7th LuK Symposium

11./12. April 2002

Publisher: LuK GmbH & Co.
Industriestrasse 3 • D -77815 Bühl/Baden
Telephon +49 (0) 7223 / 941 - 0 • Fax +49 (0) 7223 / 2 69 50
Internet: www.LuK.de

Editorial: Ralf Stopp, Christa Siefert

Layout: Vera Westermann
Layout support: Heike Pinther

Print: Konkordia GmbH, Bühl
Das Medienunternehmen

Printed in Germany

Foreword

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

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

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

We look forward to some interesting discussions with you.

Bühl, in April 2002

Helmut Beier

Helmut Beier

President
of the LuK Group

Content

500 Nm CVT

LuK Components in Power Split

Christian Lauinger
Martin Vornehm
Andreas Englisch

Introduction

Following the successful market launch of the Audi multitronic® [1], [2], presently for applications up to 310 Nm, a general demand currently exists for continuously variable transmissions for more powerful engines. The target here is for around 500 Nm of engine torque.

When developing a CVT concept (Continuously Variable Transmission) for applications with considerably higher torque, along with numerous other aspects, a fuel consumption-optimised transmission structure and the variator load are of particular importance during the selection and evaluation process. The limiting factor for a further increase in the maximum transmittable torque is the transmitting element.

According to [3], on a non-power-split conventional CVT with the further-developed LuK chain, the torque, at a total ratio spread of 6 and axis distance of 171 mm, is currently limited to approximately 400 Nm. Smaller spreads would have adverse effects on fuel consumption.

Figure 1 gives an overview of the total ratio spread and the maximum engine torque of a conventional CVT with various transmission elements. Currently, the most demanding application is the LuK variator in the Audi multitronic® with a maximum engine torque of 310 Nm and a total ratio spread of 6. Potential for a torque increase to nearly 400 Nm through the further-developed LuK chain [3] of the same size is also indicated. Further increases in engine torque up to 500 Nm with spreads between 6 and 7 require structural enhancements such as the arrangement of the variator in a power split drive train.

Within the scope of this theoretical study, two possible concepts for a power split, dual-range CVT were investigated to ascertain whether the step to the required 500 Nm is possible within the specified range of total ratio spread.

A dual-range CVT has, compared to a conventional CVT, the advantage that both the variator torque and the variator ratio spread required for a given total ratio spread can be reduced. Both result in a reduction of variator load, which allows for an increase of the transmission torque capacity and consequently the engine torque and total ratio spread.

In the following chapter, the various design possibilities for a dual-range transmission will be considered with regard to their suitability and subsequently evaluated.

Thereafter, the particular requirements regarding the clamping system and the hydraulics will be discussed. Finally, a possible control strategy will be illustrated. The focus here will be on the development of a strategy for a comfortable mode change.

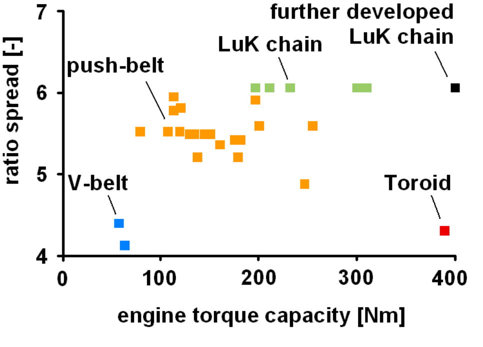

Fig. 1: *Maximum Permissible Engine Torque and Spread of Standard CVT with Various Transmitting Elements*

Mechanical

Dual-range CVT

In CVT's currently on the market, the variator is arranged in a conventional drive train, a so-called single-range CVT. The torque capacity has already been described in the introduction. Therefore, an increase in engine torque up to 500 Nm with simultaneous increase of the total ratio spread requires structural transmission enhancements such as a dual-range CVT.

In its simplest form, a dual-range CVT has a second fixed-ratio shaft arranged in parallel with the variator. The two branches are brought together at the output shaft through a summation gear set (e.g. a simple planetary gear set). The drive train, together with the variator, can be driven either directly or in a power split mode by disengaging or engaging a clutch in the fixed-ratio section. A dual-range CVT is therefore more elaborate than a single-range CVT from both a mechanical and control point of view.

Dual-range CVT designs are also possible where the variator is permanently located in a power split drive train; however, to reach the same total ratio spread the designs of the summation gear sets are more complex.

The literature contains references to other known multi-range concepts [4], which were not included in this study.

The illustration in figure 2 shows the design and operation of a dual-range CVT. The summation gear set is a simple planetary gear set with negative set ratio i_1 (negative gear set). On a negative gear set, when the carrier is held, the sun and the ring gear turn in opposite directions [5]. The clutch in the fixed-ratio split has the designation K_H. In order to limit the degrees of freedom of the planetary gear set when operating in unsplit mode, a further clutch K_L, which connects the sun gear and the carrier, is necessary. The planetary gear set then rotates locked-up and therefore has a fixed ratio of 1. For simplicity the schematic drawing does not show the drive-off element (clutch or hydrodynamic torque converter) and the axle differential. An additional brake B, which facilitates braking of the planet carrier, is required for the reverse gear. The output shaft is connected to the ring gear.

In unsplit mode, K_L is closed and K_H is open. In power split mode, the relationship is reversed. The point where switch-over of the clutches takes place and which separates the two ranges is designated as the mode change point.

The concept from figure 2 makes it possible to achieve power split either during drive-off (higher driving range is then unsplit) or in the higher driving range (drive-off is then unsplit). To do so, the fixed ratios and the planetary gear set ratios must be changed and the actuation of the clutches K_H and K_L interchanged. Additional clutches are not required. In the following, the transmission from figure 2

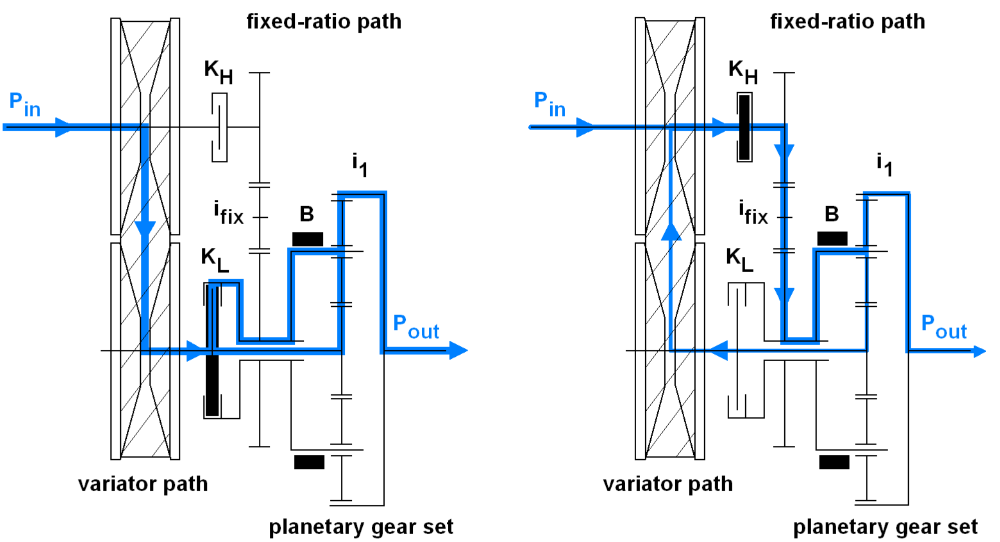

Fig. 2: *Schematic Layout of a Dual-Range CVT Showing Power Flows Unsplit (Left) and Power Split (Right)*

is shown unsplit in the lower driving range (from now on called LOW range) and power split in the higher driving range (HIGH range).

Figure 2 additionally shows a schematic view of the power flow through the variator in unsplit and power split modes. In the unsplit mode, figure 2a, the entire engine power flows through the variator. When switching over to power split, a reactive-power flow forms in the split transmission section as shown in figure 2b. Consequently, the variator torque changes its sign, i.e. the variator is driven from the secondary pulley set. The extent of the power flow through the variator is dependent on the total ratio and can be analysed as follows. When the drive-off clutch / torque converter clutch is engaged, the engine and variator input speeds are identical. The power is calculated from the product of speed and torque with the result that the ratio between variator torque and engine torque is identical to the ratio between variator power and engine power. On a single-range CVT, this ratio is equal to 1. In power split applications, this is no longer the case.

Figure 3 shows the ratio of variator torque to engine torque depending on the total ratio. In unsplit mode, i.e. between total transmission underdrive (UD) and mode change, the ratio is constant and equal to 1.

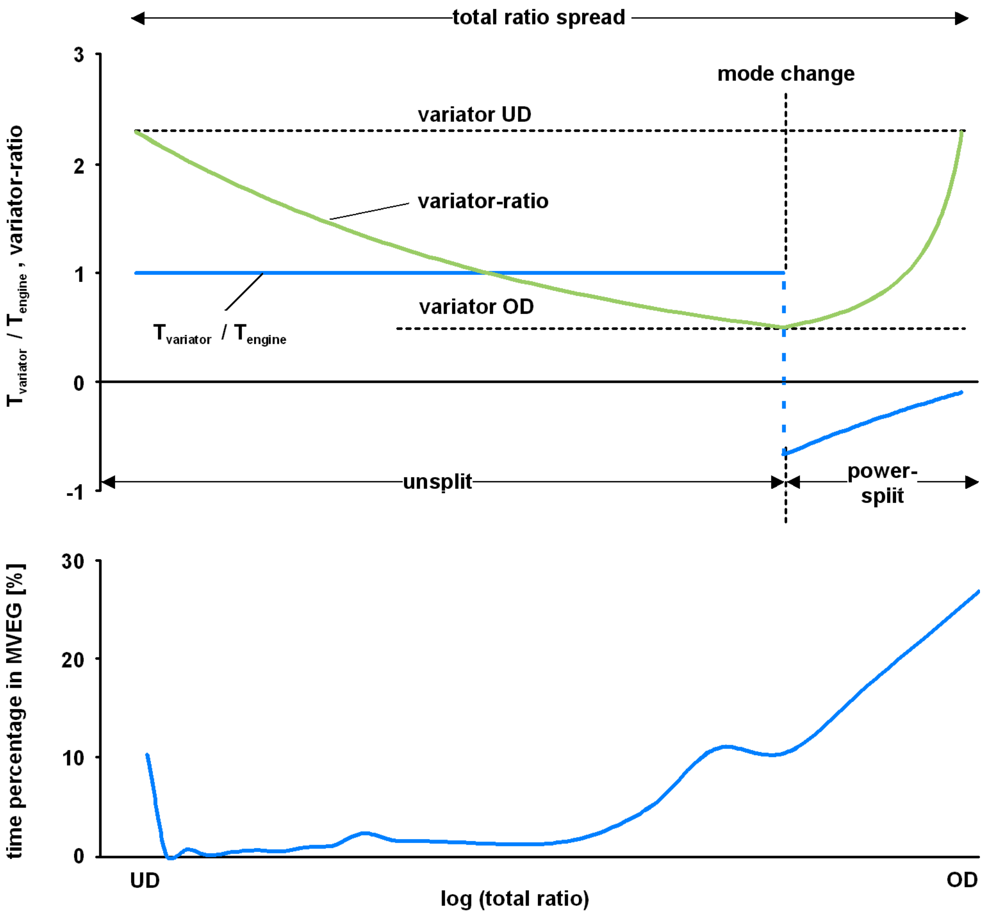

Fig. 3: Ratio of Variator Torque to Engine Torque and Variator Ratio (Top) and Drive-Time Distribution in MVEG (Bottom) in Relation to the Total Ratio

With mode change into power split, the sign of the variator torque changes due to the reactive power present, see figure 2b. As the total ratio moves further towards total transmission overdrive (OD), the ratio of variator torque to engine torque continually reduces. On the arrangement shown here, the amount of variator torque in total transmission overdrive (OD) is only around 10% of the engine torque. Consequently, a reduction of the mechanical load of the variator is achieved at these operating points.

The reduction of power flow through the variator in power split mode results in an improvement of the transmission efficiency with a consequent positive effect on fuel consumption. Figure 3 shows the drive-time distribution in MVEG. In the out-of-town segment (EUDC), the variator operates predominantly in power split mode. The variator is then only subject to low torque levels with consequently low losses. This results in an overall improvement in fuel consumption. The hydraulic pressure for the clamping force and consequently for the pump [6], [7] can be reduced in proportion to the applied variator torque.

If a dual-range CVT is used, the variator spread can be reduced. Figure 3 also shows the variator ratio over the total ratio. Between drive-off and mode change, the variator ratio changes from variator UD (greatest variator ratio) to variator OD (smallest variator ratio). As the total ratio moves further from mode change towards total transmission OD, the variator adjusts in the opposite direction back towards variator UD. Because the ratio range of the variator is run through twice, the variator spread can be reduced in comparison with a single-range CVT of the same total ratio spread. In this illustration, the reduction amounts to 34%.

The reduced variator spread in the dual-range CVT results in lower chain loads. This is illustrated in figure 4 where the normalised chain running radii on pulley set 2 are compared for conventional and dual-range CVT.

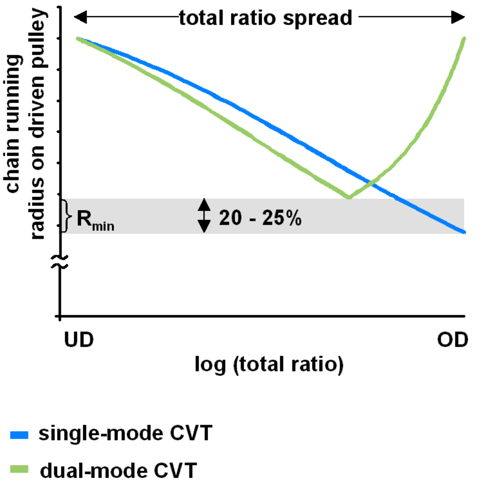

Fig. 4: Comparison of Standard Chain Running Radii on Single and Dual-Range CVT

By limiting the variator spread, the minimal chain running radius is increased by roughly 20 - 25% with a corresponding reduction in chain pulling forces. Consequently, the minimum number of rocker pins engaged in the variator is correspondingly increased and the individual rocker pins have a lower stress level due to a decrease in bending. This results in a more constant force distribution in the link plates [8] with less load on the chain.

This reduction in the variator torque in HIGH range and the lower load on the chain due to the reduced variator spread means that the variator is subject to less load in a dual-range CVT. Using a load cycle with a maximum torque of 500 Nm, the variator in a dual-range CVT has a life time around 10 times longer than in a single-range CVT.

The results shown so far are for summation gear sets arranged as negative gear sets. In principle, a dual-range CVT can also be achieved with a planetary gear set with a positive set ratio [5] (positive gear set). On a positive gear set, when the carrier is held, the sun gear and the ring gear turn in the same direction, e.g. reverse gear set from a standard transmission. If a positive gear set is used, the variator split, fixed-ratio split and output shaft must be connected to the planetary gear set in a different combination. In this case, the sun would be the output, which would be unfavourable for design reasons. A further disadvantage of positive planetary gear sets is a lowered efficiency due to the larger number of rotating gears [5] and the resultant higher fuel consumption. Furthermore, due to the higher number of planet gears the costs and complexity of positive gear sets are greater. For these reasons, only negative gear sets are considered in this study.

Range concepts

Generally, on a dual-range CVT, three different range concepts are possible:

- Unsplit in drive range LOW and power split in HIGH (as in figure 2)
- Power split in LOW and unsplit in HIGH (generally geared-neutral-capable)
- Power split in both drive ranges

The last mentioned variant is significantly more elaborate than the first two concepts due to the greater number of clutches and/or planetary gear sets. It was also explained in the previous section that a reduction in variator load can be achieved in a dual-range CVT due to a smaller variator ratio spread and due to reduced variator torque through the power split. In view of these properties only the first two concepts listed will be considered.

The negative ratios of the planetary gear sets in the two concepts are different. The ratios were selected with the aim of minimising the variator load for each variant at the specified dimensions.

Figure 5 shows the normalised chain running radii on pulley set 2 depending on the total ratio. Figure 5a shows the curve for the concept with power split in HIGH. These are the same values as in figure 4. Figure 5b shows the same curve for the concept with power split in LOW.

As figure 5b shows, during drive-off with power split in LOW the variator adjusts from variator OD towards variator UD. The adjusting direction of the variator is therefore opposite to that of the concept with power split in HIGH. The illustration shows that the severely damage-prone smallest running radius in the concept with power split in LOW can be increased by up to 35%.

Figure 6 shows the ratio of variator torque to engine torque depending on the total ratio for both concepts. The concept with power split in HIGH is shown on the left (figure 6a), power split in LOW is shown on the right (figure 6b). In total transmission UD the variator torque with power split in LOW is several times greater than that of the other concept. This results in considerably higher loading for the variator and chain than on the concept with power split in HIGH.

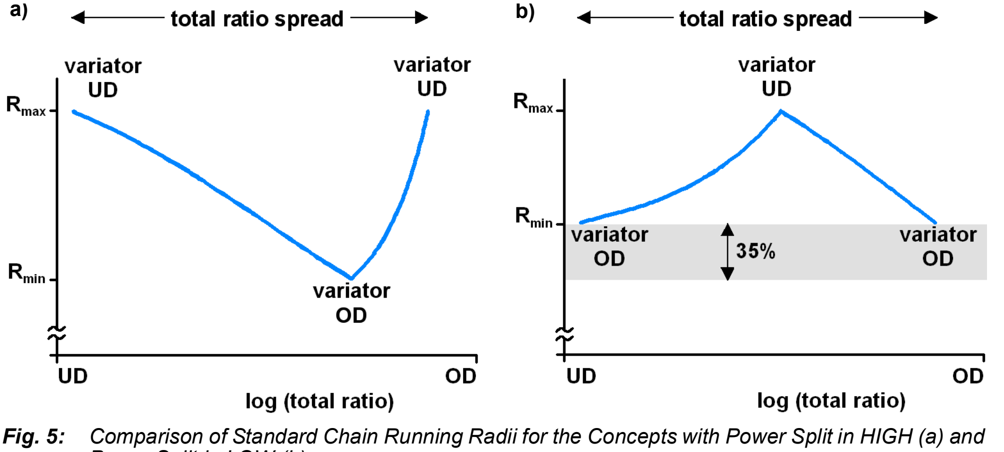

Fig. 5: Comparison of Standard Chain Running Radii for the Concepts with Power Split in HIGH (a) and Power Split in LOW (b)

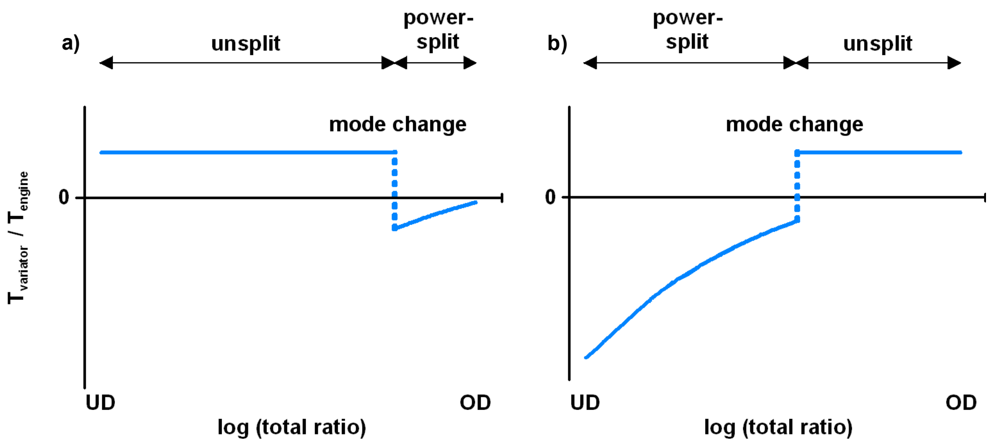

Fig. 6: *Relationship of Variator Torque to Engine Torque Depending on Total Ratio for the Concepts with Power Split in HIGH (a) and Power Split in LOW (b)*

The damage accumulation calculations for both concepts show a higher chain load for the concept with power split in LOW. This means that the benefits resulting from the reduced variator spread are overcompensated for by the high variator torque.

The variant with power split in LOW also has disadvantages with regard to fuel consumption. In the main driving ranges of MVEG, the variator operates conventionally; i.e. the variator torque is identical to the engine torque. The variator loss is consequently higher and the efficiency respectively lower. Furthermore, the hydraulic pressure for the clamping force and the pump cannot be lowered, which is possible with the concept power split in HIGH. Overall this results in improved fuel consumption with power split in HIGH.

The results show that out of the numerous possible dual-range CVT variants, the ones with power split in HIGH represent the most favourable concepts with regard to variator load and fuel consumption.

Optimum variants for the summation gear set

The following section deals with the question of what are the other options for the summation gear set with power split in HIGH and which is the most favourable. Previous considerations assumed a summation gear set arranged in the form of a simple negative gear set.

With a coupled planetary gear set (consisting of two planetary gear sets), it is possible to reduce the variator spread even more while simultaneously limiting the variator torque to levels comparable with those in figure 6a with power split in HIGH and a simple negative gear set as a summation gear. Consequently, it is possible to halve the variator wear compared to the case of a simple negative gear set. Additionally, this arrangement allows for higher efficiency levels.

Therefore, considering both fuel consumption and variator load, a coupled planetary gear set is the most favourable variant.

Selection of the planetary gear ratios

The following illustrates how to determine the gear ratios i_1 and i_2 of the two coupled planetary gear sets. These ratios have a direct effect on the variator load and the fuel consumption. Through appropriate selection of both ratios, the jump in variator torque at the mode change point (figure 3) and the variator torque in total transmission OD are minimised. The specification of total transmission UD and OD along with the permissible range for the reverse gear ratio defines the main parameters for the arrangement. The relationship between total ratio and variator ratio is then fixed, if the ratio at the mode change point is also defined.

Figure 7 shows the ratio of variator torque and engine torque, $T_{Variator}/T_{Engine}$, at the mode change point in power split depending on set ratios i_1 and i_2. Along the continuous black line the ratio of $T_{Variator}/T_{Engine}$ is constant.

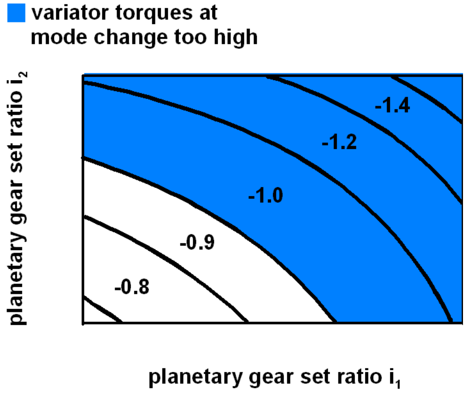

Fig. 7: *Ratio $T_{Variator}/T_{Engine}$ at the Mode Change Point in Power Split depending on Set Ratios i_1 and i_2*

Figure 8 shows progressively the variator ratio, the variator torque and the calculated chain damage for a time portion taken from a WOT acceleration. In the illustrated time portion, the vehicle speed increases from around 160 km/h to 220 km/h. The maximum chain damage occurs directly at the mode change point. One cause for this is the small running radius of the chain on pulley set 2 at the mode change point as shown in figure 6. On the other hand, the chain speed is at its maximum at the mode change point due to a constantly high engine speed. The large chain centrifugal force is therefore a further reason for the high chain damage in the vicinity of the mode change point. However, the amount of damage in power split is already minimised by selecting the set ratios so that the absolute values of the variator torque, $|T_{Variator}/T_{Engine}|$, is as small as possible. Subsequently, it will be shown that this ratio cannot be further reduced, because other criteria (overall size, reverse gear ratio) must also be fulfilled. Consequently, there is a minimum $|T_{Variator}/T_{Engine}|$. The range with the undesirably high $|T_{Variator}/T_{Engine}|$ values is shown in blue in figure 7. To calculate the time period shown in figure 8, optimal gear set ratios i_1 and i_2 were chosen. Other set ratios with higher values of $|T_{Variator}/T_{Engine}|$ are shown in figure 8 to run into the blue zone and would result in higher chain damage.

As already mentioned above, there are other criteria which limit the value ranges for i_1 and i_2 besides the total ratio spread and jump size in the variator torque. The desired total ratio spread, the target value for the reverse gear ratio, packaging and the resulting limitations for the overall size of the planetary gear set further limit the selection of the set ratios.

The shaded area in figure 9 shows the criteria and the associated non-permissible ranges in the i_1-i_2 level. The permissible i_1-i_2 values in this application are taken from the overlap of all non-shaded areas. This value range for i_1 and i_2 is marked in green.

The maximum permissible engine torque for a dual-range CVT, depending on total ratio spread and axis distances, was calculated using the optimising measures described in the previous sections. The calculations are based on the further-developed LuK chain [3]. Figure 10 shows the results. Additionally, the values for a single-range CVT with a total ratio spread of 6 are shown. The values for both the current standard chain and the further-developed LuK chain are plotted. According to figure 10, a CVT with a total ratio spread of 6 for 500 Nm is possible, with an axis distance between 170 mm and 190 mm. To achieve a total ratio spread of 7, the maximum permissible engine torque has to be reduced by roughly 5%.

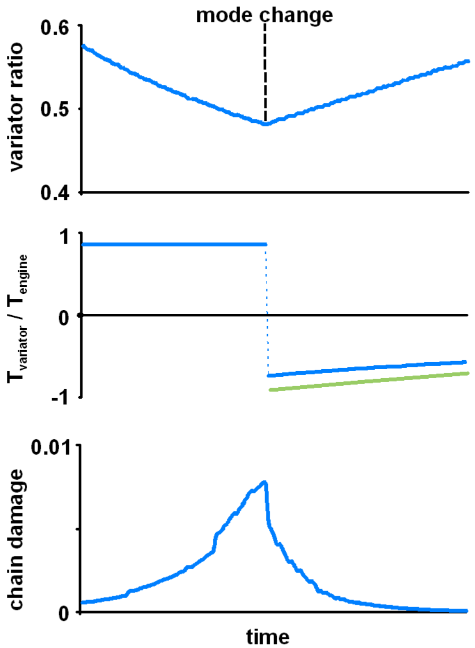

Fig. 8: Variator Ratio, Variator Torque and Chain Damage versus Time during a WOT Acceleration in the Vicinity of the Mode Change Point

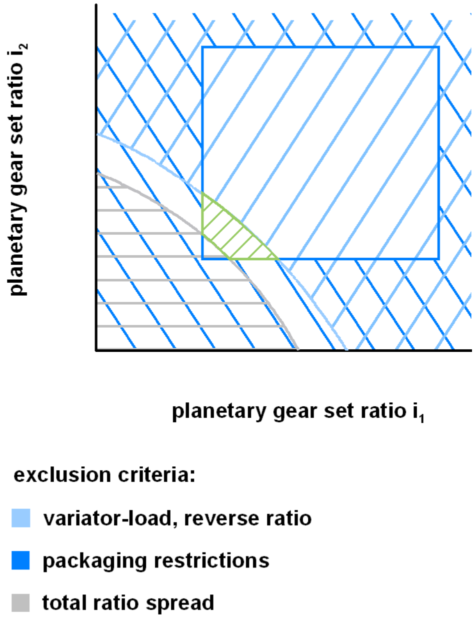

Fig. 9: Exclusion Criteria for Determination of Set Ratios i_1 and i_2 of Coupled Planetary Gear Set

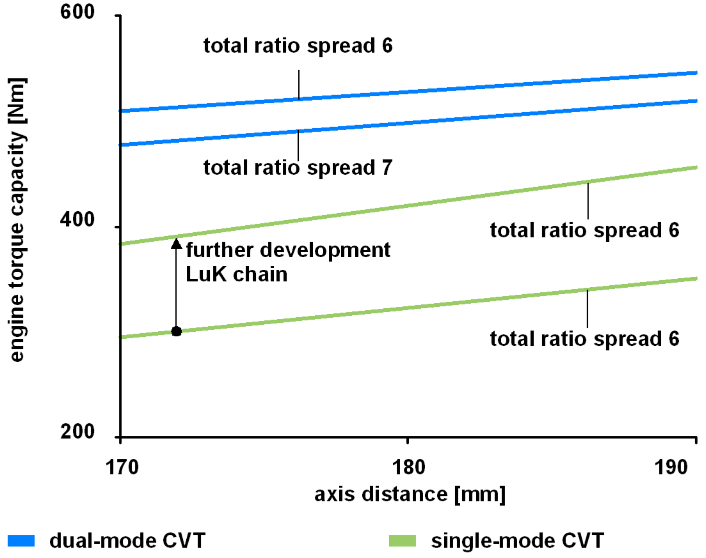

Fig. 10: Maximum Permissible Engine Torque of a Dual-Range CVT in Comparison with a Single-Range CVT for Various Axis Distances and Spreads

Hydraulics

It was shown in figure 3 that the variator of the dual range CVT favoured in this case is often in overrun condition. This is significantly different than a single-range CVT and has consequences for the clamping system. For example with a dual-stage hydro-mechanical torque sensor [6], [7], this would then require a layout compromise for both the overrun and drive conditions. Consequently, a dual-stage torque sensor would no longer provide fuel consumption benefits as on the multitronic®. For this reason, an electronically controlled clamping is envisioned for the power split CVT, which leaves all options for the different operating conditions open [10].

The following control functions must be provided on the dual-range CVT by the hydraulic system.

- Clamping
- Adjustment
- Forward and reverse clutches
- Mode change
- Cooling of clutches

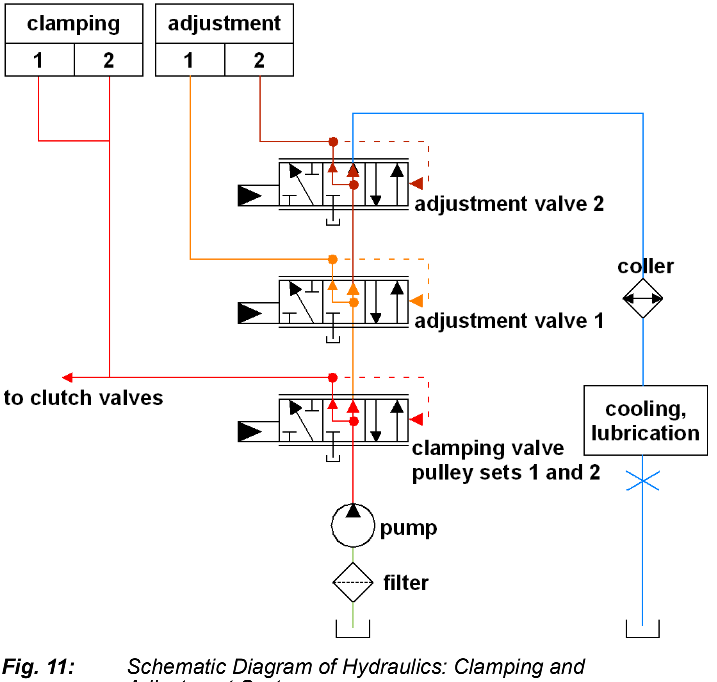

Fig. 11: Schematic Diagram of Hydraulics: Clamping and Adjustment System

Further optimisation is possible due to the fact that the hydraulic system is designed for particularly low back pressure. In power split mode, as mentioned in the previous chapter, the variator is driven with only a fraction of the engine torque. Therefore, the clamping force can also be reduced. If the hydraulic back pressure is low, the pump would also be subject to reduced pressure resulting in lower pump drive torque.

Cascade hydraulics

	cascade-hydraulics	multitronic®
electro-valves	6	3+ torque sensor
valves	11	9
pumps	vane size 122%	gear size 100%

Figure 11 shows the design of a cascade hydraulic system, which is arranged for these particular requirements.

The cascade operates a priority system: the clamping force, together with the clutches, has the highest priority; the adjustment elements are operated secondarily.

The number of functional elements has risen moderately in comparison to the multitronic®:

Loss reduction

In order to keep the flow demand during adjustments as low as possible, thereby minimising the pump drive torque for the transmission concept illustrated here, a double-piston system as on the multitronic® is required.

Because of the use of three proportional valves (clamping, adjustment 1 and adjustment 2) to control the clamping forces on two pulley sets, there is an additional degree of freedom. The control strategy can exploit this in order to minimise the system pressure (i.e. minimise the maximum value of the three required pressures). The advantage of the double-piston principle is thereby further increased with regard to loss minimisation.

Control strategy

The same demands with regard to driving comfort are placed on a dual-range CVT as on a single-range CVT. This does not only apply within a mode range, but also in particular during mode change. The control strategy is therefore an extremely important factor in achieving the comfort requirements.

Strategy aim

Because of the good fuel consumption, comfort and driving performance characteristics of the multitronic® [1], [2], a similar strategy is assumed here: The driver expresses the desired performance through the pedal position and the transmission ensures that this performance is delivered at an optimum consumption level of the engine (low engine speed). Acceleration takes place evenly and comfortably.

This comfort demand also exists during pedal changes or ratio changes: the output power must change comfortably. It must not be noticeable that there are several ranges, or that a change between the ranges is occurring. Simply ‘pausing briefly at the mode change point, shifting, and then continuing to adjust’ is not sufficient.

The behaviour during ratio changes in this case has proven to be particularly relevant to comfort, as internal rotational masses in the transmission act as flywheel accumulators, which absorb and release power.

Figure 12 shows an example of a fast UD adjustment, which means: the transmission should rapidly raise the engine speed (at constant output speed). The ideal speed progression of the engine then shows no comfort-reducing characteristics. The speed progression of the internal pulley set 2, however, shows curve sections of differing steepness and even a kink at the mode change point. It is shown in figure 13 that this results in a loss of comfort. Immediately before the mode change in power split mode, the pulley set 2 has to accelerate even more than the engine due to the variator adjustment. After the mode change, in the unsplit range, the speed of pulley set 2 is nearly constant.

Because an increase in speed always requires torque and consequently power, the tractive force changes in the shown speed progression of pulley set 2. This change in the tractive force is not included in figure 12, but is assumed in the simulation in figure 13.

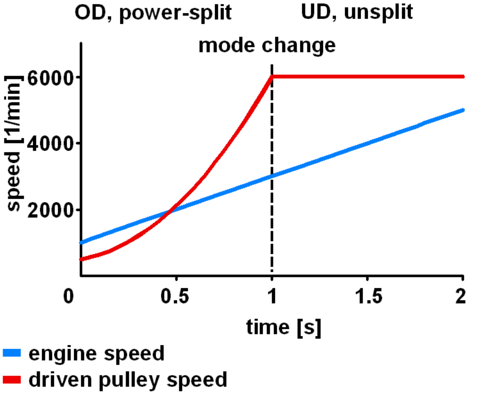

Fig. 12: Schematic Drawing of Engine Speed and Pulley Set 2 Speed over Time during an Adjustment at Constant Vehicle Velocity

In view of this, the strategy aim is not only to comfortably change modes at the right time, but an overall more comfortable progression of the output performance. This is achieved through engine intervention and a dynamic gradient control for the engine speed.

Simulation results

Figure 13 shows the simulation results for a demanding situation: a UD adjustment after kickdown with a mode change during a speed gradient. Illustrations are of the time-dependencies of variator ratio, engine speed and vehicle acceleration.

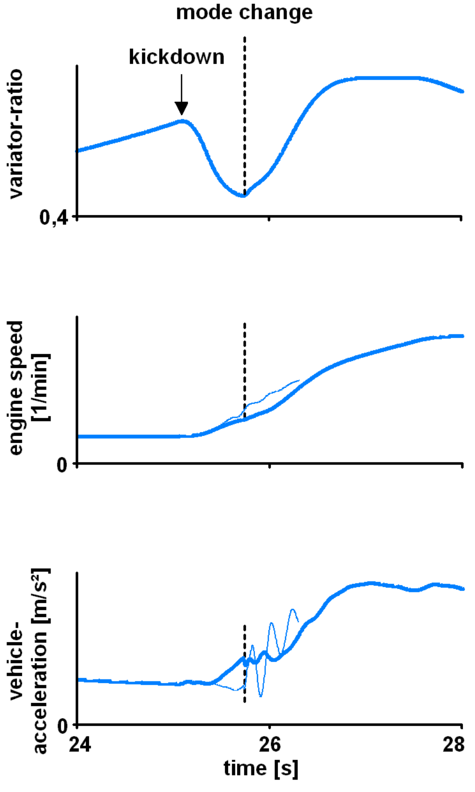

Fig. 13: *Simulation of Mode Change with an Optimised Strategy (Bold Lines)*

For Comparison, the Curves for a Strategy without Engine Intervention and without Gradient Adjustment are Shown (Thin Lines).

In the left part of the time record, the accelerator pedal is actuated by 30% and the engine is held by the transmission at an optimum consumption level of 1100 min^{-1}. The vehicle accelerates slightly and the variator is adjusted slowly in HIGH. At the time $t \approx 25$ s, a kickdown actuation is carried out, followed by an adjustment from HIGH to LOW. The center part of the illustration shows how the variator adjusts towards mode change (here at variator ratio 0.5), where the mode change to LOW takes place with simultaneous acceleration of vehicle and engine. A further acceleration of the vehicle can be seen in the right part of the illustration. The engine is running with consideration to speed tracking – similar to that in multitronic® [1], [2] – at near optimum performance.

A comfort evaluation according to the vibration amplitudes in the vehicle acceleration was performed. With an optimised strategy, the peak to peak distance in the area of the mode change is approx. 0.4 m/s^2. In comparison, a gear change of 1.25 in an automatic transmission equates to approx. 0.5 m/s^2. However, it has to be emphasised that the specified accelerations differ in slope. Another point is that the comfort demand varies according to the driving situation.

The thinly drawn curves in the figure are a counter-example of the time dependence with a control strategy, which includes neither engine intervention nor any adjustment of speed gradients. Here the largest distance from peak to peak in the vibration amplitudes of the vehicle acceleration is approx. 3 m/s^2. These vibrations decline after 4 seconds.

Two questions were theoretically answered through this strategy development:

1. Can a power split CVT be controlled with a reasonable effort?

Answer: Yes, the above strategy is suitable for embedded control.

2. Can the mode change be realised comfortably?

Answer: It is believed that the mode change can be realised without any impact on comfort and vehicle performance.

The basis of the work is the knowledge of the operational characteristic of the variator (multitronic®), the know-how regarding clutch actuation and shifting (Easytronic® [11]) as well as the development and simulation tools available within LuK.

Summary

Power split transmission structures present a promising concept for future applications of CVT's with torques up to 500 Nm and total ratio spreads between 6 and 7. With a dual-range CVT, the limitation of torque transfer capability of the chain can be raised, as the power flow through the variator at many load points is decreased. Within the scope of the present theoretical investigation, it is shown that with LuK components, a 500 Nm CVT with a power split can be realised. The available potential of a dual-range CVT can only be fully exploited if the concept is optimised with regard to efficiency along with variator and chain load.

On the hydraulic concept, the LuK double-piston principle was combined with specially adapted cascade hydraulics. A further development of the clamping system is integrated, which has advantages regarding fuel consumption and component load.

Computer simulations with a control strategy specially developed for this transmission structure show that high driving comfort is achievable even across the mode change. This is possible without any significant limitation in driving dynamics.

References

[1] Nowatschin, K.; Fleischmann, H.-P.; Gleich, T.; Franzen, P.; Hommes, G.; Faust, H.; Friedmann, O.; Wild, H.: multitronic® – Das neue Automatikgetriebe von Audi, ATZ 102 (2000) 7/8 and ATZ 102 (2000) 9.

[2] Gesenhaus, R.; Nowatschin, K.; Hommes, G.; Deimel, A.: Wie erlebt der Fahrer die neue Getriebegeneration multitronic® von Audi? VDI reports no. 1610, Getriebe in Fahrzeugen 2001.

[3] Indlekofer, N.; Wagner, U.; Teubert, A.; Fidlin, A.: Latest Results from the CVT Development, 7th LuK Symposium 2002.

[4] Förster, H. J.: Stufenlose Fahrzeuggetriebe, publisher TÜV Rheinland, 1996, and included references.

[5] Looman, J.: Zahnradgetriebe Springer-Verlag, 1996.

[6] Faust, H.; Linnenbrügger, A.: CVT Development at LuK, 6th LuK Symposium 1998.

[7] Englisch, A.; Faust, H.; Friedmann, O.: Innovative System for Clamping and Adjusting of a Chain Variator, Proceedings of the Global Powertrain Congress, Detroit (U.S.A.) 2001.

[8] Wagner, U.; Teubert, A.; Endler, T.: Development of CVT's for Passenger Car Applications up to 400 Nm, VDI reports no. 1610, Getriebe in Fahrzeugen 2001.

[9] Wagner, U.; Teubert, A.; Endler, T.: Development of CVT Chains for Passenger Car Applications up to 400 Nm, Proceedings of the Global Powertrain Congress, Detroit (U.S.A.) 2001.

[10] Faust, H.; Homm, M; Bitzer, F.: Optimisung the Efficiency of a CVT Clamping System – Reducing Consumption through Increased Slip? 7th LuK Symposium 2002.

[11] Fischer, R.; Berger, R.; Bührle, P.; Ehrlich, M.: Advantages of the electromotive LuK ASG Easytronic® taking the Opel Corsa as an example, VDI reports no. 1610, Getriebe in Fahrzeugen 2001.