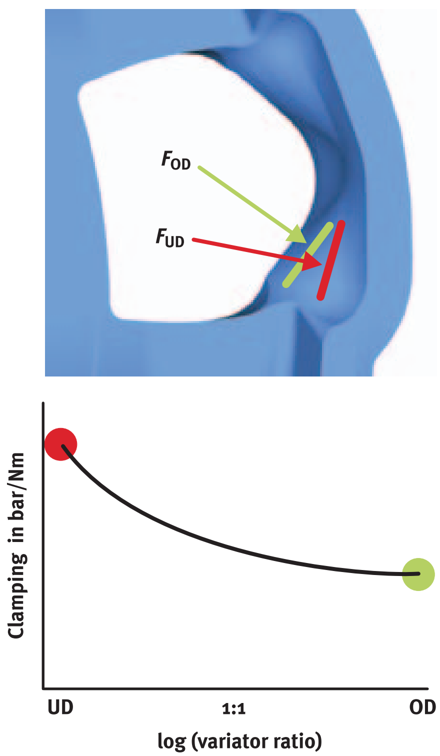

Figure 5 VTS ramp contour with ratio-dependent slopes and resulting clamp characteristic

The dependency of the ratio is thus achieved insomuch as 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 (also shown in green in figure 4).

The detailed enlargement of the blue component with the various slopes ramps, figure 5 is used to illustrate the ratio dependence of the continuously variable torque sensor.

In the result, the pressure related to the torque is the greatest in underdrive, declining continually till 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 6.

In the measurement shown, a step in the drive torque 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. Quantitatively, the adjusted pressure at each instance is also congruent with the target pressure calculated from the measured torque.

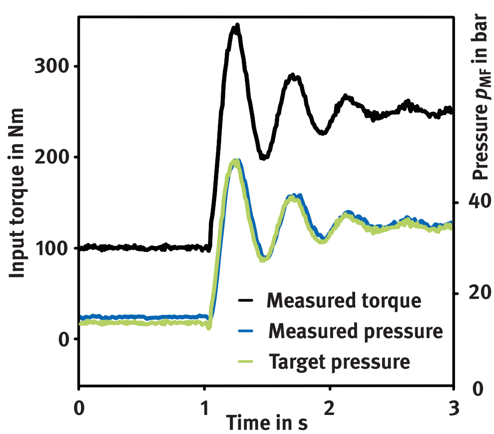

Figure 6 Promptness of the VTS clamp system when exposed to a step in the torque

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. It forms the basis for the following hydraulic and pump development.

Independent actuation of the adjustment pressures

Despite upscaling the entire system and the piston surfaces, the considerably higher coast torque due to engine brake assemblies, as only one reason cause an increase in the required peak 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 in actuation precision.

The tasks of the clamping and adjustment system and the solution implemented by this hydraulics are clearly illustrated in figure 7. For steady operation, the variator requires two forces on the pulley sets which are in a particular proportion dependent on the ratio, the so-called force-balance Zeta ζ or also Kp/Ks. In driving mode, the force-balance value for the LuK CVT chain is 1.05 (UD) to max. 1.6 (OD). In coasting mode, it is the reciprocal force-balance-value with inverse ratio, i.e. around 0.95 (OD) to 0.6 (UD). Respective to the required force-balance, the pivot point of the rocker shown in grey may be thought of as displaced. The task of the clamping pistons shown in red is to generate the basic clamping forces on both pulley sets, whereupon high-pressure oil is exchanged between the pistons during the variator adjustment. The task of the small adjustment pistons shown in orange is to generate the additional residual forces and adjustment forces required for equalisation.

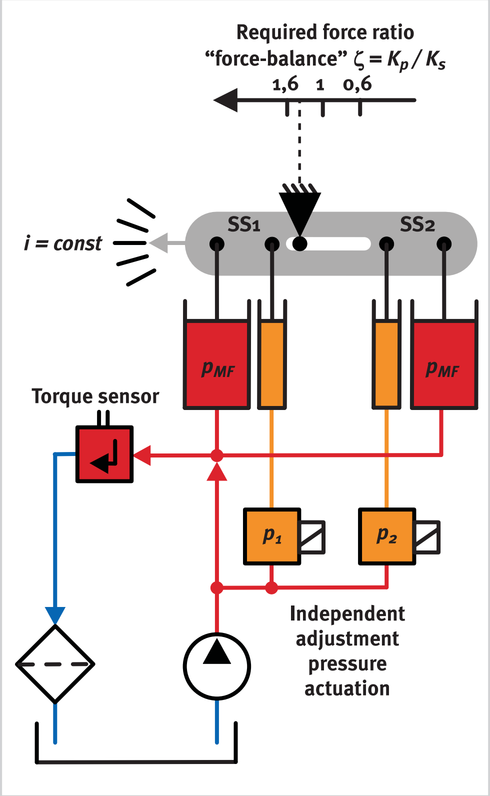

Figure 7 Double piston principle and independent adjustment pressure actuation to balance the required forces in the variator

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.

When adjusting the ratio, the forces balance is left 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 and the independent pressure control of the adjustment chambers offers the freedom of software-controlled free clamping force. A combination with a slip control [10] of the clamping force is also possible with this system, without losing the benefits of the torque sensor.

Pumps and cooling system

The lubricating oil and cooling requirements of the clutch, variator, gears and bearings make the use of a low-pressure pump advisable in transmissions with a power classification in excess of 400 kW. Following identification of the design-relevant operating point (fully loaded hill start with maximum clamping force and clutch cooling), the result for the intended gearbox-architecture is a low-pressure, gerotor-style pump with 29 cm^3 delivery volume.

The low-volume high-pressure pump required, for example, for the clamping force of the variator is designed as a symmetrically divided, dual-flow, fully compensated vane pump with a delivery volume of 10 cm^3 in total. A similar pump developed by LuK is also in production in the Autotronic© from DaimlerChrysler [11].

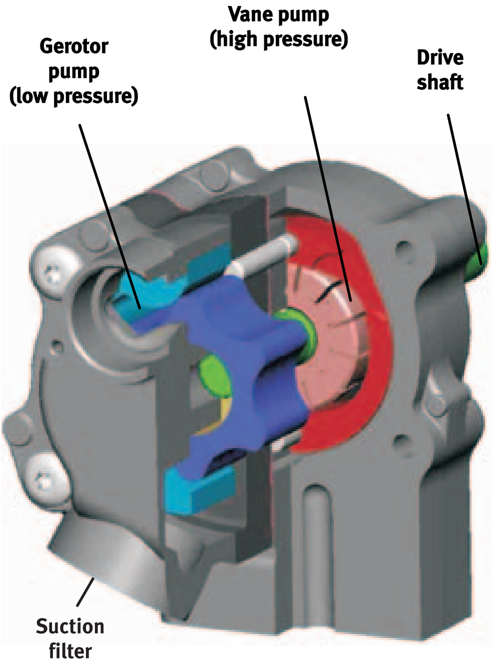

Figure 8 Hydraulic tandem pump from the low-pressure gerotor assembly (blue) and dual-flow high-pressure vane pump (red)

Combined with the low-pressure pump, it forms a tandem pump as a unit, figure 8, 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.

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 controls. Its function is explained in figure 9.

The position of the flow control valve illustrated on the left shows that the second flow of the vane-cell pump is switched to circulation. The pump's drive torque is thus drastically reduced in the majority of driving situations. Note that this oil is not lost to the low-pressure consumers for lubrication and cooling.

In the section of the diagram repeated on the right, 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 the fastest adjustments of the variator, e.g. when starting up or when changing driving range.

Thanks to the optimised design, the pump losses of this transmission are low, as is also the case for the multitronic© and 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.

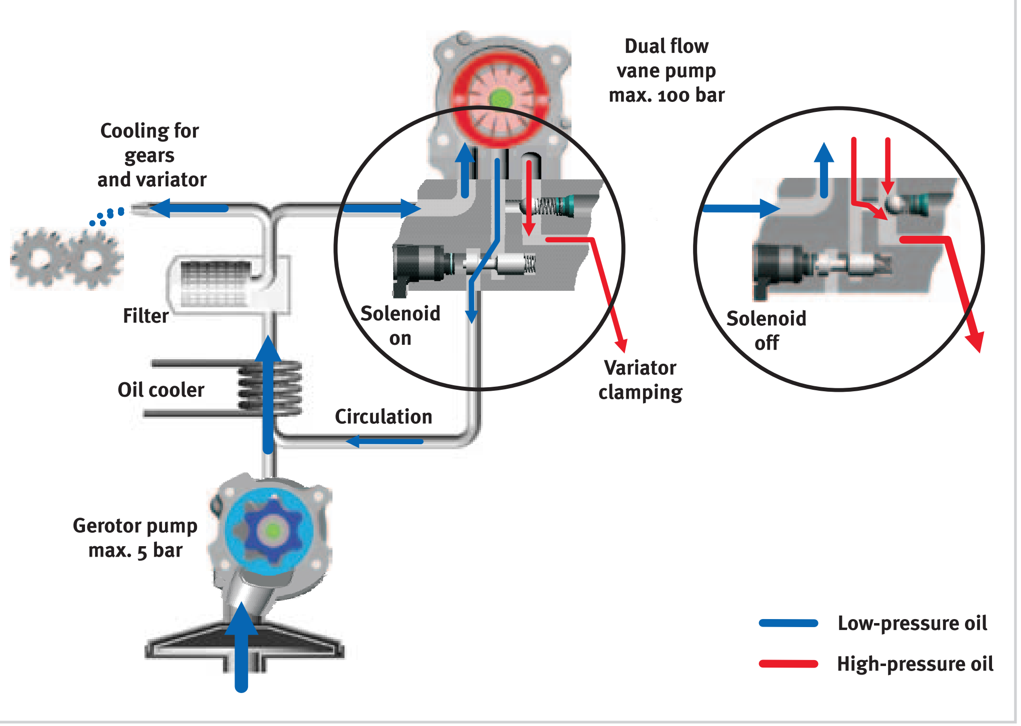

Figure 9 Control of the high and low-pressure volume flows for different devices

Precautions for long-term use

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

The CVT chain

Strength

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, strengthening and torque capacity described in LuK Symposium 2002 [12] is even exceeded with large center distance.

The range of chains in figure 10 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.

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 and commercial vehicles (including the reduction achieved through power splitting) is shown in figure 11. 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.

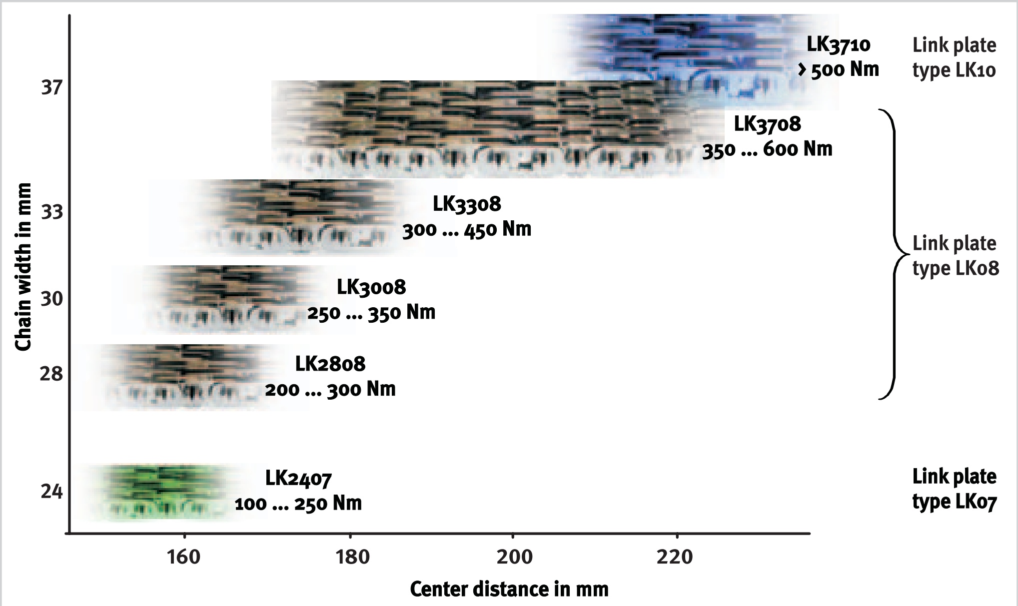

Figure 10 Torque capacity of the variator with different chain types

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 quality 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 on several high-performance durability test stands. Figure 12 shows a few intermediate results relating to the long-term stability of the metallic tribological system. The change in the force-balance value ζ over several thousand (!) hours is depicted here. The force-balance value is a good indicator for changes in the friction values.

The upper half of figure 12 shows the results, in black, of a

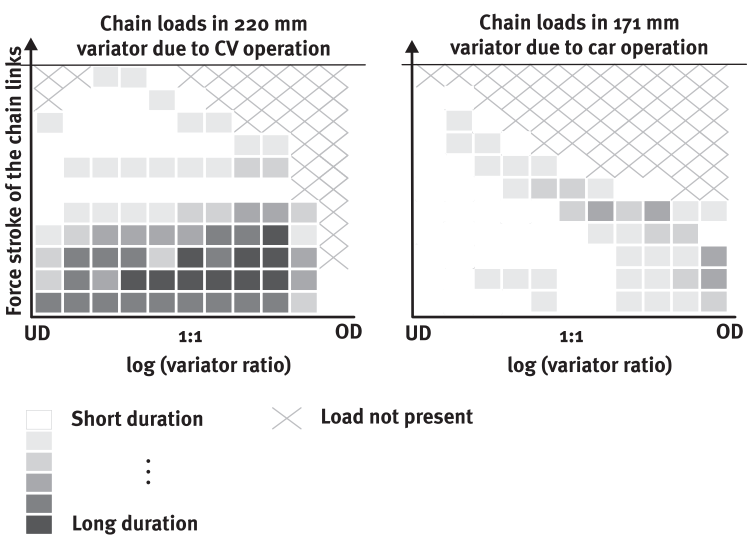

Figure 11 Summary of the link force strokes for passenger and commercial vehicles

reference car application with a center distance of 171 mm. The discernibly low change in the force-balance value ζ shows that the friction coefficient μ has only minimally changed. The targeted service life of 3000 h for a passenger car can be checked using a condensed test procedure within approx. 150 h. The tribological system is very stable and offers noticeably high reserves.

Shown in blue are the results of the same assembly with a center distance of 171 mm, but which has been exposed to the scaled loads of the commercial vehicle application. This scaling means that the torques were reduced in such a way that the resulting pin-specific forces correspond to a 220 mm system. This intermediate step contingent upon the trial method already supports the service life potential of a center distance enlargement. The targeted service life of more than 15000 h of a commercial vehicle could be achieved using a condensed test procedure of 1000 h duration. The test was even extended to a running time of 2500 h as a safeguard. The different symbols thereby indicate different test procedures, both of which have been run. In the procedure represented by a square, the ratio is changed in several stages so that a high load concentration is exerted on the rocker pin but not on the pulley. The test procedure represented by a diamond takes place with a fixed ratio so that the contact loads are concentrated both on the rocker pin and on the pulley (however both friction partners can thus also be well adjusted to each other). The results of the variator tested with a fixed ratio once again confirm that mixed cycles represent the most rigorous and therefore most efficient test method.

Shown in green are the results of a true 220 mm variator which is being exposed to the unscaled commercial vehicles load cycle applying up to

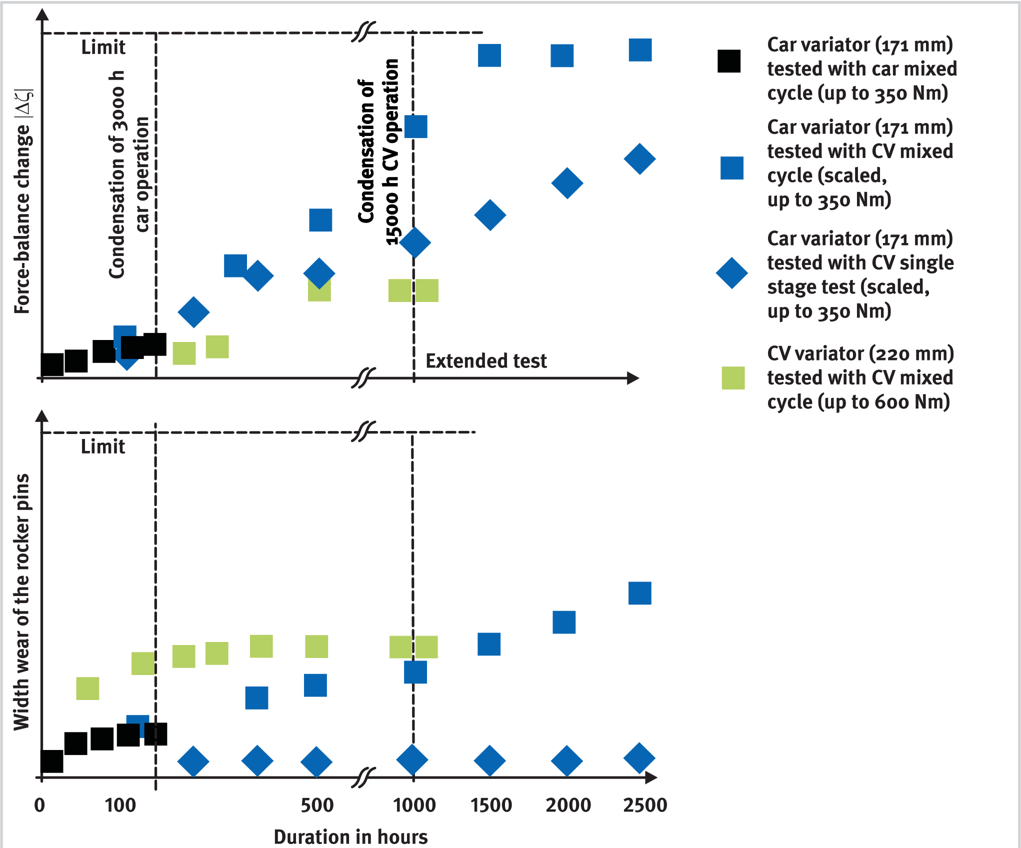

Figure 12 Top: Change of the force-balance of different variators in durability tests.
Bottom: Width wear of the rocker pins in these durability tests.

600 Nm. The initially fast change in force balance as well as the high width wear at the beginning (lower half of the figure) are the result of the highest load points being tested first in the test program. The stable behaviour is thereby also the result of a pitch sequence optimisation of the chain to be described.

The tribological system of chain, pulley set surface and appropriate oil used here in the chain variator demonstrates overall stable behaviour of the friction values and confirms the achievability of the ambitious service life targets.

Durability of the rocker pin end faces

The second endurance aspect is the wear of the end faces on the chain side, which leads to a decrease in the chain width. In the long-term tests shown above, this end face wear was determined with periodic inspections. These results are shown in the lower half of figure 12 using the same colour and symbol selection.

The overall low end face wear is a characteristic of optimised heat treatment.

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 13 shows an analysis of the contact points and characteristics on these end surfaces, calculated using the three-dimensional chain calculation program 'CHAIN', which also takes into account all elastic deformations from the shafts to the rocker pins when doing so.

The coloured areas are the contact ellipses, the stress of which is visualised using the colour. This calculation using LuK's CHAIN calculation program takes into account all elastic and dynamic effects on pulleys and chain as well as the joint kinematics of the rocker pins. Thus, load details are traceable and can be taken into account in the chain design.

With regard to any further increase in the running time, optimisation of the pitch sequence can also make a contribution, figure 14. 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.

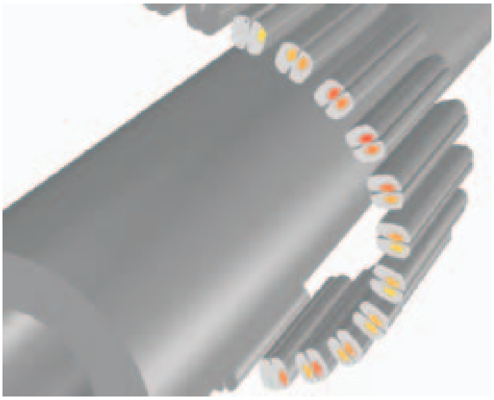

Bild 13 CHAIN screenshot with visualisation of the contact properties ratio in underdrive

Summary

Following the successful production launch of the Audi VL380 with 420 Nm variator torque, achieved with the LuK CVT components (pulley sets with the novel, continuously variable torque sensor, optimised LK3308 chain and hydraulic control with vane cell pump) ways are being sought in which this technology can also be utilised for commer-

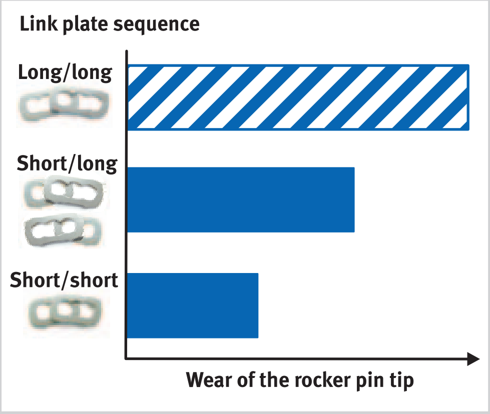

Bild 14 Correlation of the local width wear with the sequence of long and short links. The optimised chain avoids the direct sequence of two long links.

cial vehicle applications with power splitting and variator torque up to 600 Nm. The challenges with respect to structural development, variator design, pump and hydraulic development as well as chain design required careful preliminary consideration and the courage to act which is rewarded by the positive test results.

The interplay of a great deal of detailed work has now resulted in a promising situation for the application of power split cvt in commercial vehicles as well as in cars with an increased torque.

The first prototypes of a special transmission structure are being built in collaboration with an established customer in the commercial vehicle sector.

References

[1] Fleischmann, H.-P., Gutz, H.; Kumpf, G.; Martin, F.; Schöffmann, M.: Die neuen Getriebe im Audi A6. ATZ 106(2004), Sonderheft Audi A6, p. 128-138.

[2] Wagner, G.; Remmlinger, U.; Fischer, M.; CFT30 – A Chain Driven CVT for FWD 6 Cylinder Application. SAE Technical Paper Series 2004-01-0648.

[3] Lauinger, C.; Vornehm, M.; Englisch, A.: Das 500 Nm CVT. 7. LuK Symposium (2002), p. 91-106.

[4] Englisch, A.; Lauinger, C.; Vornehm, M.; Wagner, U.: 500 Nm CVT – LuK Components in Power Split. CVT 2002 Congress, Munich, 7/8 October 2002, VDI Reports No. 1709 (2002), p. 147-163.

[5] Tenberge, P.; Müller, J.; Sewart, J.: CVT für höchste Drehmomente – CVT mit Umschlingungsvariator und Leistungsverzweigung. Getriebe in Fahrzeugen 2004, Friedrichshafen, VDI Reports No. 1827 (2004), p. 669-712.

[6] Schumacher, A.; Harms, H.: Potential von leistungsverzweigten Getrieben in leichten und schweren Nutzfahrzeugen. Nutzfahrzeuge 2005, Böblingen, VDI Reports No. 1876, p. 63-78.

[7] Sattler, H.: Abschlußbericht Forschungsvorhaben No. 221 "CVT Wirkungsgrad"; Forschungsvereinigung Antriebstechnik e.V. (Publ.), Frankfurt.

[8] Lebrecht, W.; Ulbrich, H.: Vergleich von CVT-Umschlingungsgetrieben. 4th International CTI Symposium of Innovative Vehicle Transmissions, Berlin 5.12.2005, Lecture/Article E6.

[9] Englisch, A.; Faust, H.; Homm, M.; Teubert, A.; Reuschel, M.; Lauinger, C.: Entwicklungspotentiale für stufenlose Getriebe. ATZ 103(2003), Issuet 7/8, p. 676-685.

[10] Faust, H.; Homm, M.; Reuschel, M.: Efficiency Optimised CVT Hydraulic and Clamping System. CVT 2002 Congress, Munich, 7/8 October 2002, VDI Reports No. 1709 (2002), p. 43-58

[11] Greiner, J.; Kiesel, J.; Veil, A.; Strenkert, J.: Front-CVT Automatikgetriebe (WFC 280) von Mercedes-Benz. Getriebe in Fahrzeugen 2004, Friedrichshafen, VDI Reports No. 1827 (2004), p. 421-445.

[12] Indlekofer, N.; Wagner, U.; Fidlin, A.; Teubert, A.: Neueste Ergebnisse der CVT-Entwicklung. 7. LuK Symposium (2002), p. 63-72.

[13] Linnenbrügger, A.; Baumann, M.; Endler, T.: High Performance Chain CVTs and their Tribological Optimisation. Tribology of Vehicle Transmissions 2005, Tsukaba/Japan 16/18 February 2005, Proceedings p. 14-19.

[14] Englisch, A.; Faust, H.; Homm, M.; Teubert, A.; Vornehm, M.: Hochleistungs-CVT-Komponenten. Getriebe in Fahrzeugen 2004, Friedrichshafen, VDI Reports No. 1827 (2004), p. 649-668.