Toroidal CVT with compact roller suspension

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In this contribution a new, very compact roller suspension for a toroidal CVT is presented. The main topics in this case relate to a very simple and easy design, which allows a sufficiently quick and stable control. This new roller suspension needs essentially less components in the mechanical and in the hydraulic system than all other known suspensions. This leads to a more compact and more low-cost variator concept and with that it increases the market chances of such continuously variable transmissions.

1 Introduction

In Japan toroidal traction drives for vehicle applications are already in mass production. In Europe full- and halftoroidal transmissions have been intensively investigated in the last few years, but we are still waiting for the first mass production for an European vehicle. The main advantages of this kind of transmissions are the highest torque capacity for all CVTs and a very low operation noise.

Halftoroidal CVTs have a bit higher efficiency than fulltoroidal transmissions. But their production seems to be relative costly in comparison to automatic transmissions. One of the reasons therefore are their expensive roller suspensions with a lot of prismatic precise and grinded parts. On the other hand the design of fulltoroidal variators are much lighter and less complex. But up to now because of the higher thermal mechanical surface stress the discs and rollers of fulltoroidal variators do not achieve reliably comparable lifetimes as halftoroidal systems and as automatic transmissions.

In order to increase the chances for toroidal transmissions on the European market, it needs lighter gearbox designs. These designs should fit into the increasing slim driveline tunnels and offer a very stable operating behaviour as well as the highest possible torque capacities. Furthermore their production costs should be significant lower than today.

2 The structure of modern toroidal gears

As shown in fig. 1 modern toroidal transmissions consist of two half gears (cavities), which are braced axially against each other between the input discs in order to short-circuit the high preload force for the generation of the contact normal forces.

Each half gear has a toroidal input disc and a toroidal output disc. In a roller suspension, which is arranged between the toroidal discs, two or three rollers are mounted as it is shown in fig. 2. All the rollers have traction contacts to both discs. To change the ratio the rollers can be tilted within the torus space.

For a synchronous ratio change at all rollers in both cavities the toroidal variator is force controlled. By means of hydraulic pressure the traction force, which has to be transmitted, is adjusted with the same value to each roller. If such a pressure force does not keep the balance with the in fact operating traction force, the roller suspension allows a deflecting motion of a roller, which causes a tilting motion within the torus.

Basically by the deflecting movement a side slip arises and generates a tilting torque that adjusts the roller onto an other race radius (tilting). The ratio changes and with that the forces at the roller and the roller suspension change as well until the force balance is found again.

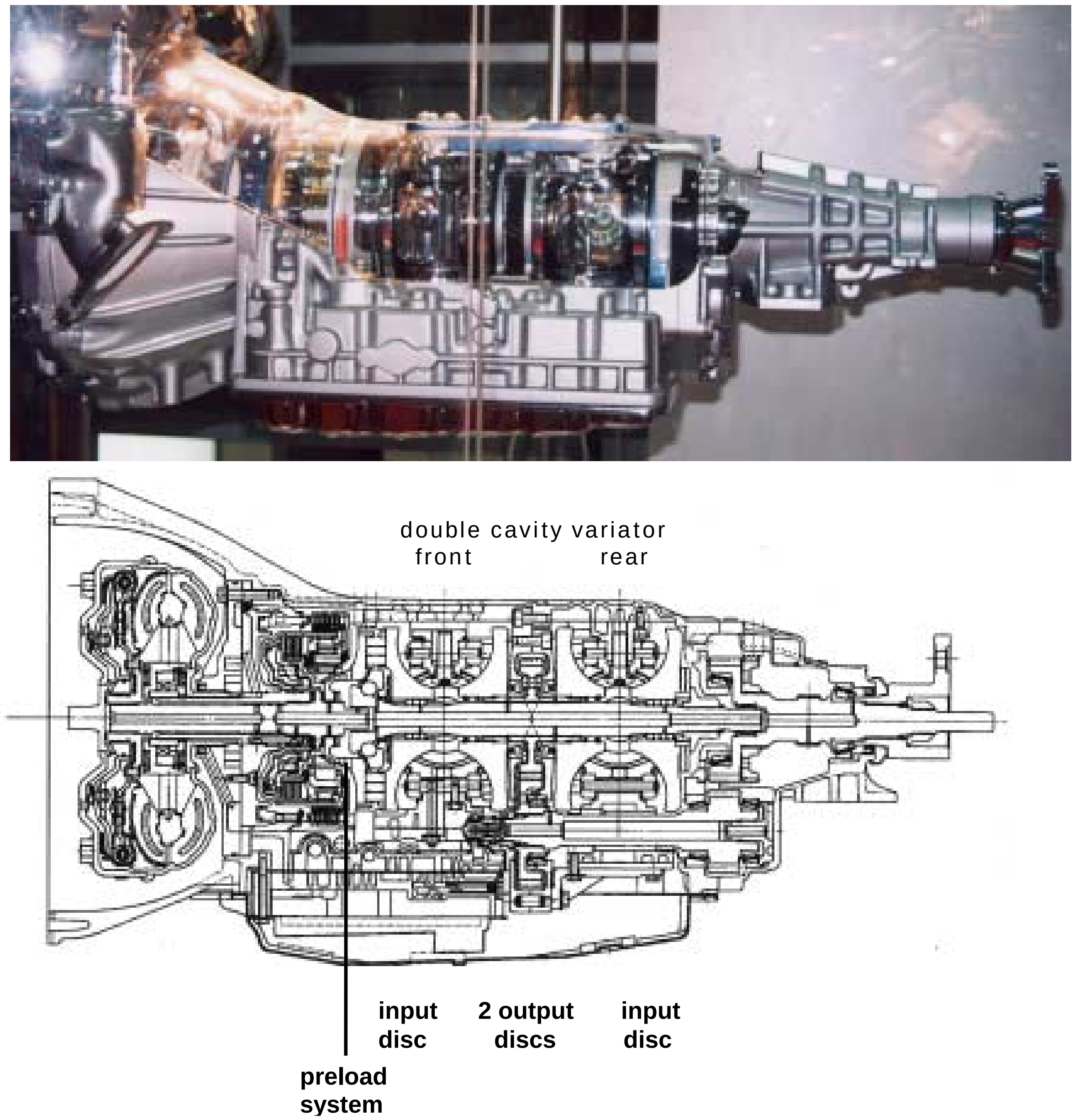

Fig. 1: Design of Nissans traction CVT for the Gloria and Cedric cars with a halftoroidal double cavity variator from NSK

With a feedback of both roller movements (deflecting and tilting) of at least one roller onto a control valve a ratio control can be built up.

Due to elastic deformations, production tolerances, unequal lengths of the hydraulic pipes to the individual rollers, etc. and due to temporal variations of for example some damping parameters some oscillations can arise if the variator is driven dynamically. These oscillations have to be stabilized by mechanical-hydraulic or electrical-hydraulic controllers, otherwise unstable operation could lead to the so called "hunting", which means to a permanent or at least long lasting oscillation with a large amplitude.

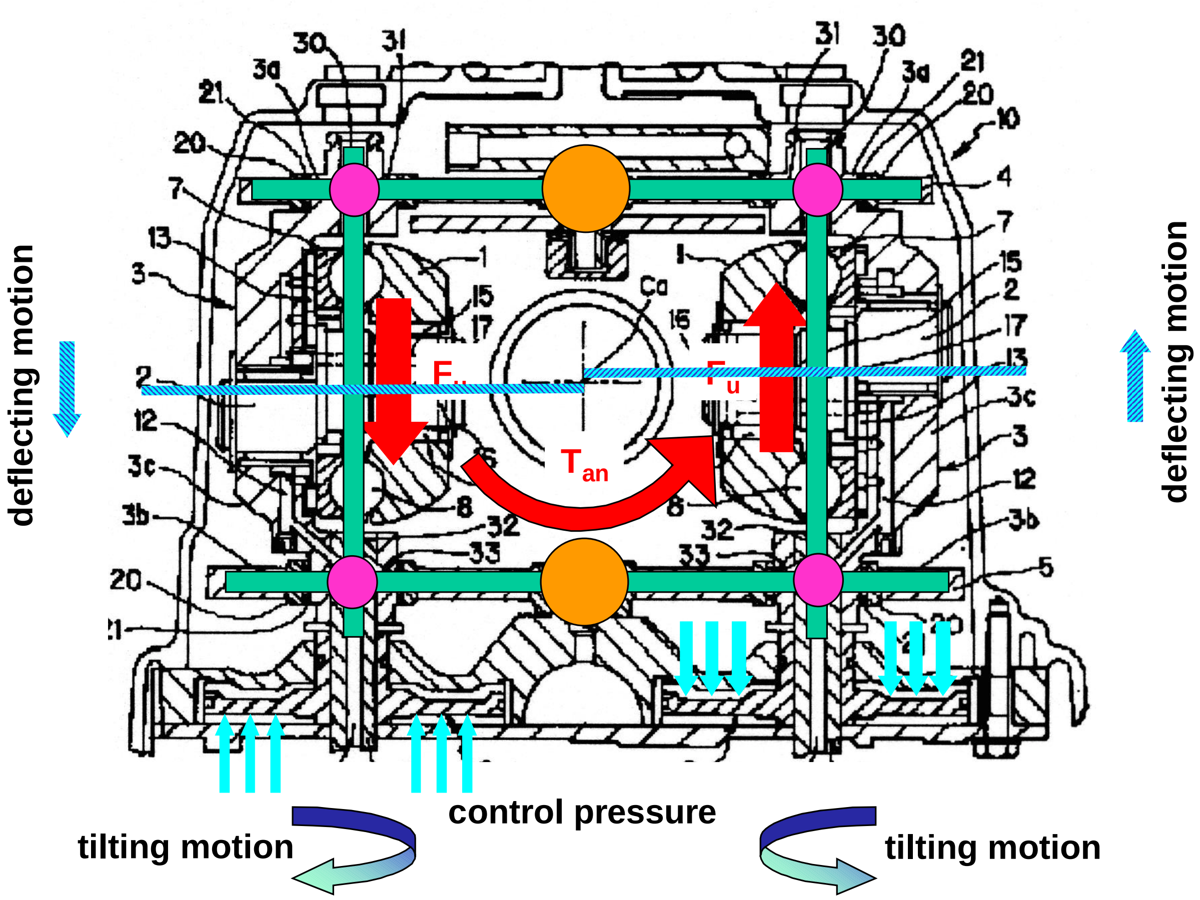

Fig. 2: Roller suspension of the NSK-variator when changing ratio to underdrive

To calm down the dynamic behaviour in some variator designs the deflecting motion happens in a plane which is inclined (with the so called castor angle) to the torus middle plane. Such an inclined deflecting motion requires an according tilting motion to keep the roller stable in the torus. In this case even a small change of a tilting position immediately generates side slip and a reacting tilting torque to stabilize the ratio.

But only with constructive precautions at the roller suspension and sufficient dampers unfortunately it is not possible to achieve stable performance in all driving situations. A lot of simulations to the dynamic behaviour and measurements at some test variators show that the control system reacts quite sensitive on parameter variations. Even small side slip angles suffice to generate large tilting torques. Because of the small inertias of the rollers and their trunnions this leads to large accelerations in tilting direction and by that to very quick ratio changes.

From outside a lot of impacts effect onto the gearbox, for example if the car is driven through a pot-hole. If such impacts affect only the position of a few rollers or different rollers in different direction this can lead fast to considerable malpositions between certain rollers. The result of such malpositions is an uneven power splitting between all the rollers. In an extreme case it is even possible to get some bracing torques within a variator or only within one cavity and much higher losses by that.

In order to avoid such operating states surely, the known halftoroidal variators have a mechanical system, which couples all deflecting movements by means of some yokes, and they have a safety system, which permits all the roller movements in the torus and by that the hydraulic force control only in a small control range. Larger deviations of the roller positions, however, are blocked mechanically by ropes and limiting stops.

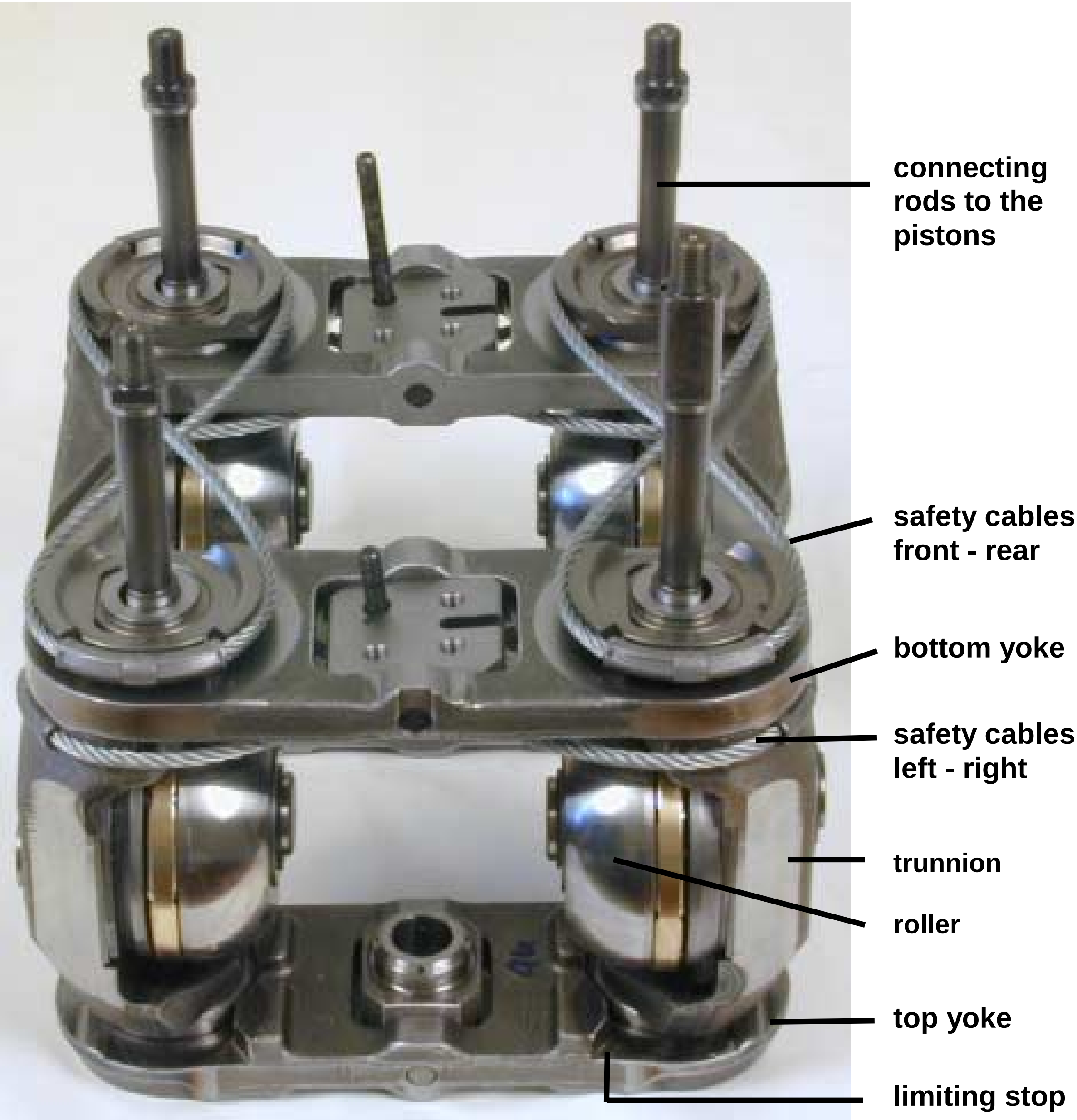

Fig. 3: Roller suspension of the NSK-variator with mechanical coupling of the deflection movements and with a mechanical safety system

But just such a mechanical safety system makes the variator design expensive, large and heavy. Therefore many engineers are looking for lighter roller suspensions and more simple and more stable control devices for such continuously variable transmissions.

3 Power splitting, efficiency, operating behaviour with roller malpositions

Table 1: Datas of an exemplary operating point

$T1 = 300$ Nm	$n1 = 4000\ \mathrm{min}^{-1}$	$A_{K1} = 9.2\ \mathrm{mm}^2$	$A_{K2} = 10.5\ \mathrm{mm}^2$
$u = 1.414$	$v_u = 19.6\ \mathrm{m\,s}^{-1}$	$a_{K1} = 2.39$ mm	$a_{K2} = 2.28$ mm
$F_n = 17872$ N	$VF = 0.05971$	$b_{K1} = 1.22$ mm	$b_{K2} = 1.46$ mm
$Fax_{an} = 43724$ N	$BWV = 0.218$	$pHmax_{K1} = 2915$ MPa	$pHmax_{K2} = 2556$ MPa
$s1 = 1.346$ %	$s2 = 0.878$ %	$\theta max_{K1} = 146$ °C	$\theta max_{K2} = 127$ °C
$\zeta_{B1} = 1.839$ %	$\zeta_{B2} = 1.263$ %		

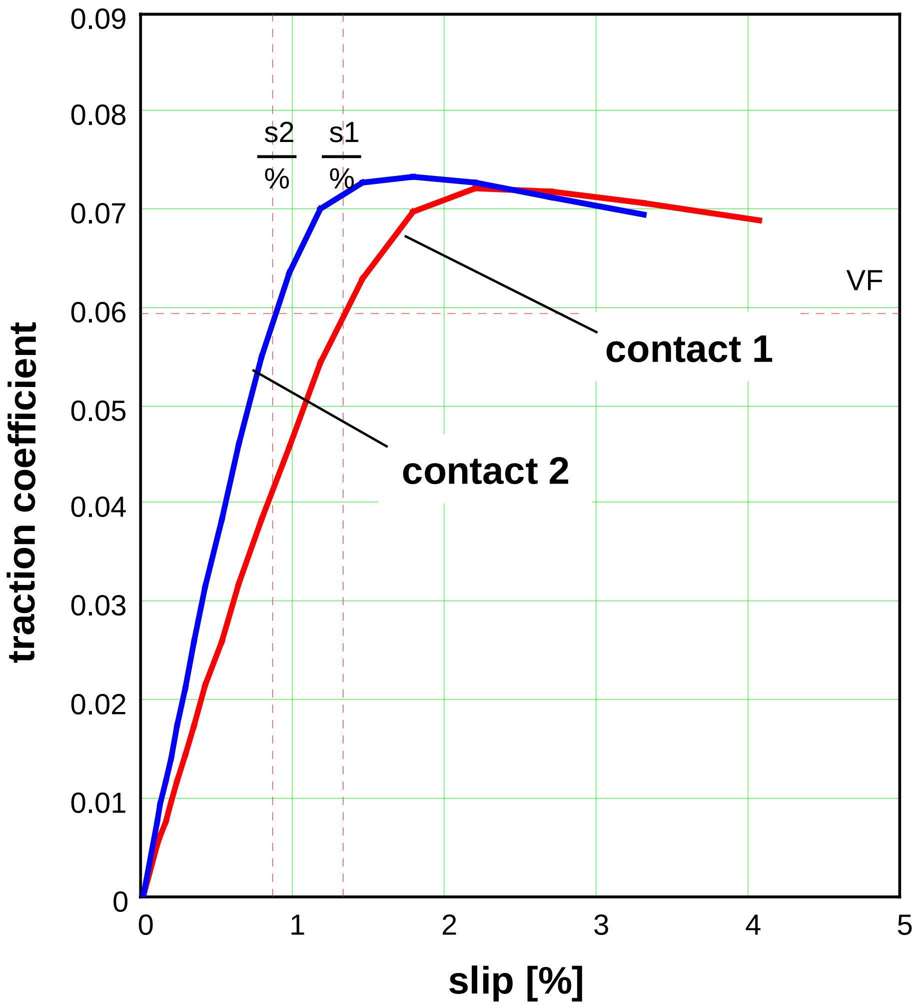

Fig. 4: Traction coefficient via slip of both contacts for the exemplary operating point

For each roller in each position in the torus all geometrical datas of the contacts to both discs are well known. With the radii of curvature and the contact normal forces all the contact areas can be determined. With the solution of the shear stress equations from Johnson and Tevaarwerk and depending on side slip and on tangential slip we can calculate the local shear stresses. With the local sliding movements depending on slip and spin we can determine the local power losses. By means of the equations describing the heat conduction we finally get the surface temperature in time. The used traction coefficient results from the integration of the local shear stresses in relation to the contact normal force. The total traction loss in a contact results from the integration of the local power losses.

In fig. 4 the traction coefficient via the slip of both contacts is shown exemplary for an operating point of a halftoroidal variator without side slip. Due to the different contact areas and different spin values these graphs are slightly different in both contacts. In fig. 5 it is shown how the average contact temperature and the maximum contact temperature increase with the slip. In this case the mass temperature is kept constantly to 95°C. That means that the cooling increases with the losses. With additional side slip the traction coefficients would decrease, the total contact losses and the contact temperatures would increase.

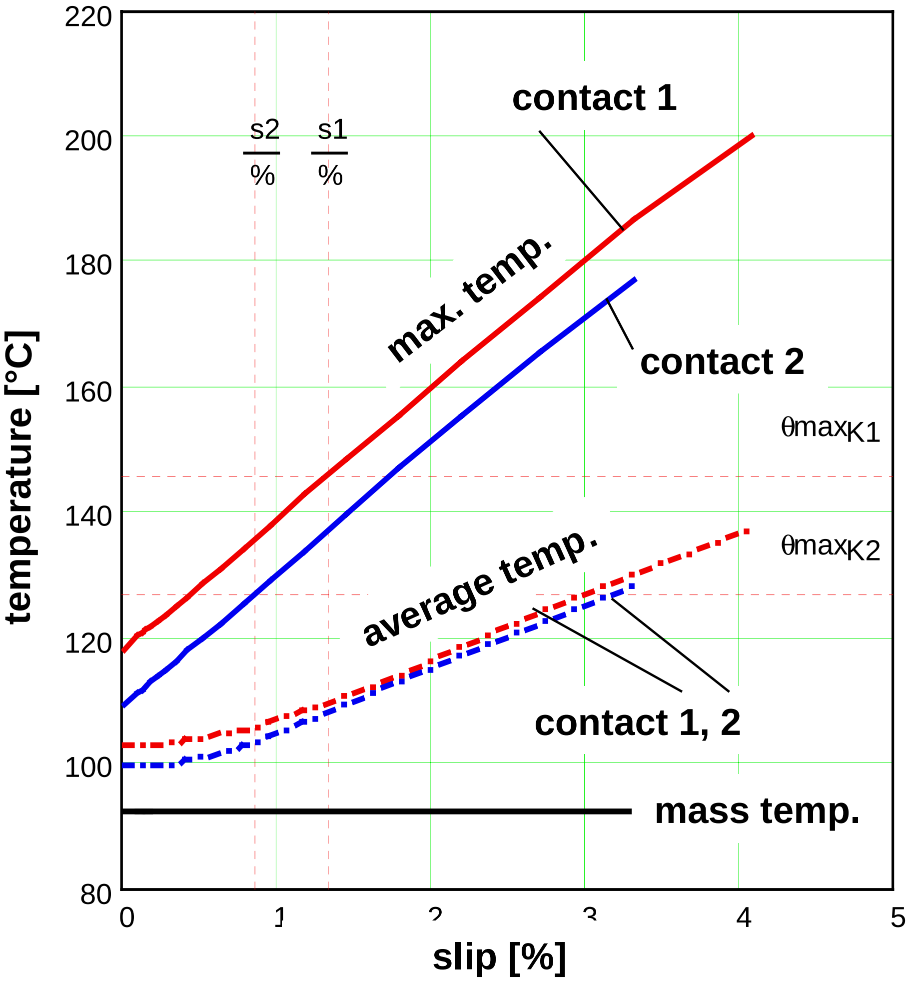

Fig. 5: Contact temperatures via slip of both contacts for the exemplary operating point

For an efficiency-optimal operation the axial preload of the variator is chosen so that the entire loss degree becomes minimal. Our calculations have shown that then also the contact loss due to slip and spin becomes minimal because the small slip changes basically affects only this loss degree.

With the variator preload, corresponding to an input torque at a certain ratio, an used traction coefficient is chosen. The slips and losses in both contacts result from that. Fig. 6 shows that with the here applied efficiency-optimal preload there remain some traction reserves in the contacts. In addition the fig. 4 to 6 show that small variations of the slip from this optimal value have considerable effects on the traction coefficient and contact temperature, but only little effects on the loss degree.

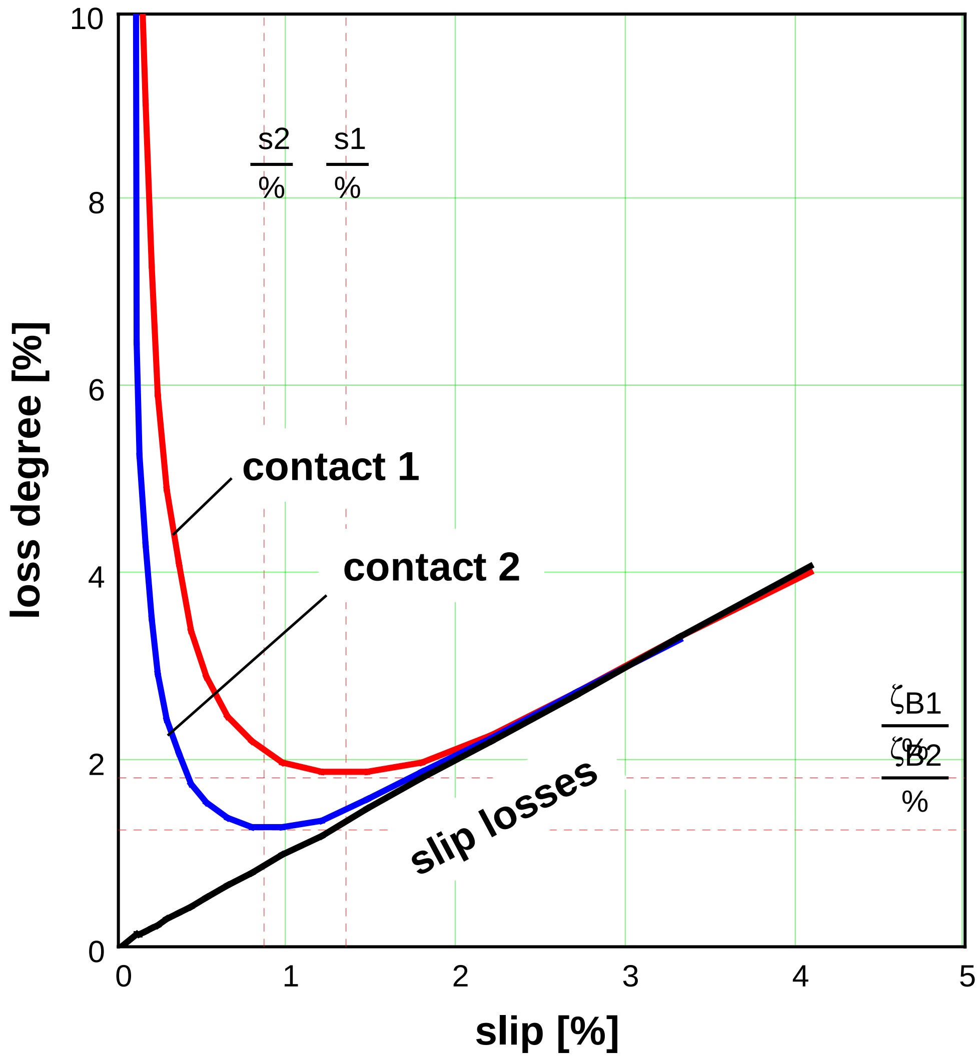

Fig. 6: Contact losses due to slip and spin of both contacts for the exemplary operating point

The next question is what slip values arise at the individual rollers of a variator, if these rollers do not run at the same ratio positions. This certainly happens in all variators which oscillate within the limits of the mechanical safety system. To explain this effect we extended our simulation programs so that we can input the tilting positions of each roller.

The average geometrical ratio of all rollers results from the average roller position. Furthermore the input speed and the input torque should be known. With that the controller can generate the axial preload for an optimal traction coefficient. The axial preload splits onto the different rollers according to the elasticities of all contacts and force ways and generates the contact normal forces there.

Table 2: Power splitting in a halftoroidal variator with six rollers with either equal or slight unequal positions

Operating point with equal roller positions						**Operating point with little mispositions at some rollers**					
tilting angles [°]		contact radii at input disc [mm]		contact radii at output disc [mm]		tilting angles [°]		contact radii at input disc [mm]		contact radii at output disc [mm]	
-31	-31	77,38	77,38	38,75	38,75	-31	-31,5	77,38	77,73	38,75	38,54
-31	-31	77,38	77,38	38,75	38,75	-31	-31	77,38	77,38	38,75	38,75
-31	-31	77,38	77,38	38,75	38,75	-30,5	-31	77,04	77,38	38,97	38,75
		traction coefficient in contact 1		traction coefficient in contact 2				traction coefficient in contact 1		traction coefficient in contact 2	
input /output speed [rpm]		0,0551	0,0551	0,0551	0,0551	input /output speed [rpm]		0,0607	0,0734	0,0607	0,0734
		0,0551	0,0551	0,0551	0,0551			0,0607	0,0605	0,0607	0,0605
2500	-4935	0,0551	0,0551	0,0551	0,0551	2500	-4927	0,0155	0,0605	0,0155	0,0605
roller speeds [rpm]		slip in contact 1 [%]		slip in contact 2 [%]		roller speeds [rpm]		slip in contact 1 [%]		slip in contact 2 [%]	
5139	5139	0,389	0,389	0,764	0,764	5136	5127	0,443	1,054	0,864	1,256
5139	5139	0,389	0,389	0,764	0,764	5136	5136	0,443	0,442	0,864	0,865
5139	5139	0,389	0,389	0,764	0,764	5131	5136	0,098	0,442	0,201	0,865
		total slip loss		1,152%				total slip loss		1,494%	
		loss degree due to slip and spin [%]		loss degree due to slip and spin [%]				loss degree due to slip and spin [%]		loss degree due to slip and spin [%]	
		0,618	0,618	1,249	1,249			0,611	1,083	1,223	1,362
		0,618	0,618	1,249	1,249			0,611	0,612	1,223	1,226
		0,618	0,618	1,249	1,249			3,369	0,612	3,218	1,226
		total slip + spin loss		1,868%				total slip + spin loss		2,193%	
		input torques [Nm]		output torques [Nm]				input torques [Nm]		output torques [Nm]	
in total		250		125,206		in total		250		124,972	
		41,7	41,7	20,9	20,9			45,5	55,8	22,8	27,7
		41,7	41,7	20,9	20,9			45,5	45,8	22,8	22,9
		41,7	41,7	20,9	20,9			11,6	45,8	5,9	22,9
		power splitting [%]		power splitting [%]				power splitting [%]		power splitting [%]	
in total		100		100		in total		100		100	
		16,7	16,7	16,7	16,7			18,2	22,3	18,2	22,1
		16,7	16,7	16,7	16,7			18,2	18,3	18,2	18,3
		16,7	16,7	16,7	16,7			4,6	18,3	4,7	18,3
mass temperature		maximum contact temperature [°C]		maximum contact temperature [°C]		mass temperature		maximum contact temperature [°C]		maximum contact temperature [°C]	
		103	103	118	118			104	115	120	127
95 °C		103	103	118	118	95 °C		104	104	120	120
		103	103	118	118			98	104	108	120

Now the next step is to estimate the speed of the output disk for example with the average geometrical ratio and with a slip which corresponds roughly to the input torque and to the preload. With an input speed, an output speed and the actual roller positions the roller speed of each roller can be determined in a way that such slip values arise in the contacts that the traction coefficients in both contacts of each roller are exactly the same. Only then the traction forces at each roller are balanced excluding the bearing friction and the roller does not speed up. If all the traction coefficients of all contacts are known, then the input torque and the output torque are known. The lower the output speed, the higher will be the slip values in all contacts and the higher will be the traction forces. With this correlation now the actual output speed can be defined so that the actual input torque can be transmitted.

The torques, which are transmitted from each roller, result from the used traction coefficients multiplied by the normal contact forces and the actual contact radii. The relations between these torques make the power splitting in the variator clear.

Table 2 and fig. 7 illustrate the influence of roller mispositions on power splitting and efficiency. All these calculations are concerned with the same operating point which occurs frequently. The variator runs at overdrive ratio of approximately iV=-0,5. With an input speed of 2500 rpm and an input torque of 250 Nm the input power is round about 65 kW.

In the operating case shown on the left side of table 2 all rollers run with the same tilting angle of -31° within the torus. The output speed is –4935 rpm. The variator ratio is iV=-0.506. The power splits exactly uniform onto all rollers. With 5139 rpm all roller speeds are exactly identical. Due to different geometrical datas of the contacts and different spin values the slip values are different in the contacts to the input disc and to the output disc. The total loss degree regarding spin and slip comes to 1,87%, with 1,15% only for the slip loss.

The datas on the right side of table 2 apply to the same operating state. Roller 3 of cavity 1 has, however, an around 0,5° larger and roller 1 of cavity 2 an around 0,5° smaller tilting angle. The differences in contact radii at the torus discs are only some few tenths mm. For the same input torque now a different output speed (-4929 rpm) and different roller speeds become adjusted. At the roller 3.1 slip and traction coefficient decrease, at the roller 1.2 these both values increase. Roller 1.2 achieves the maximal possible traction coefficient in the contact to the input disc. Now this roller transmits more torque than the other rollers which it disburdens by that. The power splitting changes from initial 6 x 16,66%, when all rollers have identical positions, to an uneven splitting, whereby roller 3.1 transmits only 4,6 % of the power and roller 1.2 transmits 22,3 % of the power. The power quotas of the other rollers are between that. The total loss degree regarding spin and slip, however, increases only to 2,19 % including 1,49% only for the slip loss.

If the mispositions of the rollers 1.2 and 3.1 would be enlarged the power splitting would not be deteriorated considerably, because the roller 1.2 with the highest power quota could not transmit more input torque at all. But the slip value at this roller would increase, so that this roller would find its balanced state only on the declining section of the traction via slip graphs with again a little smaller traction coefficient. At the disburdened roller 3.1 the slip value would decrease further. This could lead to a situation, in which this roller would transmit power in the opposite direction, whereby an internal bracing could occur. Then the other rollers would have to take over of course higher power quotas. Their slip values and traction coefficients would increase therefore. This would increase the slip loss onto 2,25 % and the total loss degree regarding slip and spin to 2,68 %.

Very similar results come out if all rollers have different tilting positions within these tolerances. All in all our investigations on this kind of halftoroidal variators at all possible ratios have shown that mispositions of individual rollers of maximal 0,5° have only small effects on efficiency and power splitting. However, larger mispositions then have increasingly more critical effects.

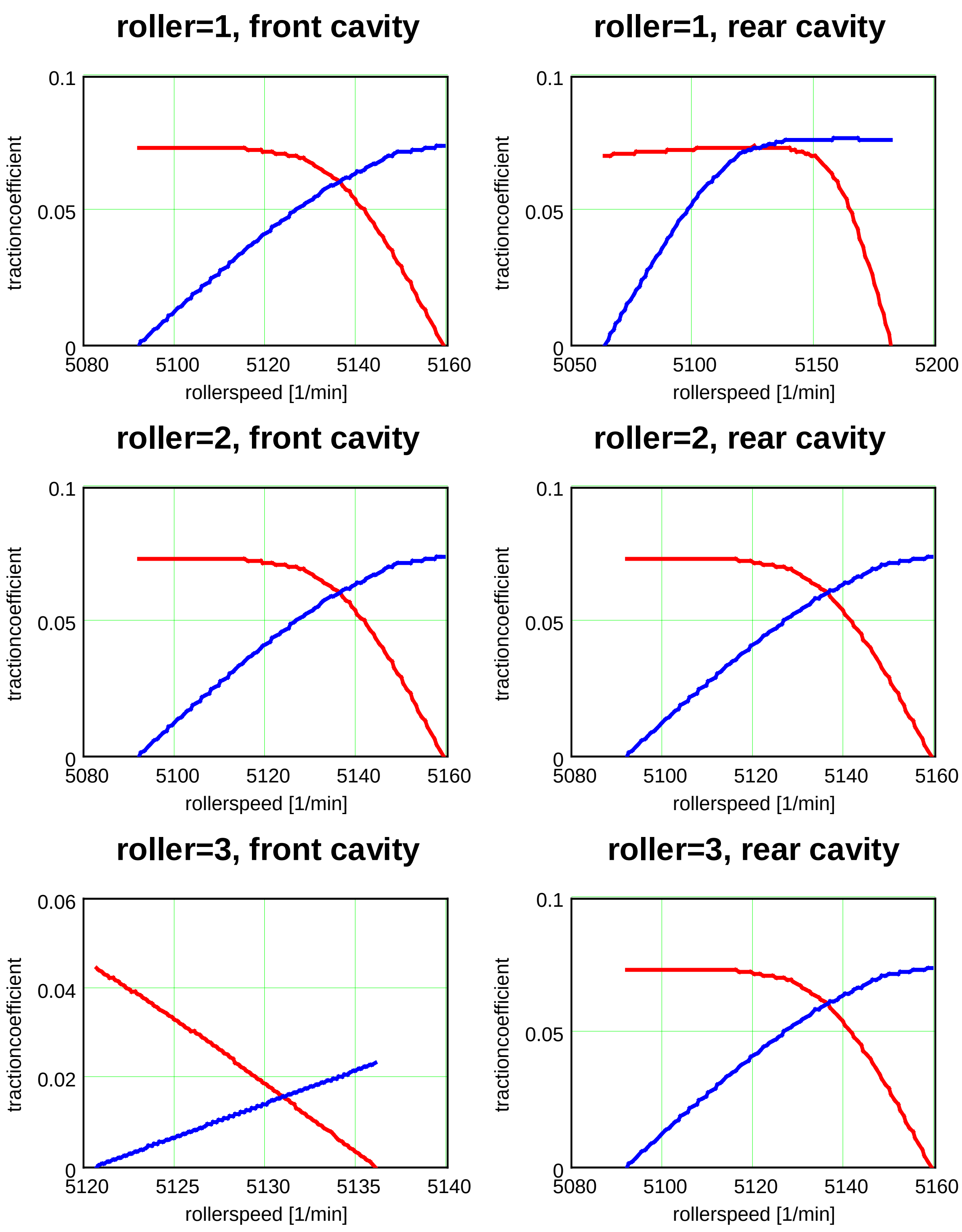

Fig. 8: Traction coefficients via roller speeds of all six rollers for the exemplary operating point with mispositions of roller 3 in the front cavity and roller 1 in the rear cavity

With only one roller a toroidal variator can only run in a stable mode, if the operating point is in both contacts on the increasing bough of the traction/slip curves. The more the operating points come near to the traction maximum, the more flat the traction/slip graphs take their course and all the more the system becomes oscillation-sensitive.

But if the variator has four or six rollers, then an individual roller could quite run in an operating point on the declining bough of the traction/slip-curve because the other rollers hold it in a stable mode. At first if several rollers reach the limit of traction, what as described above can be reinforced also by bracings, the risk of a total sliding increases.

4 Mechanical linked roller suspension with force control

In order to avoid oscillations in the variator and in order to hold the power splitting in the variator maximally uniform, the tilting angles of individual rollers in a halftoroidal variator should not differ more than 0,5° from each other. Figure 9 illustrates a roller suspension for a halftoroidal variator with six rollers, at which the tilting motion of the three rollers of one cavity are synchronized mechanically. Therefore their tilting angles can never differ more from each other than what is pre-set by the variator design and its manufacturing and assembly tolerances.

In a three roller arrangement two trunnions embrace always an angle of 120 °. The tilting angles for the ratio adjustment in such a transmission vary between -40 ° to +40°. By means of small bevel gears two slave trunnions can be operated to one master trunnion. For the small range of tilting angles of +/-40° the bevel gear wheels need only bevel gear wheel segments which can be produced with sufficient accuracy and assembled in the proper position on the trunnions. All three trunnions are mounted in a common yoke so that the clearances in the bevel gears remain small.

In such a type of the roller suspension without caster angle the roller axes always cut the axes of the torus discs, if the roller suspension and the torus discs run concentrically. By that sideslip is avoided, that previously reduces the used traction coefficient and increases the losses from slip and spin.

But up to now the sideslip was used also for changing the ratio. Without sideslip now ratio change must be solved in a different way.

The torque transmission in a variator leads to traction forces at the rollers which with their contact radii can be added up to a reaction torque at the roller suspension. This reaction torque has to be supported to the transmission housing. Now this support can be arranged hydraulically again as it is known from other variator designs. For this purpose the yoke of the roller suspension is not joined directly to the housing, but it is supported on a casing-firm star concentrically to the torus discs. Thereby now the roller suspension can rotate around the torus axes.

With a lever the yoke is joined with a single, relatively large-surface plunger which is mounted in the gearbox control device down in the transmission housing. There the plunger can move +/-35 mm. The lever is joined so flexible with the plunger that the yoke can also be pressed a bit in axial direction towards the elastic star which supports the difference of the axial loads of input and output disc.

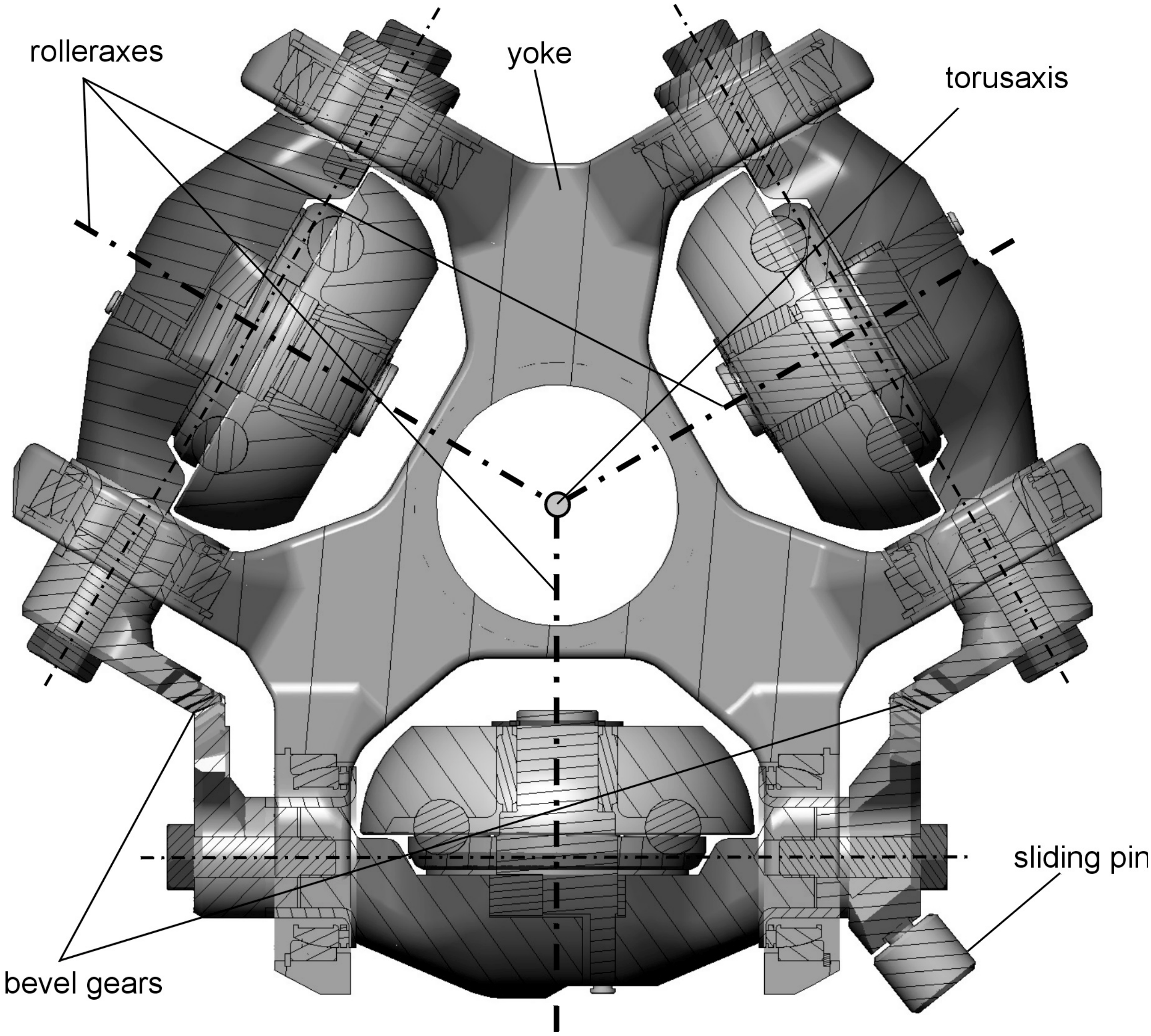

Fig. 9: Yoke of the new roller suspension with three rollers and two bevel gears between the master roller and the two slave rollers

The reaction torque at the yoke now leads to a force at the plunger, which can only be balanced by hydraulic pressure. If here the balance is not given, the plunger displaces and the yoke rotates a bit relatively to the star. The master trunnion or one of the two bevel gear segments which are fixed to this trunnion has a pin that meshes in a groove of a casing-firm slide block. If the yoke with the master trunnion rotates round the torus axis, the master trunnion becomes tilted by the pin which moves in the groove relatively to the housing. By means of the bevel gear segments then the master trunnion drives the both other trunnions to the same tilting angle.

As in the previous variators, which were force controlled, this new system does change the contact radii until the contact forces get such values that a force balance occurs at the plunger again.

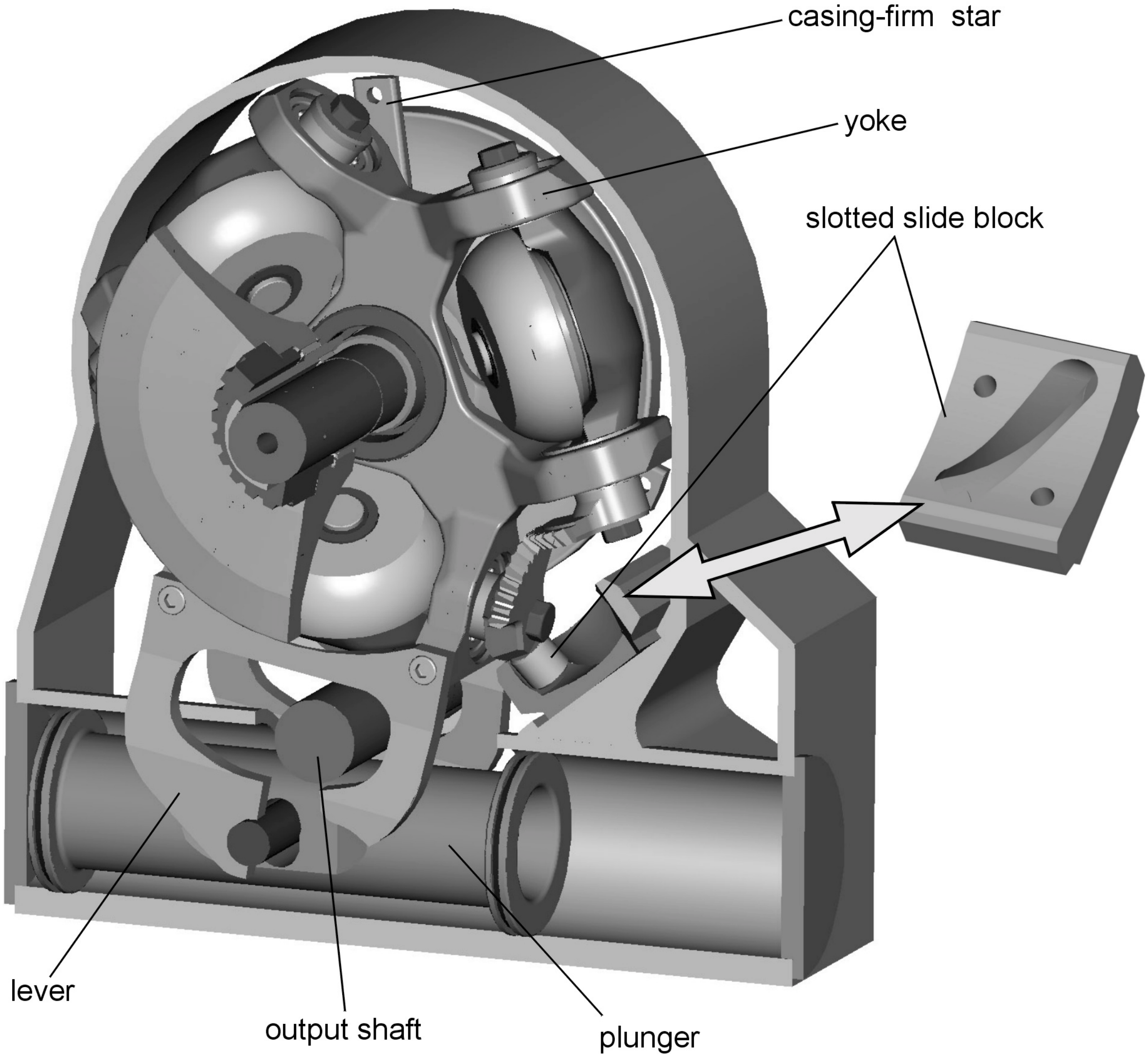

Fig.10: Complete roller suspension with the star to support the yoke, the casing-firm slide block to tilt the master trunnion and the plunger to move the suspension

Furthermore the plunger force is also a reaction force from the yoke via a slide bearing to the casing-firm star.

With the slotted slide block there is now also a defined relation between the plunger movement and variator ratio. And if there is an additional feedback from the plunger position to the control valve for the plunger pressure a ratio control can be created as it is known from the NSK-variator.

The large piston diameter and the long lever arm lead to a small pressure level in the plunger. The piston diameter and its maximal displacements have to be tuned in such a way that the necessary speed for changing the ratio is just reached with the available volume flow. The known variators which are adjusted by the tilting torques from the sideslip adjust usually a lot too fast and have to be damped strongly for a stable operation. In this new variator design the dynamic behaviour is limited by the suspension design and in addition it could be damped at only one single plunger per cavity.

The new control system operates with an essential smaller number of plungers, with a lower pressure and due to that with a smaller leakage. Because of that the energy consumption of the hydraulic system decreases. Thereby the variator efficiency increases probably more than it could decrease due to some differences in the tilting positions of the rollers in comparison to an ideal configuration, because these mispositions can only be very small within the small tolerances and clearances of the bevel gears.

The two cavities of a variator are either hydraulically connected with each other or each cavity has its own control valve. Then they are connected electrically via the controller for the stepper motors for the control valves.

Depending on preload and ratio the torus discs become deformed and the rollers with their trunnions are pressed out of the torus. In order to keep the traction bodies in contact, the hydraulic preload system displaces the discs. In order to be able to follow the discs, the rollers must be able to shift a little bit relatively to their trunnions. In the known NSK-Variator some eccentric bearings take over this function. But in this new variator design such eccentric bearings would effect that the roller axes would not cut the torus axes in all operating points. Unintentional tilting torques would arise from that. Therefore in this new roller suspension a linear guide is used. This linear guide indeed allows a relative movement between a roller and its trunnion but it always keeps the roller axes in cut with the torus axes.

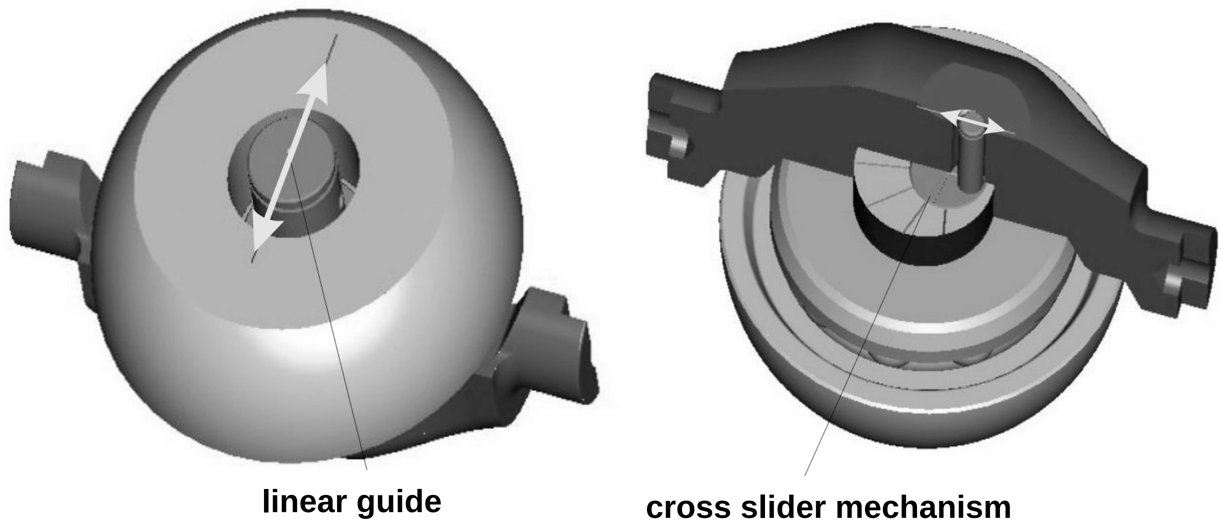

Fig. 11: Generation of a linear guide by using a simple cross slider crank mechanism with two eccentric parts

5 Conclusions

In order to increase the market chances for toroidal CVTs it needs lighter designs and more simple and more stable operating controller units with the retention of all the existing advantages of these gears.

In the here presented new roller suspension of a halftoroidal variator all trunnions of one cavity are supported in only one yoke in a way, that the axes of rotation of all rollers always cut the torus axis. With segments of bevel gears the tilting movements of all trunnion can be synchronized. If the tilting deviations remain smaller than 0,5° the losses increase only a bit in comparison to ideal conditions. In reality they will be smaller than in an oscillating variator today. In such a new variator a ratio change will happen by a hydraulically supported rotation of the whole yoke, which tilts a master trunnion by means of a slotted link. The master trunnion then takes along the other trunnions of a cavity.

A force control or a torque control can be realized with only one single hydraulic piston. Due to the simple hydraulic system the hydraulic losses decrease more than the losses from spin and slip increase due to the possible small (0,5°) malpositions of the rollers. That means that the total efficiency probably increases.

However, much more decisive are the advantages in space and weight of this new variator design in relation to other known variator concepts.

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