# Clamping Systems and Ratio Control of Belt - or Chain CVTs

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*This paper deals with new concepts for the hydraulic clamping system of continuously variable metal-chain or metal-belt drives (CVT) to improve and enlarge the basic functions and to reduce energy consumption. In addition, sufficient clamping forces must be guaranteed for the tough demands in tractors. Starting with a comparison of two actual control systems (for clamping forces and transmission ratio) which are designed for use in vehicles, an improved constant-flow clamping system is presented. It meets the demands of agricultural tractors, i.e. this constant-flow system includes a ratio-shifting mode during standstill of the drive-line.*

*Afterwards, measurement results for a new energy-saving hydraulic clamping system with pressure control are reported. It retains significant advantages of the constant-flow system. For use in tractors, it is very important to maintain immediate response of the clamping pressures even if torque increase is very high. This goal has been reached by substantially redesigned torque sensors. Simulation calculations confirm the energy-saving potential and provide fundamentals for the development of the control strategies.*

**Keywords:**
continuously variable transmissions, CVT, chain converter, tractors, hydraulics, control strategies

## Background and Tasks

The introduction of continuously variable transmissions into the series production of standard tractors (Fendt Vario) and system vehicles (Claas Xerion) has begun [1]. These transmissions are meeting with quickly increasing acceptance. Their significant advantages over gear transmissions are recognized [2]. The transmissions from Claas and Fendt are based on hydrostatic power-split concepts. To achieve high transmission efficiency, two avenues can be pursued: either particularly high-quality hydraulic components can be used to transmit a large part of the power (in this case, few gears are sufficient to cover the entire range of applications), or the percentage of power transmitted hydraulically can be reduced by increasing the number of gears. The latter approach enables cheaper hydraulic engines from general series production to be employed [3]. At the optimal point (without axles), the efficiency of hydrostatic power-split transmissions is slightly over 90% [4].

The belt- and chain transmissions from P.I.V. Reimers (chain converters) and Van Doorne Transmissie (VDT, steel thrust belt) constitute inexpensive technology for continuously variable transmissions. They are based on power transmission through frictional connection. Reference [5] states that, for tractors, the strengths of chain converters (high overall efficiency even without power split) reside in the lower to medium power range. Several cars with chain converters have been put on the market: the latest example is an Audi A6 2.8 (142 kW) with a tension chain converter, which is a further development of the P.I.V. concept used in the Munich research tractor [6]. If these cars are produced in large series as planned, positive effects (performance limits, costs) for tractor drives with tension chain converters can be expected. The principal task in chain converter development is the optimal exploitation of the efficiency potential (greatest mechanical efficiency: about 95%), which is extraordinarily high as compared with other continuously variable transmissions. To achieve this goal, the hydraulic system that controls the clamping pressure and the transmission ratio must be designed appropriately. This in particular requires that excessive clamping pressure be avoided [7] and the energy consumption of the supply hydraulics be optimized [8]. The improvement of chain converter hydraulics has been the goal of research underway at the Technical University of Munich since 1993 as part of the Special Research Programme 365 „Environmentally Friendly Drive Technology for Vehicles“. It is the objective of the Special Research Programme 365 to integrate a transmission featuring a so-called i2 structure with a P.I.V chain converter [9] into a drive train management system and to achieve high transmission adjustment speed, as well as precise transmission setting, in addition to high efficiency. The intended possibility of adjusting the chain converter during standstill while not under load provides great flexibility with regard to the position of the chain converter in the drive train. In order to avoid damage due to the chain slipping through, the clamping system must always be able to generate sufficient clamping force, even during the strongest torque surges, which may occur in the drive train particularly when engaging the clutch [6]. These objectives not only apply to road vehicles, but to a large extent (sometimes even in particular) to the use of chain converters in agricultural machinery, especially in tractors.

## System Comparison of Introduced Clamping Systems

Before an appropriate hydraulic clamping system is developed, a system analysis of the two basic, representative concepts which are currently being favoured for use in square taper washer belt- or chain transmissions is conducted. One variant, which has been developed at the Technical University of Eindhoven for the steel thrust belt transmission from the company Van Doorne Transmissie [10] has been compared with the concept from the company P.I.V. Reimers [11]. The largely separated adjustment of clamping pressure and transmission ratio is common to both concepts.

The constant pressure system in the transmission used at the Technical University of Eindhoven allows the capacity to be automatically adapted to the requirements (**figure 1**). Two independent 3/3 proportional distributing valves serve to set the clamping pressure on the shaft in the clamping cylinder upon which the higher torque acts, while the other shaft is used to set the adjustment speed. The system features electronic control with the advantage of being highly flexible in view of hard- and software. Torque registration based on the evaluation of the stored engine map is a considerable disadvantage because excessive dynamic loads are not sufficiently taken into account. For safety reasons, relatively high excessive clamping is therefore accepted, which has adverse effects on the efficiency of, and the strain on, the chain converter. The heavier stress on the chain causes greater losses, higher surface pressure, and requires more hydraulic power as compared with optimal clamping [7]. An electronic torque measurement shaft would provide more precise signals. However, models that are sufficiently reliable and cost effective are not available. The installation of a torque limiter between the chain transmission and the driving wheels is a recent approach which enables excessive clamping and its negative effect on efficiency to be slightly reduced [12]. Thus, the variator can be protected against output-related load surges, but this design does not allow high torque to be fully transmitted to the driving wheels for vehicle acceleration or for work resistance to be overcome. The simpler hydraulic clamping systems for VDT steel thrust belts [13], which have been known for several years and whose basic concept is always very similar, are not considered in this system comparison because they do not provide any advantages as compared with the cycle analyzed above, while causing only additional disadvantages. The clamping cylinders on the drive- and output side have different piston surfaces and are therefore not suitable for i2 use. Moreover, the volume flow required for speed increase must always be available. Since pressure in the clamping cylinder on the output side is directly coupled with system pressure, it could slump briefly, and the slipping through of the belt or chain might cause considerable damage.

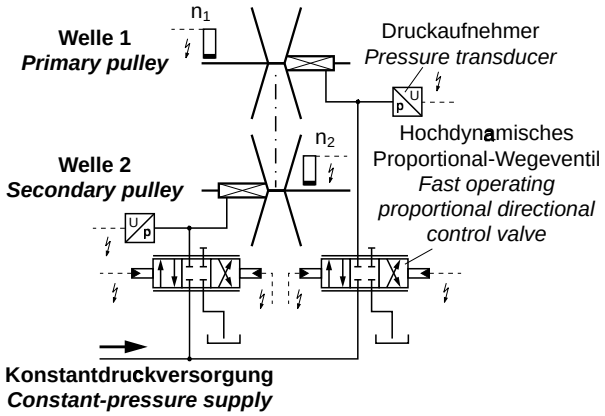



Figure 1: Constant pressure clamping system for metal-v-belt drives (TU Eindhoven)

In the P.I.V. system (**figure 2**), however, constant flow hydraulics (i.e. flow in the cycle is constant (at least 5 l/min), independent of the adjustment of the V-pulleys and the drive rpm) have been used for a long time. This is necessary for the function of the torque sensors, which have a throttling effect. These hydromechanical components in the power flow adjust the system pressure in a highly dynamic way as a function of the torque actually transmitted by the converter. If torque gradients are very steep, they act like pistons and pump additional oil into the clamping cylinders in order to achieve a sufficient pressure gradient. Converter adjustment is entirely independent of clamping pressure generation by the torque sensors. In a valve with four open control edges, throttling analogue to a Wheatstone bridge connection is used to generate the pressure relation between the two clamping cylinders that is required for the desired adjustment speed. In the P.I.V. system, the setting of the clamping pressure and of the converter ratio is therefore virtually independent.

The great advantage of the double sensor system is that it mechanically uncouples the variator from the rest of the drive train both on the drive- and the output side as long as the torque sensors do not reach their limit stops. This is because, due to their design, the sensors can only transmit incoming torque to the converter when in their interior sufficient pressure has built up which communicates directly with the clamping cylinders via the open control edges of the square spool valve. This system does not need any additional torque information and is entirely self-sufficient in this respect. However, the control circuit that maintains the desired transmission ratio has thus far been entirely mechanical. The position of a path disc serves as a feedback variable. The taper establishes a connection between the position of this disc and the transmission ratio. Together with the torque sensors, this provides an extremely robust and reliable clamping system. On the other hand, wear of clamping elements, elastic deformation and alteration of the chain length due to strong tensile or clamping forces cause ratio control errors. In addition, mechanical control requires an inflexible spatial connection between the disc set, the regulator, and the control valve.

The main drawback of this clamping system is the relatively high, permanent volume flow. This is because the volume flow cannot be generated only when it is required for the adjustment of the transmission ratio (which would be optimal under energetic aspects), but, due to the torque sensors, must also be available when the converter ratio is constant. Solutions which allow the necessary volume flow to be reduced to a certain extent (such as the LuK double piston system developed for the Audi drive mentioned above) slightly attenuate this disadvantage [14].

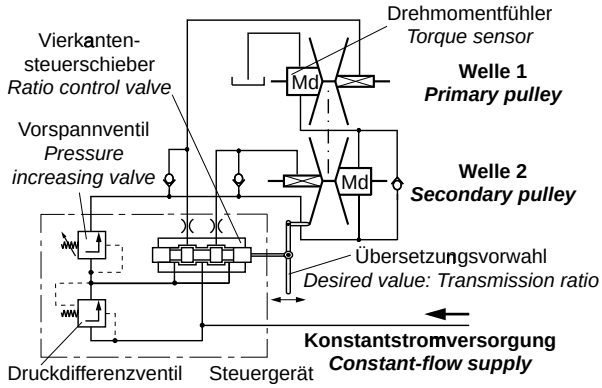



Figure 2: Clamping system for chain converters (P.I.V. Reimers)

## Modified Constant Flow Hydraulics

### *Concept Approach*

Due to reliable clamping pressure adjustment, the P.I.V. constant flow system provides the hydraulic basis for an initial step towards the improvement of the chain converter control described above. In order to adapt it to the increasing networking of the drive train, an electronic system for transmission ratio control with rpm measurement has been developed. The clamping pressure proportional to the torque is set by the two torque sensors very precisely with little excessive clamping. Therefore, this part of the clamping hydraulics is taken over without substantial alterations.

The new hydraulic system for the variator is shown in **figure 3**. Two additional command valves have been integrated into this system. They allow the transmission ratio to be adjusted even when the chain converter is out of operation.

Hydraulic fluid can either be supplied cheaply by the vehicle hydraulics or by a separate pump, which is optimized for the pressure level in the chain converter (up to ca. 60 bar) and the necessary volume flow of approximately 8-10 l/min. Variable displacement pumps, which only supply the desired flow independent of the combustion engine rpm, are most suitable for this purpose. Such a unit has been developed at the Institute of Agricultural Engineering [15]. Instead of the mechanical transmission ratio control of the original concept, it features a significantly more flexible electronic control circuit in a type C 167 control unit [16]. This is the ESX model from the company Sensortechnik Wiedemann. This microcontroller is gaining increasing acceptance for use in agricultural machinery. The valve with a square spool valve has been developed at the Institute of Agricultural Engineering. The spool valve is pilot-controlled hydraulically and integrated into a separate position control circuit. Under normal operating conditions, the frequency of 15-25 Hz at a system pressure of 5 bar enables the chain converter to be controlled quickly and precisely [17].

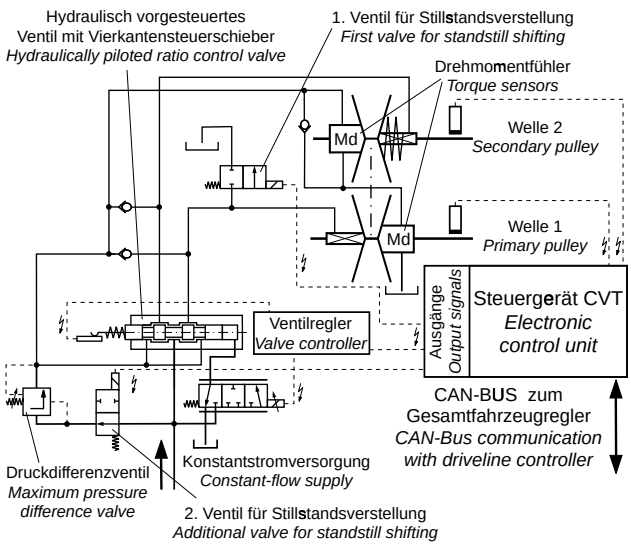



Figure 3: Modified concept for constant-flow clamping system

### *Characteristics of the Transmission Ratio Control System*

With regard to transmission ratio control, the two main requirements to be met by the variator comprise high adjustment speed, which allows it to react quickly to disturbances or alterations of the reference input, and precise setting of the rpm relation of the chain converter so that the rpm relation can be optimally adapted to the operating state. The transmission ratio must be set smoothly because strain due to excessive impulses has an adverse effect on component load and comfort impression.

The setting of the pressure differential valve has the greatest influence on the travel speed of the variator´s path discs (=transmission ratio adjustment). Since, due to the large variation (converter transmission range: 5), fast adjustment leads to high rpm gradients, maximum adjustment speed (and, consequently, set pressure difference at the valve) may not be too high in order to avoid damage to the variator caused by overload as a result of high acceleration moments. Trials showed that the electric test stand drive already reached its maximum torque of 300 Nm at a set value of 20 bar. Drive rpm was 1,000 $min^{-1}$, and the mass moment of inertia on the output side, which was accelerated through variation over the entire transmission range, amounted to 1 $kgm^2$. Approximately 1.85 s were necessary to alter the transmission ratio for speed increase, while reducing the speed required about 0.2 s less because speed reduction is supported by forces acting in the chain converter [7]. With the original control unit (p = 15 bar), this process took more than 2 s. Other trials with set values of up to 25 bar were conducted with significantly smaller moments of inertia on the output side (which approximately corresponds to the conditions in low gears). In these trials, the time required to alter the transmission ratio was reduced to less than 1.3 s for higher speeds and to 1.2 s for lower speeds (**figure 4**). Higher rpm favour short adjustment periods of the variator (due to the diminishing azimuthal angle of incline of the chain). p = 18 bar allows the same times do be achieved at nan = 2,000 $min^{-1}$ as 20 bar at nan = 1,000 $min^{-1}$. Further improvement of quick adjustment would require that the maximum pressure difference would be able to be modulated according to the transmission ratio. At low output rpm, for example, higher values are needed than during overdrive. A proportional pressure valve or switching between several valves with fixed settings would enable this problem to be solved. When control dynamics are high, it is the task of the transmission controller to keep the remaining control deviation at a minimum and to minimize the influence of disturbances (such as torque, rpm, temperature, etc.) (**figure 5**). Since, due to its ring-shaped adjustment cylinder, the chain converter exhibits I-behaviour, using a simple P-controller is

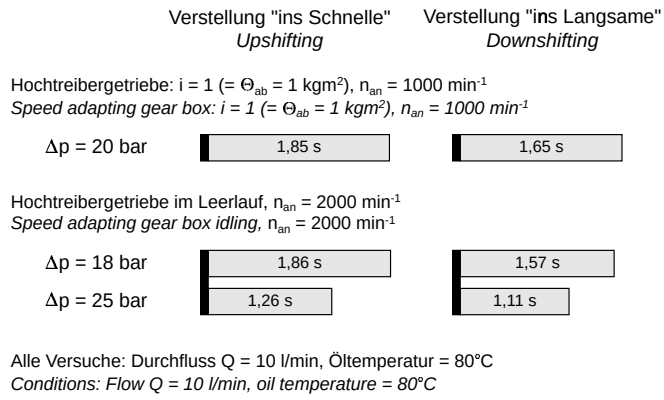
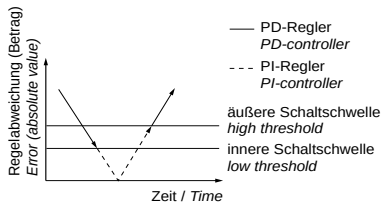



Figure 4: Maximum CVT power shift speed (full ratio)

theoretically sufficient to establish a control circuit (PTI-behaviour) with compensation. A PD controller allows the control deviation to be reduced even faster. In practice, however, different influences such as friction and the load level cause a certain control deviation to remain. Since, especially for large control deviations, the use of a global PID controller is extremely critical with regard to stability due to the I-behaviour of the open line according to the Nyquist criterion, a simple controller with variable structure was employed, which exhibits a PD structure if control deviations are large and a PI structure if deviations are very small. At the Institute of Agricultural Engineering, such a controller has already been introduced successfully in a similar project [18]. The PD controller quickly reduces large control deviations. When deviation falls below a certain threshold, the PI controller is activated. Only at a second, higher threshold does the system switch back to the PD controller. The introduction of this hysteresis allows constant switching between these two controllers and the resulting instabilities to be avoided. In the case of digital programming, this is relatively easy to realize. Measurements confirmed that this controller structure is not affected by disturbing load changes.

- Kettenwandler mit Anpress-System: I-Strecke
  *Chain converter with clamping system: Integral plant*
- Variante 1: PD-Regler
  *Configuration 1: PD-Controller*
- Variante 2: PD/PI-Regler mit Strukturumschaltung
  *Configuration 2: PD/PI-controller (structure switching)*




| Vergleich zwischen den Lastfällen $M_{ab}$ = 100 Nm und $M_{ab}$ = 0 Nm *Comparison between the loads $M_{ab}$ = 100 Nm und $M_{ab}$ = 0 Nm* | relative bleibende Regelabweichung *relative steady-state error signal* | |
|---|---|---|
| $ü = n_{ab}/n_{an}$ | PD | PD/PI |
| 0,5 | -1,3 % | 0,0 % |
| 1,0 | -1,1 % | 0,0 % |
| 1,5 | -0,9 % | 0,1 % |
| 2,0 | -0,8 % | 0,2 % |

Abtriebsseitiges Massenträgheitsmoment = 1 kgm², $n_{an}$ = 1000 min⁻¹, Q = 10 l/min, Öltemperatur = 80°C
*Inertia torque on output end = 1kgm², input speed $n_{an}$ = 1000 min⁻¹, flow Q = 10 l/min, oil temperature = 80°C*

Figure 5: Structure of ratio control minimizes load depending on steady-state error

### *Variator Adjustment in Standing Vehicles*

For driving in practice, it seems necessary that the converter ratio can be adjusted while the vehicle is standing, for example when the drive-off ratio must be set again after a car brakes on the road or after a tractor stops on the field with a lowered plough. For this purpose, two variants are conceivable: adjustment with or without engaging and disengaging the clutch.

If the main drive-off clutch is situated between the chain converter and the output shaft, this clutch can be disengaged. The variator is driven by the combustion engine, and its ratio is thus set for driving off. In this case, one can drive off again by re-engaging the main driving clutch. Drawbacks include very limited possibilities of designing the transmission structure. Moreover, an additional clutch between the combustion engine and the chain transmission is needed. This clutch is already necessary when the engine is started since for the desired service life of the chain converter to be guaranteed the clamping system must be supplied with fluid, i.e. the combustion engine must first start the hydraulic pump and then drive the chain converter.

If the main drive-off clutch is situated between the variator and the driving engine and if there is no possibility of ratio adjustment in the standing vehicle, reaching the drive-off ratio would require a lengthy shifting process: after disengaging the main clutch, the drive train would have to be interrupted at a clutch behind the converter. Then, the main clutch would be engaged again, and the converter ratio would be set for driving off. Afterwards, the main clutch would be disengaged, the back clutch would be engaged, and finally the main clutch would be used for driving off.

The second case in particular shows the great advantages which result from the variator being adjusted while the shafts are not moving (using the cylinder pressure). For this reason, the use of up to two command valves for the control of the pressure in the clamping cylinders was already given consideration when designing the new valve with a square spool valve. Tests have proved that these measures allow the variator to be adjusted while the vehicle is standing. The length of time it took to shift the chain on the input shaft (shaft 1) from its largest to its smallest running radius was measured for different configurations. With the valve and the square spool valve alone, it is virtually impossible to adjust the ratio while the shafts are not moving. Only when the first valve is opened in addition for adjustment during standstill (figure 3) and the pressure in the clamping cylinder of shaft 1 is thus reduced to ambient pressure, can the path disc be pressed outwards without meeting strong resistance (**table 1**).

Table 1: Time for standstill shifting (idling, full ratio coverage

| Oil flow | Time for full range shifting | |
|---|---|---|
| | First valve set | Both valves set |
| Q = 5 l/min | 2,8 s | 2,7 s |
| Q = 10 l/min | 2,6 s | 1,8 s |

input speed $n_{an}$ = 0 min⁻¹, oil temperature 80°C

The pressure force required to move the path disc on shaft 2 depends on the setting of the pressure differential valve (ca. 20 bar). The pressure differential valve is blocked by opening the second valve. This means that significantly higher pressure becomes available. However, pressure should not exceed approximately 60 bar in order to avoid damage to chain converter components. However, this only results in higher adjustment speed if sufficient system volume flows are generated so that the clamping cylinder of shaft 2 can be filled fast enough. The question of whether the pre-tension spring in the clamping cylinder is integrated into shaft 1 or 2 does not have any significant influence. All in all, the P.I.V. chain converter is very suitable for ratio adjustment in standing vehicles because the angle of the square taper washer is far enough from self-locking effects.

### *Behaviour at Deep Temperatures*

With the aid of a cooling unit and an insulating hood over the test stand, the behaviour of the chain converter and of the modified constant-flow clamping hydraulics was studied at oil temperatures of up to -20° C (air temperature: ca. -25° C). The proof of unrestricted function of the hydraulically pilot-controlled valve/square spool valve as an actuator for the transmission controller is particularly important. For this purpose, especially the position control circuit of the spool valve (and, in particular, the pilot-control system) must work reliably. The trials were conducted with small volume flows because pressure in the hydraulic circuit was already relatively high at these volume flows and caused the oil to heat up early. The trial described here (**figure 6**) illustrates that oil viscosity diminishes (as shown by pressure level reduction over time) and that the transmission control system works without malfunction despite the very low temperatures. Measurements regarding standstill adjustment of the chain converter at low temperatures are not available.

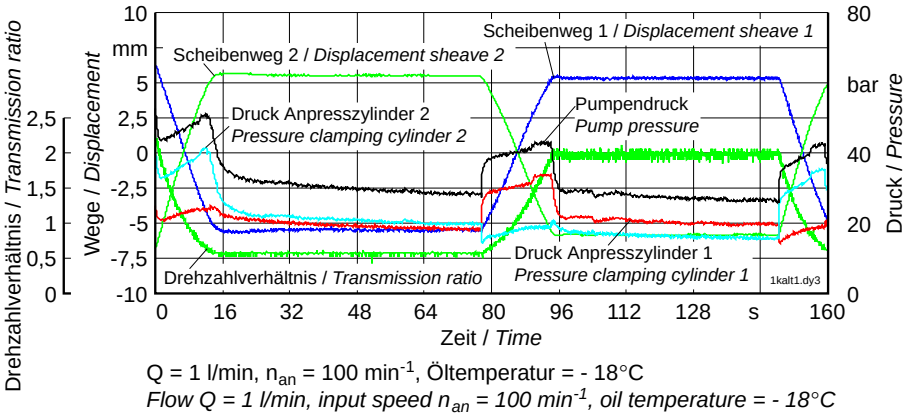



Figure 6: Ratio control at low temperatures

### *New Hydraulics Concept Based on Pressure Control*

As shown in earlier publications, a pressure-controlled hydraulic system with a variable displacement pump theoretically enables energy consumption to be kept as low as possible because pressure and volume flow can be adapted to the current requirements. Simple simulation calculations regarding measurement rides with the Munich research tractor under performance control [19] showed great energy-saving potential [20]. Simulations were also conducted for a significantly larger selection of measurements regarding field- and transport work with the same tractor. These models also take pressure losses in pipes and at throttle points into account, as well as flows of leaking oil, and a pump efficiency of 0.62 (measured efficiency of the variable displacement pump). All in all, it was shown that, as compared with an optimized constant flow system, the use of a pressure-controlled clamping system causes the percentage of hydraulic energy in the total amount of energy generated by the combustion engine to diminish from ca. 1.7%...2.5% to 0.4%...0.6% (depending on the kind of tractor use). This means that average overall chain converter efficiency can be expected to increase by ca. 1%. If overall efficiency is 90%, this improvement translates to a loss reduction of 10%.

### *Design and Functional Principle*

For use in road vehicles and, to an even larger extent, for use in tractors and machines, the hydraulic system with clamping pressure control must be able to follow even very steep torque rises so that seizing up due to chain slipping can be avoided [6]. The torque sensor of the P.I.V. constant flow system is a very efficient solution to this problem. At the Institute of Agricultural Machinery, it has been redesigned in such a way that it can also be employed in a pressure-controlled clamping system. The new design is extensively described in reference [17]. Its most important characteristics include electronic torque measurement based on the evaluation of sensor distortion and the display of the pump function by a sensor chamber locked by a movable plate. The movable plate and the sensor plate are connected with a diaphragm spring set (**figure 7**). The hydraulic circuit diagram of the clamping system was kept very simple (**figure 8**).

Under normal circumstances, the pressure control valves are used to adjust the pressure in the clamping cylinders depending on the desired transmission ratio and the current torque (established by electronic measurement). The system pressure provided by the pump is only as high as required for safe valve operation (ca. 5 bar). However, it may not fall below the minimum level needed for the currently used adjustable vane pump with electronically assisted pressure control because otherwise pump adjustment becomes impossible. With other supply concepts, this restriction might no longer apply.

If torque rises very quickly, the pressure control circuit cannot react fast enough due to the delay of up to 100 ms (which is almost entirely caused by the valves). The required rapid pressure increase is therefore generated otherwise: for a short time, the axial force generated by ball ramps from the torque is larger than the pressure force in the sensor chamber. Therefore, the movable plate discharges oil volume into the associated clamping cylinder. Pressure rises immediately (**figure 9**).

### *Trials on Test Stands*

To study the pumping effect, one half of the clamping system was mounted on a component test stand specially developed for this purpose. This system comprises the pump, the pressure regulating valve, the new torque sensor, and the clamping cylinder. A dry disc brake behind the sensor allows very steep torque rises to be generated. A torque measurement shaft for measurement recording is fitted between the sensor and the brake. The volu-

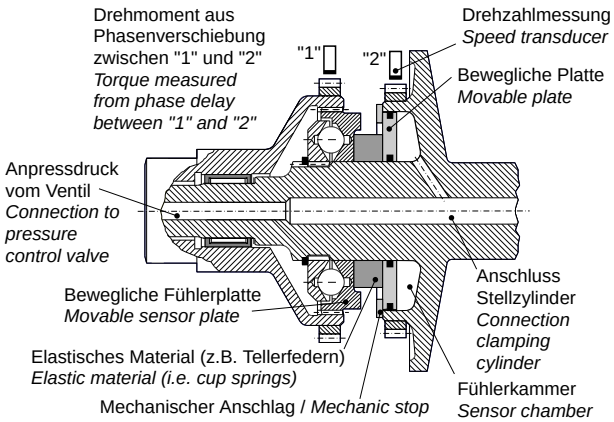



Figure 7: New concept for the torque sensor with "pumping function"

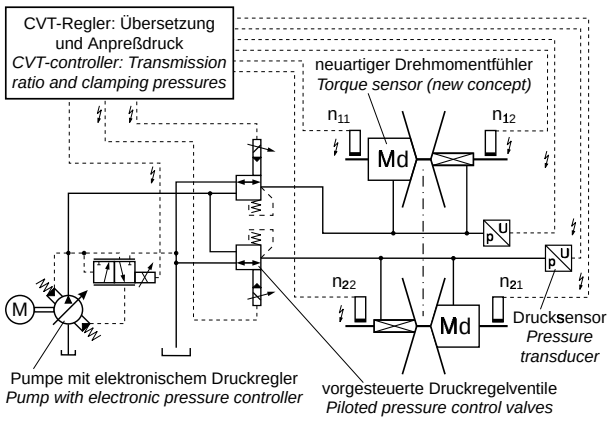



Figure 8: New concept for CVT hydraulics with pressure control

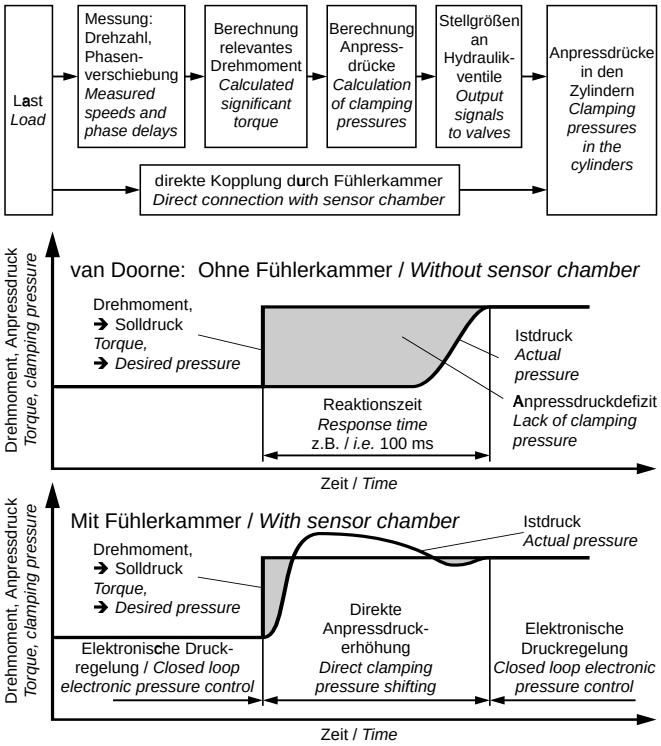



Figure 9: Basic operation mode of "pumping function"

me of the clamping cylinder is modelled using a cylinder of the same size. The trials are conducted to answer the questions of whether the pump movement causes pressure to rise sufficiently, how long this increase lasts, and how much time is required to refill the sensor chamber.

The oil discharged from the chamber is not exclusively pumped into the clamping cylinder. Part of it flows through the pressure regulating valve and back into the tank because the valve tries to maintain the set pressure and hence to reduce excessive pressure.

In the first trial series, the duration and the level of the pressure increase were measured (**figure 10**). For this purpose, the set value for the pressure regulating valve remained constant. Therefore, pressure increase is caused by the pump alone. The pumping effect lasts approximately 200 ms. In any case, this is sufficient to cover the time until the valve has readjusted. If CVT fluids are used, a pressure increase to 8 bar is sufficient for the torque of approximately 90-105 Nm to be transmitted for a short time. In the target system, the set value for the clamping pressure is increased as soon as a torque jump is recognized. **Figure 11** shows how the pressure regulating valve and the pump function work together in this case. As of the beginning of the torque jump, pumping causes the clamping pressure to rise to 10 bar without delay until, after a delay of approximately 40 ms, the pressure regulating valve can adjust the pressure to the new set value.

### *Further Studies*

After the test stand trials have shown that the new torque sensor functions well, the next step is to complete and then examine the strategies for pressure control. The use of dynamic simulation models helps to accomplish this task. Figure 11 shows a comparison of measurement and simulation. All in all, the model developed with MATLAB/SIMULINK provides good simulation of the real behaviour.

Subsequently, the thus validated models of the pressure regulating valve and of the torque sensor will be integrated into the simulation of the complete pressure-controlled hydraulic system and joined with the already existing model of the complete chain converter test stand [21]. This model will be used to finish the development of the control strategies, which will then be examined as an entire system in experiments on the chain converter test stand (equipped with the new hydraulics).

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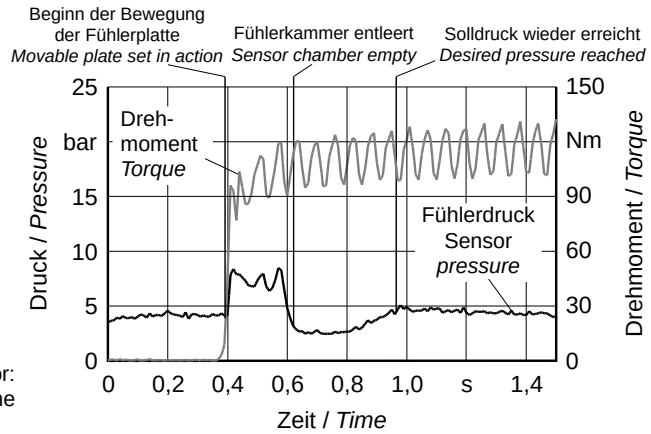


Figure 10: New torque sensor: Maximum duration of the "pumping function"

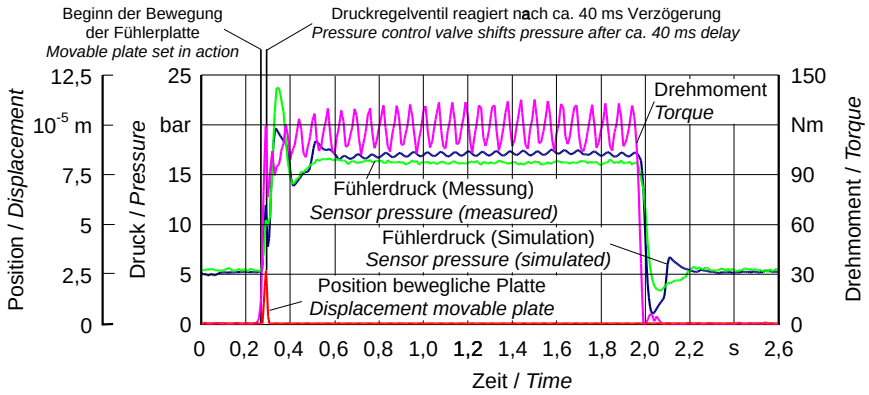


Figure 11: "Pumping function": Compensation for the time delay in the pressure control loop

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