8

The Crank-CVT

More economical than a manual transmission and more comfortable than a conventional CVT?

Oswald Friedmann
Wolfgang Haas
Ulrich Mair

Introduction

Hardly conceivable that such a transmission could exist. But LuK sees a glimmer of hope of achieving such an ambitious aim. This has required an unconventional approach, which will result in a new transmission concept with new and most probably unusual characteristics.

Presently the only choice a car buyer has is between the comfortable but expensive automatic transmission or the less expensive manual shift versions.

A costing exercise showed that on all existing production transmission types, considerable expenditure is required to cope with driving conditions, which although occurring only rarely, are still important. Start-off elements and reverse gear account for a large proportion of transmission costs, particularly when the necessary associated hydraulic or electromotive actuation is included.

On geared-neutral transmissions, such as friction-wheel units, attempts have already been made to facilitate starting-off and reversing without the use of additional elements. For this, it must be able to slow the variator down to 0 1/min. Nevertheless, control problems seem to be preventing exclusion of the start-off element. LuK has therefore sought a completely different type of variator, which will allow replacement of the loss-incurring start-off element, the reverse gear and the fixed ratio gears.

This article presents a CVT with a variator that offers precisely these possibilities. Such a concept transmission has already been built by LuK and its basic functionality has been proven. Indeed, initial trials in a vehicle have already taken place.

The Basic Principle

The general principle is based on the known concept of transmitting the drive force through variable levers [1]. Figure 1 shows the schematic layout. The drive consists of a lever, which is linked via a conrod with a free wheel at the drive output. Because the crank radius at the drive end is considerably smaller than the distance from the pivot point at the output end or free wheel, the free wheel swivels back and forth when the crankshaft turns. The free wheel then drives the output end whenever its peripheral speed is greater than that of the drive output.

This however will not transmit a sufficiently uniform torque to the drive output. Therefore, several of these arrangements must be connected in parallel at timed crank angles. figure 2 shows how a crank with two conrods works on two free wheels, which are offset to one another by almost 180°. Additionally, for example, six of these crank units can be arranged axially in sequence so that the twelve free wheels engage one after the other.

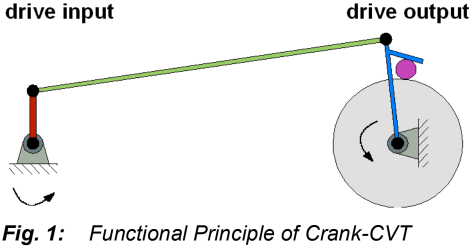

Fig. 1: *Functional Principle of Crank-CVT*

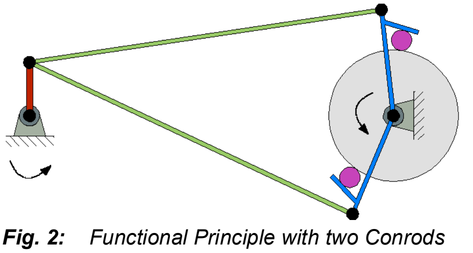

Fig. 2: *Functional Principle with two Conrods*

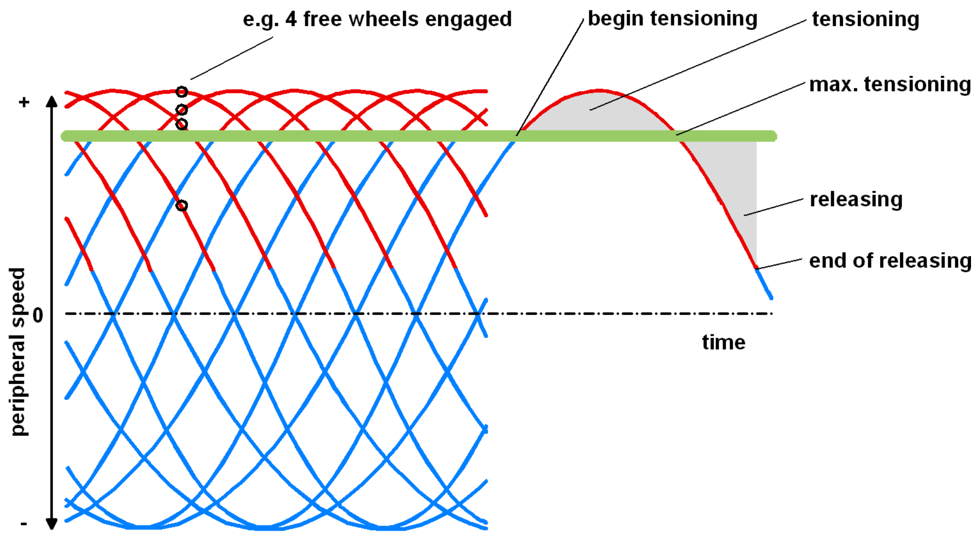

Fig. 3: *Process of Peripheral Speeds*

Figure 3 shows the progress of peripheral speeds over the period for such an arrangement. Below the zero line, the free wheels move backwards relative to the housing, forwards at the top. The green line equates to the drive speed. The actual engaged free wheels are shown red, the non-engaged free wheels are blue.

It would then be expected that the output speed will generally follow the relative maximum crank speeds. Because of the elasticity of the free wheels and conrods, which in practice allow a few degrees, the drive output speed will be below the maximum speeds of the free wheels.

At the point where the free wheel is released again without elasticity, the greatest torque is transmitted. Thereafter the torque falls away. However, the free wheel remains engaged for some time. The period during which a conrod can transmit torque is consequently significantly increased.

It only runs free again after complete release. The areas embraced during tensioning and releasing are then the same.

Because several free wheels engage simultaneously and further elasticities and masses smooth the torque fluctuations, an almost constant drive output speed results. The number of simultaneously engaging free wheels increases with the torque, which also simultaneously distributes the total torque over several free wheels.

Setting the Gear Ratio

If, according to the torque transmission principle presented, the total ratio is to be varied according to driving conditions from infinite (stationary) to around two (overdrive), then the drive radius of the crankshaft must be set to zero. Figure 4 shows how this is achieved.

The drive as on a crankshaft consists of a fixed crank throw and an eccentric that can revolve around it. Both pivot centre distances are the same. The rotating eccentrics have internal teeth which mesh with a central pinion shaft whose head diameter acts as a bearing. If the central pinion shaft is turned in relation to the crankshaft, all rotating eccentrics are simultaneously swivelled around the crank throw. The more the eccentric is swivelled inwards, the smaller the effective crank radius becomes until finally zero is reached.

This arrangement therefore facilitates adjustment of the eccentricity of the conrod drive from zero for a stationary vehicle to the maximum value for overdrive.

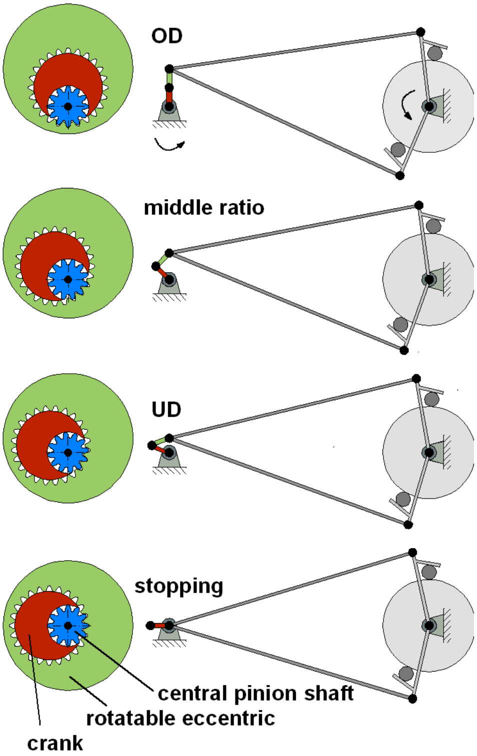

Fig. 4: Functional Principle of Adjustment

The Concept Transmission

Considering these basic principles of the Crank-CVT, the design of a concept transmission was started. The heart of this transmission, the variator, is shown in figure 5 in the complete view with adjusting mechanism. The layout is for a car with front-transverse arrangement up to around 75 kW. The principle is generally also conceivable for higher torques and other arrangements. The center distance between engine crankshaft and differential axle is fixed at 180 mm.

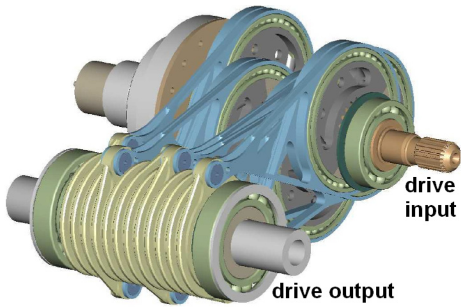

Fig. 5: Complete view of variator

All six eccentrics are each arranged in pairs with a total of twelve cranks. The total ratio lies between 2.1 and infinity thereby replacing both the start-off element and the fixed ratio gears.

The Drive Shaft

The drive shaft is of central importance in this design. The design with adjusting element for the eccentric is shown in figure 6 as a complete assembly and in exploded view. The normal carrier toothing of the drive shaft between damper and gearbox is located on the shaft end at the right. The drive/crankshaft with its throws is one-piece with a central drilling. This penetrates the individual crank webs to the inside. This drilling houses the central pinion shaft, whose teeth protrude through the penetrations. The rotatable eccentrics consist of two halves allowing them to be fitted over the crankshaft; they are held together by two enclosing roller bearings. The bearings in the housing are at both crank ends. The crank sequence is selected as on a combustion engine, with the aim of minimising any free mass forces and torque.

The adjusting mechanism, which can be seen at the left end of the shaft, has the task of turning the pinion shaft relative to the crankshaft thereby altering the transmission ratio.

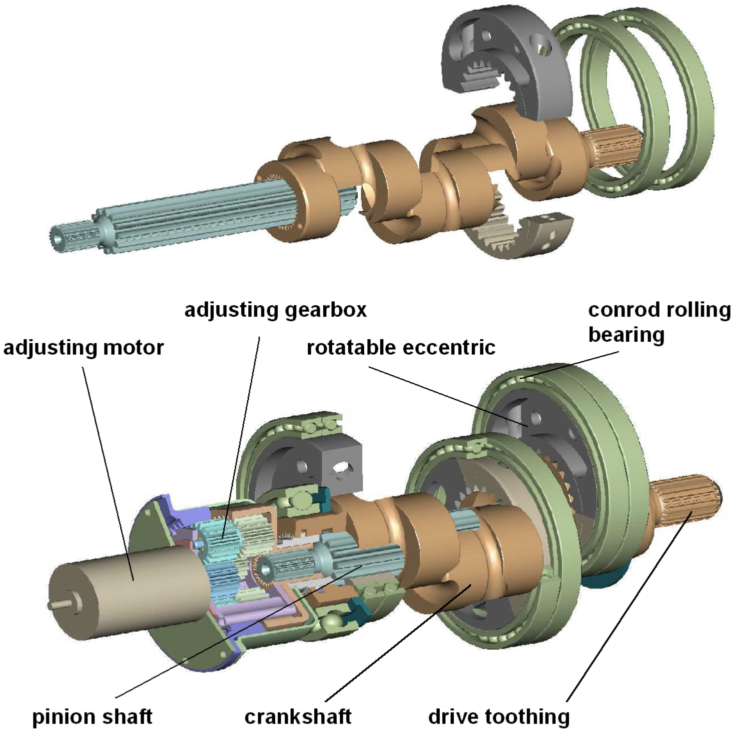

Fig. 6: *View of Drive Shaft*

Around one revolution of the pinion shaft is necessary for the entire ratio range. Between the electromotor for the adjustment (around 150 W) on the extreme left and the crankshaft end, there is a planetary gear set with a ratio of around 200, which is linked with the crankshaft and the pinion shaft. The housing of the electromotor turns with the crankshaft, the electric energy feed is via slip rings. Therefore, the electromotor and the adjustment gearbox only turns within itself when it is adjusted. If not adjusted, the crankshaft and pinion shaft turn with the adjustment gearbox as a block.

The Output Shaft

The output shaft contains the free wheels, which transmit the torque to the conrods. Figure 7 shows a view, in which a number of conrods with their free wheels are visible at the right end. The cut-out sections give a view of the rollers of the free wheels. They comprise of roller free wheels with an inner star. The small conrod eye engages directly in the outer ring via a plain bearing. The outer ring sits on the inner star over the cage-arranged plain bearing. The bearings in the housing are located at the two shaft ends, the torque is passed on to the differential cage directly from the inner star. Because of the distribution of the conrods at top and bottom, consequently pulling and pushing, the lateral forces are partially balanced due to the simultaneous engagement of several free wheels, which is beneficial to the bearing forces.

To reverse the vehicle, the working direction of the free wheels must be reversible. The side view in figure 8 shows how this works.

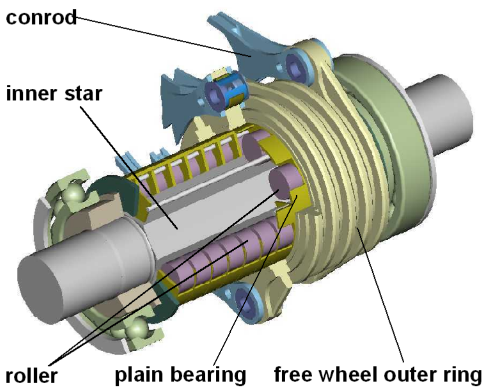

Fig. 7: *View of Output Shaft*

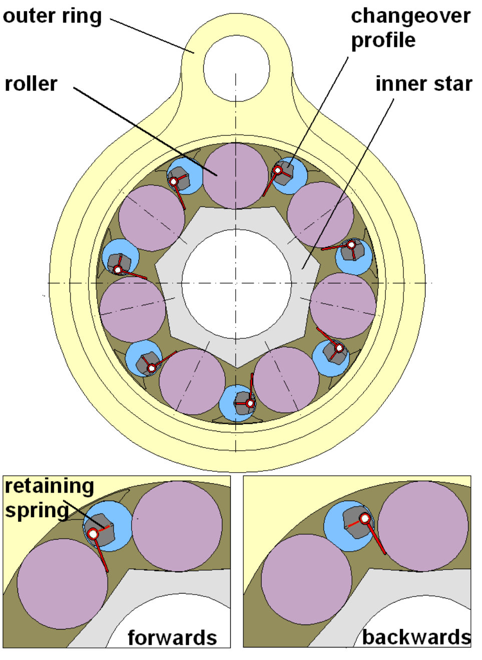

Fig. 8: *Forwards / Reverse Changeover*

The inner star is designed symmetrically for both rotational directions, the rotational direction changeover takes place by changing the force direction of the retaining springs for the individual rollers through approx. 180°. For this, the changeover profiles, which pass through all free wheels and carry the retaining springs, are simultaneously turned by a device not shown in the figure. All other functions for reverse travel are the same as for forwards travel, only the control range is limited.

For example, at a wheel torque of 1700 Nm, the individual free wheel has only to transmit max. 600 Nm because of the load distribution across several free wheels. Nevertheless, it is subjected to frequencies up to 100 Hz with strokes up to almost 40 mm.

To illustrate the dimensional proportions, the variator in figure 9 is schematic, but shown to scale with combustion engine and wheel. This is to clarify that the power flow is transmitted directly from engine to axle without further gearing and clutches.

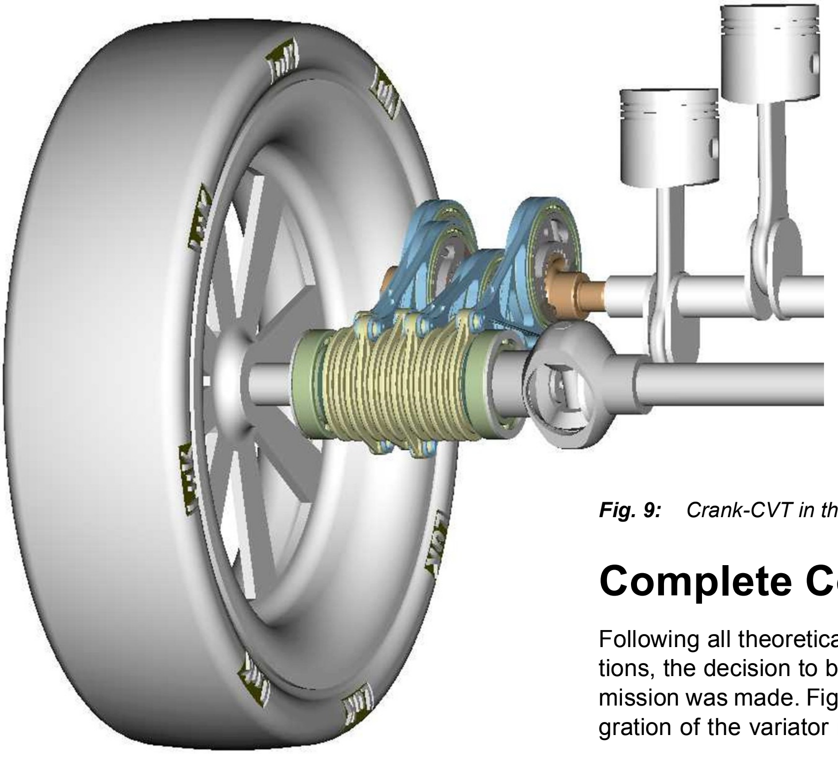

Fig. 9: *Crank-CVT in the Vehicle*

Complete Concept

Following all theoretical and design preparations, the decision to build a prototype transmission was made. Figure 10 shows the integration of the variator in a total transmission concept. To generate the thrust that the driver is accustomed to, a solution with an electric motor is shown here. This electric motor can be connected to the drive output via a gear set and/or directly connected to the combustion engine. This makes many combinations of starter, alternator, combustion engine with ad-

The Crank-CVT has a total weight of less then 40 kg without electric motor, which roughly equates to an equivalent manual transmission plus clutch. The number of different parts is less than on a manual transmission, but the number of rolling bearings is greater.

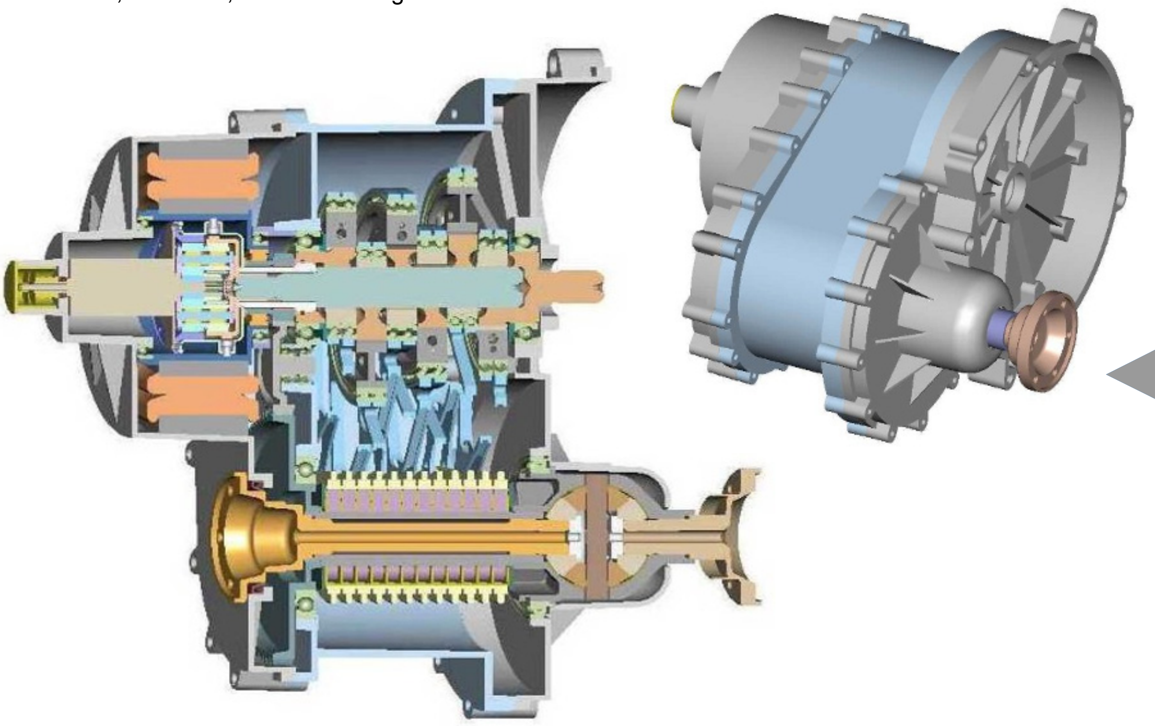

Fig. 10: *View of Complete Transmission*

ditional functions, such as recuperation and booster, possible. The normal axial space for a transverse transmission is generally sufficient for an 8 kW unit in axial position of combustion engine.

The critical design size of the Crank-CVT is the axial distance between the flange joint to combustion engine and the flange joint to short drive shaft. This is mainly determined by the length of the variator and presently is around 20 mm longer than on the corresponding manual shift transmission.

The Efficiency

Once such a transmission without E-motor and without a changeover facility for reverse travel has been built, the question of efficiency must be raised. The values measured to date are shown in figure 11 in comparison with a manual shift transmission and various CVT's. The upper range of the CVT efficiencies shows the theoretical possible efficiency of a CVT with similar torque capacity, using the efficiency-optimised LuK CVT technology [2], [3].

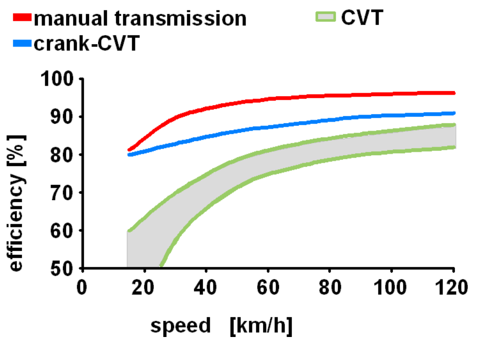

Fig. 11: Efficiency Comparison

Clearly the Crank-CVT shows its best potential primarily at lower speeds and loads, trailing only slightly behind the manual shift transmission. It can be assumed that a CVT of today can more than compensate for its poor efficiency in comparison with the manual shift transmission through the high ratio spread and the constant variability. A further point is that in comparison with the manual transmission or CVT there are no start-off and shift losses.

The greatest loss source in the Crank-CVT is theoretically the free wheels while they are being overhauled. Consequently, the pre-tension force and the running of the roller when not engaged are particularly important. Further development will certainly be focused on aspects concerning the high dynamic requirements of the free wheel.

Start-Off

When starting-off and during standstill, the elasticity of the conrods and free wheels described in section 2 is of benefit to the Crank-CVT. This can alleviate the creeping motion associated with the stationary vehicle and small eccentricities of the crank drive, and with extremely low losses, which is not the case with torque converters and clutches.

Figure 12 shows the maximum wheel torque and engine torque via the crank radius. Crank radius zero equates to the ratio infinite, the max. radius of the OD. The area (‘C’) is on the right, where the maximum wheel torque by the maximum engine torque times the effective ratio is given. In the middle area (‘B’), the slipping wheel itself determines the max. wheel torque. Extreme right is finally the area (‘A’) in which the wheel torque is determined by the crank position. In this area the ratio is infinite, as the slip limit is not reached and the vehicle is standing still. Disregarding losses the engine will not transmit any torque when the accelerator is pressed and the engine will freely accelerate. The speed process for this case is shown in figure 13 in conjunction with figure 3. It shows that the free wheel transmits torque practically the whole time, because the tension phase starts again directly after the release. Therefore, the wheel torque can be relatively accurately specified with the adjusted crank radius.

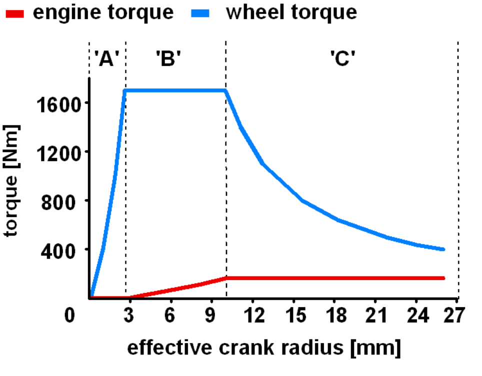

Fig. 12: Max. Wheel and Engine Torque

The infinite ratio along with the avoidance of clutch/converter losses ensure excellent start-off dynamics.

The effective crank radius of zero is determined with a mechanical stop, so that the infinite position can be simply reached without regulation. This is not possible on other geared-neutral concepts [4].

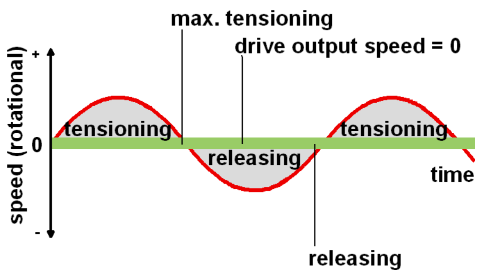

Fig. 13: Speed Loss of a Free Wheel on Stationary Vehicle

A creeping motion can however be set using this method if desired, as the high ratio for the adjustment motor allows a high dispersal of the adjusted eccentric. The concept transmission confirmed a well-balanced behaviour when starting-off. Figure 14 shows the measurement of a creep start-off with the engine at idling speed.

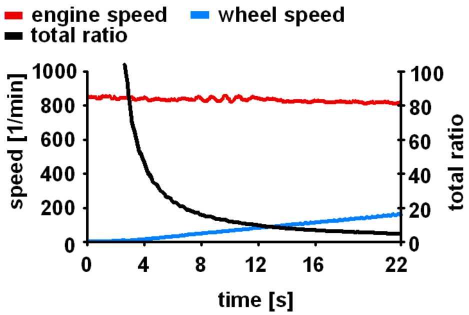

Fig. 14: Measurement of Creep Start-off

The Non-uniformity at Drive Output

Figure 15 shows a typical measurement of the speed fluctuation from engine and transmission output over a period. Because the test vehicle has a 4-cylinder petrol engine, two pulses in engine speed equate to one revolution at the drive output. This example shows the 12th order in the process of the drive output speed to be dominant, but other orders also overlap. This presently results in noise generation. Therefore, the further development of the NVH characteristics, along with the previously mentioned free wheel optimisation, will be a further focus of future work.

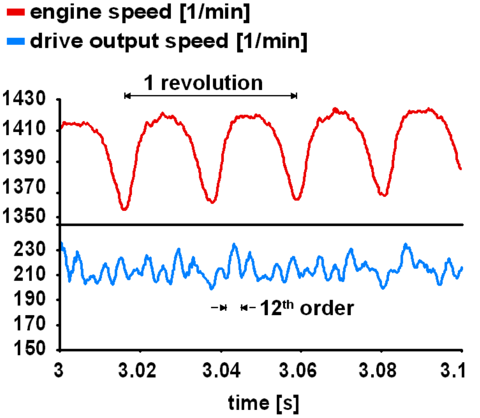

Fig. 15: Measurement of Non-uniformity

Overrun and Downhill Driving

Because the free wheels cannot transmit any overrun, solutions must be found for critical overrun situations such as downhill travel.

During normal driving conditions, the absence of overrun also has its advantages. The test vehicle so far appears uncritical with regard to longitudinal vibration, likewise the stopping whereby the total ratio is always adjusted somewhat shorter than the effective ratio. The problem of increasing engine braking effect as the ratio shortens is thereby resolved.

Figure 10 already showed how an alternator can be either linked with the drive output or the combustion engine. If both links are active, the combustion engine and the E-motor are attached to the drive output. The fixed ratio to the drive output is then selected so that at top speed the combustion engine and the E-motor rotate at their maximum speed.

Figure 16 shows the possible braking powers for overrun in relation to the wheel over the road speed. The braking power is made up of the air and rolling resistance (black line) and the braking torque of the engine. The total braking power in the different gears at maximum and low shift speed are shown by the red lines, which embrace the areas which can be influenced by the driver. The steps represent the individual gears.

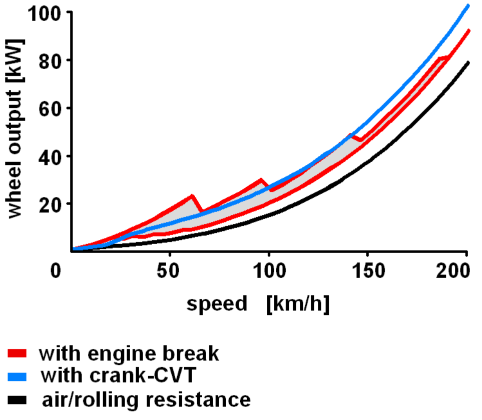

Fig. 16: Braking Power during Downhill Travel

On the Crank-CVT, it is not possible to use the different gears to influence the braking effect. If the alternator (nominal current 180 A) and the combustion engine are linked to the drive output via the fixed ratio, the resultant total braking power is shown by the blue line. As can be seen, the suggested solution is only exceeded at extreme combustion engine speeds and should therefore be sufficient for practical purposes. If the battery is already charged during prolonged downhill travel, the energy can be safely diverted to the cooling water using an electrical heater, as the combustion engine hardly produces any heat during this time. These are of course only initial suggestions. The overrun management must be developed together with the vehicle manufacturer.

Summary

The main outstanding feature of the Crank-CVT from LuK in the versions examined so far is the absence of assembly groups. This results in a light and cost effective CVT suitable for front transverse installation with the potential for economic consumption and a high level of comfort.

The development risk can be seen in the altered overrun characteristics, the achievement of durability and in the noise behaviour.

References

[1] Stölzle, K., Hart, S.: Freilaufkupplungen, Berechnung und Konstruktion 1961.

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

[3] Faust, H.; Homm, M.; Bitzer, F.: Optimising the Efficiency of a CVT Clamping System, 7th LuK Symposium 2002.

[4] Jürgens, G.: Dissertation: Potenziale und Risiken von Geared-Neutral-Strukturen stufenloser Getriebe 1995.