

Control and Operating Behavior of Continuously Variable Chain Transmissions

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ABSTRACT

A new adaptive controller layout for chain converters for universal use in different applications and set-ups is being presented. Based on the control structure chosen for synchronization of the claw couplings of a chain converter type i^2 -transmission during range shifts, a universal control layout was developed. Both the conventional clamping system for the well known PIV¹-based chain converter and a newly developed pressure controlled clamping system were taken into account. As the most important non-linearity the ratio of clamping forces between driving and driven pulley (ζ -value) for steady state operation was mapped in a characteristic diagram and is being adapted during normal operation. Besides improvements in performance and better availability of information for diagnostic purposes, an open loop control of the rate of ratio change becomes possible.

INTRODUCTION

Chain converter type of transmissions today can be found in modern passenger cars as well as in applications for industrial use (e.g. paper processing industry) and inland or maritime navigation (generator drives). Different applications of chain converters lead to different demands on the control of the transmission.

NEW APPLICATIONS

Continuously Variable Transmissions (CVT) also have become popular for modern standard tractors in Europe for middle and higher power range. Though being very different in detail, the main concept in this area is always the same: A combination of hydrostatic units with variable displacement (pump and/or motor) is being used as a hydrostatic CVT in a power split arrangement [1, 2]. Due to the combination with the mechanical path, high overall efficiencies of the CVT-gearbox can be

achieved, but most of the realized concepts lead to a rather complex and expensive gearbox design. Therefore in the very cost sensitive field of low power standard tractors there is a need of rather simple CVT concepts that nevertheless can compete in efficiency. One of the options might be the use of chain variators that have already proved themselves in modern passenger cars [3]. Although chain variators also could be used in the CVT path of power split drive trains, costs and complexity as well as the typical reactive power in conjunction with the limited torque capacity of chain variators remain problems that need to be solved. A simple structure with direct drive is to be preferred, as it can be found in passenger cars. Particularly in Japanese markets dozens of different CVT models with push type belts are being offered, whereas in Europe the pull type chain CVT started to gain in importance [4]. The input torque capacity of these gearboxes with chain CVT has reached 300 Nm (Audi A6 multitronic V6-3.0, 162 kW) and is still being improved. The spreading of these chain converters is sufficient for passenger cars (around 5–6), but not enough for tractor use. It can cover the main working range of 4 to 12 km/h, but not the total forward speed range of about 2 to 50 km/h (without creeper speeds) at nominal engine speed. This can only be achieved by using additional range gears. Depending on the chosen gearbox layout, power shifting between different ranges is possible (see below), but comes in conjunction with increased costs and complexity of gearbox design.

IMPROVED REQUIREMENTS ON CVT CONTROL

In either case there are increased demands on the control of the CVT during range shifts, because for example speed synchronization is required. The aim of the control strategy in that case is to achieve and sustain a certain speed ratio at the variator, independently from any disturbing influences such as high torque gradients. Depending on the specific application there are different control modes respectively control variables to be considered (**Table 1**).

¹ Company information: PIV drives GmbH (former P.I.V. Antrieb Werner Reimers GmbH & Co. KG), Bad Homburg, Hessen, Germany

Table 1: Classification and use of CVT control strategies

Control variable	Application
(quasi-) stationary control modes	
speed ratio	manual CVT mode (virtual gears/Tiptronic®) range shift (synchronization) restrain transmission ratio boundary ground speed PTO (tractor)
input speed	automatic CVT mode (engine speed control, – standard mode for passenger cars, – economy mode for tractor use)
output speed	generator drive paper processing (coil) vehicle speed (tractor)
dynamic control mode	
shift speed	drive train control torque control (all vehicles)

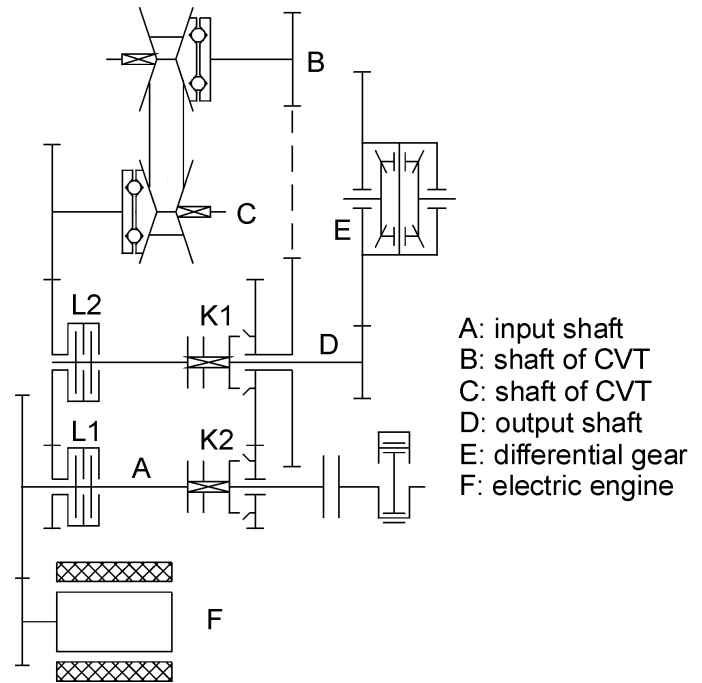


Fig. 1: Structure of the i^2 -gearbox of the Autarkic Hybrid
 $(i_{\text{GEARBOX}} = i_{\text{VARIATOR}}^2 = [4.75]^2 = 22.5)$

When being used in mobile working machinery (especially tractors), all of the control modes shown in Table 1 might be needed according to the different operation modes of a single machine: “Ratio based” control strategies are needed to realize for example tractor specific ground speed PTO, but also for range shifting or manual CVT mode. Classical “speed based” control strategies are both used for automatic (eco) mode during transportation work (engine speed control) and during field work (load limiting control or output/vehicle speed control). Due to its basic effect on comfort and wheel torque the rate of ratio change di/dt , during transient operation of the CVT [5] has also to be taken into account (shift speed control).

Provided there is a speed sensor for both input and output shaft of the CVT, input and output speed control can be treated as a special case of speed ratio control. Moreover, as will be shown later on, shift speed control can also be realized as a simple extension of the presented speed ratio control. Thus beginning with a controller for speed synchronization during range shifting processes, a universal layout for speed ratio control is presented, leading to a concept for controlling the shift speed of the CVT.

CVT CONTROL

CONTROL DESIGN FOR RANGE SHIFTS IN AUTARKIC HYBRID

The pull type chain variator has already been subject of research projects at Institute of Agricultural Machinery of Technische Universität München (TUM) for tractors [6] and passenger cars in the past: In 2002 results have been presented together with the presentation of the “Autarkic Hybrid”, a common large research project of several TUM departments, conducted by Prof. Höhn. The “Autarkic Hybrid” is parallel hybrid concept

passenger car, based on a standard Opel Astra Caravan with Diesel engine, which was equipped with additional components such as an electric engine (9,2 kW @ 6000 rpm), supplied by and charging a 120V-battery and a specially developed chain CVT gearbox with wide spreading in i^2 -design [7]. **Fig. 1** shows the structure of the gearbox.

The gearbox is characterized by two driving ranges. In first range (V1) the clutch L1 is closed and variator shaft C is driven by the engine(s). The synchronized collar coupling K1 is also closed and shaft B of the CVT is connected to the final drive (E). In overdrive of range V1 at a speed ratio of approximately $i = 0,458$ the rotational speed difference in all four clutches disappears and the so called synchronous point is reached. At that point the second synchronized collar coupling (K2) is being closed and the first wet clutch (L1) can be opened. In that situation (SYN) power still can be transmitted, now without the chain variator being engaged. Since the gearbox has a fixed gear ration at SYN, acceleration is possible only by raising the engine speed. Further acceleration leads to an operation in second range: By closing the clutch L2 while opening the clutch K1, the driving and driven pulleys are inverted compared to first range and second range (V2) is reached, starting in underdrive again. Thus an overall spreading of the gearbox of about 22.5 is realized.

In order to provide fast, comfortable and successful range shifts it is essential to exactly control the speed ratio at synchronous point. Differences in speed ratio of a few thousand parts are already preventing successful switching operations. When controlling the speed ratio at

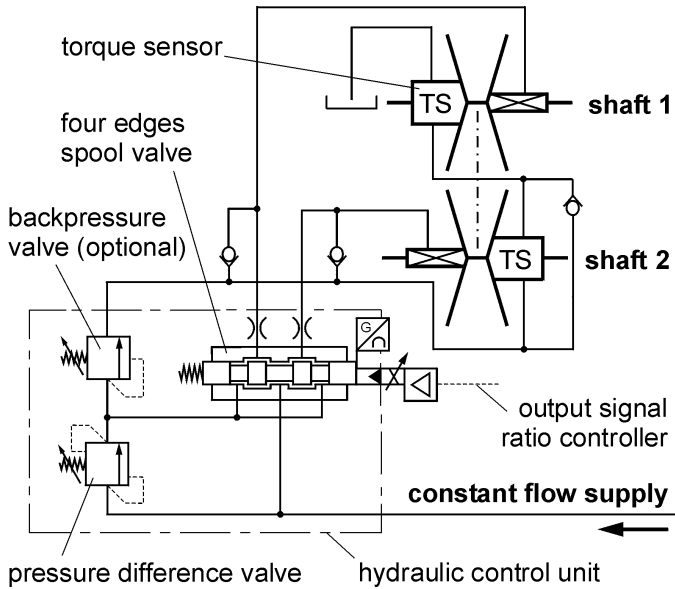


Fig. 2: PIV-based constant flow clamping system in Autarkic Hybrid

steady state, on one hand, the clamping forces need to be high enough to prevent damage by slip of the chain, on the other hand a certain ratio of clamping forces has to be achieved, the so called ζ -ratio (ratio of clamping forces between driving and driven pulley) [8] in order to sustain a certain speed ratio of the variator. In the used constant flow clamping system (Fig. 2), a base level of clamping pressure in the pulleys that prevents the chain from slipping is automatically achieved by throttling a constant oil flow in the so called torque sensor (Fig. 3).

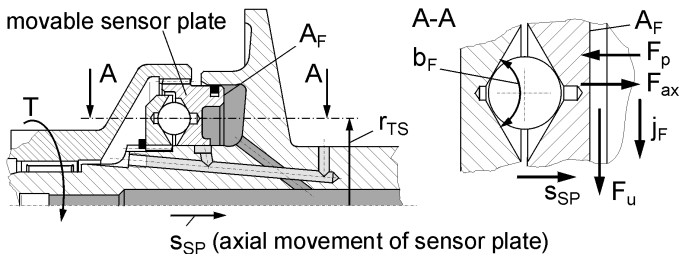


Fig. 3: Hydraulic torque sensor in constant flow clamping system

The torque sensor is a hydro-mechanically working unit that needs no superior control. It is flown through by a rather constant oil flow that passes from sensor chamber at higher pressure level (dark grey) via an throttling orifice to the secondary side (light grey) that is connected to another torque sensor or to tank. The size of the throttling orifice is directly given by the position of the movable sensor plate s_{SP} . The sensor plate itself is used to transmit the torque of the shaft over a ball ramp mechanism that leads to an axial force F_{ax} proportional to torque and an axial movement of the sensor plate. According to the axial position, the size of the throttling orifice is influenced and thus, at a given oil flow, pressure sets accordingly. Since a new equilibrium of

axial force F_{ax} and the product of the area of sensor plate A_F and pressure in sensor chamber is reached automatically, this leads to a torque sensor pressure p_{TS} at the control valve that is approximately proportional to the higher torque at both CVT shafts. Moreover, in case of extremely high torque gradients, the sensor plate acts like an additional oil pump in order to raise clamping pressure. The additional control valve (four edges spool valve; comparable to a hydraulic Wheatstone bridge) is used to raise one of the clamping pressures by further throttling torque sensor pressure. The other clamping pressure is approximately at torque sensor pressure level (Fig. 4).

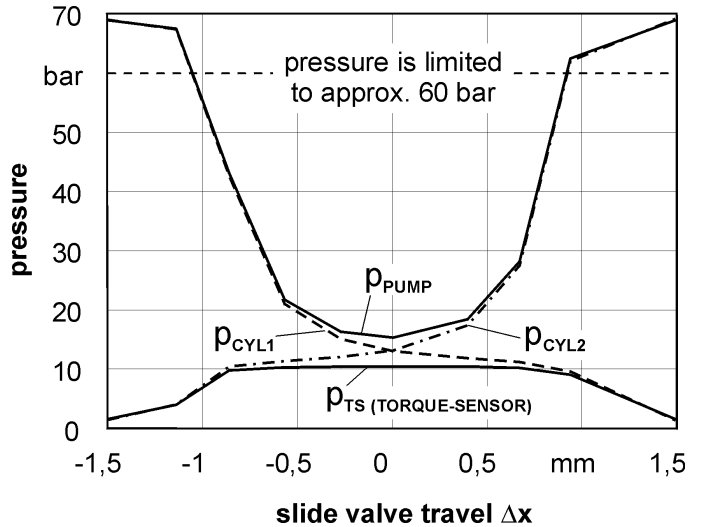


Fig. 4: Characteristic of the control valve (four edges spool valve)

The control valve is used to reach the required ζ -ratio of clamping forces for the desired speed ratio. The required ζ -ratio itself, among others depends on transmitted torque and speed ratio (see following section). Moreover the actual pulley speed leads to centrifugal forces affecting the chain and clamping forces (rotating hydraulic cylinders), so it can be seen as another disturbance variable. Assuming a rather narrow speed range in normal operation when doing range shifts and a fixed given speed ratio (SYN), the remaining disturbance variable in Autarkic Hybrid still is torque. Since the variator is changing between engaged and disengaged mode during range shifts, there are extremely high torque gradients. Furthermore torque increases due to twisting when three clutches are closed together for a short time during a shifting process and speed ratio doesn't fit exactly synchronous speed ratio.

Consequently a range shift leads to changing ζ -ratios during shifting process and thus difference in speed ratio that prevents from successfully completing or at least delays the range shift. The standard speed ratio controller (closed loop) can only react on differences in speed ratio and needs a certain time to compensate any deviation. Therefore the control was further expanded to represent a disturbance feedforward control. Since the

information about current input torque is available by measuring the pressure of the torque sensors (two sensors in series), and the manipulated variable is the position of the control valve which sets a difference in clamping pressures, the influence of torque was mapped in a characteristic curve which has the pressure of the input torque sensor as input and the required position of the valve for steady state operation as output (Fig. 5).

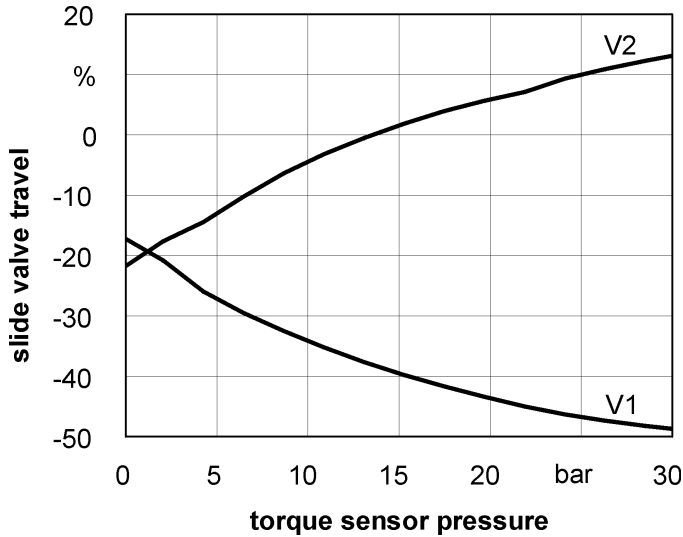


Fig. 5: Manipulated variable depending on torque (pressure) for steady state

This was also done to minimize the required calculation steps for the disturbance feedforward controller and allow short clock cycles with typical microcontroller hardware (16bit, 20 MHz). Finally the influence of speed on the required valve position was investigated and also integrated into the controller layout (Fig. 6). The taken measures resulted in a significant improvement of quality and reliability of range shifts.

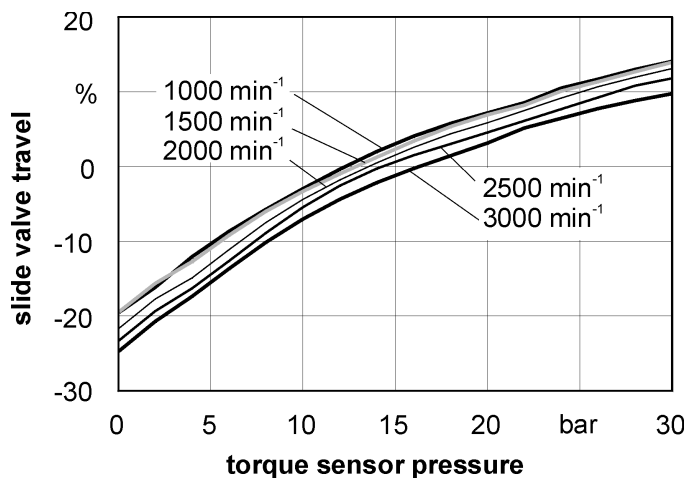


Fig. 6: Manipulated variable depending on torque and speed for steady state (V2)

EXPANDED CONTROL LAYOUT FOR UNIVERSAL USE IN CHAIN VARIATOR APPLICATIONS

When controlling the speed ratio not only in one single point, but over the whole range of spreading, a controller, variable in structure (Fig. 7), has proved itself in different CVT applications before:

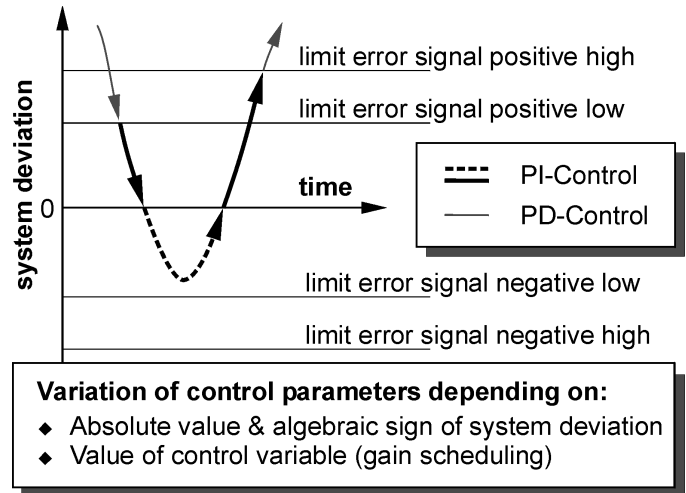


Fig. 7: (Linear) Controller design, variable in structure

By using different sets of parameters, depending on the size of system deviation and the actual value of the control variable (gain scheduling), a stable system behavior can be realized, though using integral type sets of parameters to avoid a steady state-error signal. Moreover parameters can be found more easily. By resetting any stored deviation values and the manipulated variable to predefined values when reaching the given or even additional error signal limits, a very smooth system behavior without overshooting and hunting (improved ride comfort) can be achieved.

The underlying disturbance feedforward control structure (see above), which already took into account the disturbance variables torque and speed was extended by an additional dimension (speed ratio) in order to cover the whole spreading of the transmission. To achieve better flexibility, the disturbance variables now are reduced to their origin: Clamping forces are used instead of valve positions or pressures for basic calculations.

The ζ -value for the steady state operation can be seen as the most important non-linearity of the system. It was mapped in a characteristic diagram, Fig. 8 (CVT ratio and input torque as input variables). Using the ζ -ratio in conjunction with the minimum clamping force, the required clamping pressures can be calculated. Both the influence of speed (centrifugal forces) and the characteristics of the respective clamping system (characteristic curve of the valve) are taken into account by mathematic compensation. All other influences are covered by the ζ -diagram. Since a sufficient amount of

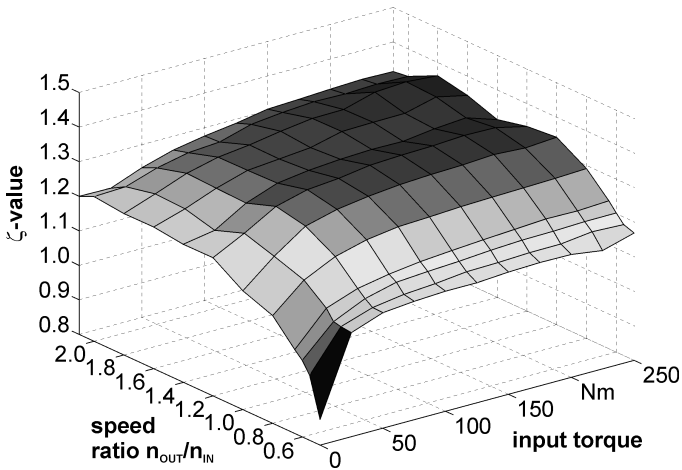


Fig. 8: ζ -map gathered and used by the speed ratio controller

measured values for the ζ -ratio was not available and the values differ between different types of variators, it is being acquired adaptively by the controller during normal operation. The criterion for adaptation is the output of the linear ratio controller, which is supposed to be leading to zero in steady state (Fig. 9). Different types of weighting functions for the adaptation process were investigated in order to minimize the required computing power (aim: 16-bit μC @ 20 MHz).

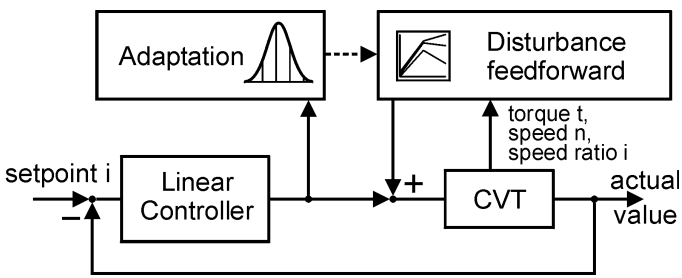


Fig. 9: Structure of the speed ratio controller

The control design can easily be fitted to different gearbox designs and clamping strategies. Both the conventional clamping system for the known PIV-based constant flow system (see above) and a newly developed pressure controlled clamping system [9, 10] were taken into account (Fig. 10). At a first step a base level of clamping pressure (respectively minimum clamping force) was assumed for the pressure controlled system, which is proportional to the higher torque of both pulleys (according to the behavior conventional PIV-system).

The control structure is also suitable for further improvements of the gearbox in terms of efficiency by automatically reducing the amount of over clamping when using the so called ζ -max method (as far as permitted by the clamping system). In that case the base level of pressure is successively being reduced until the observed ζ -value reaches its maximum.

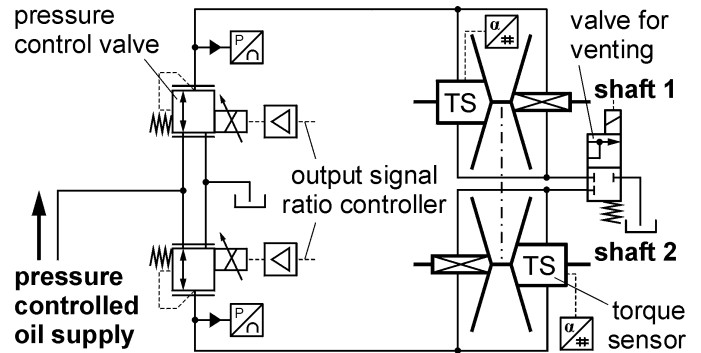


Fig. 10: Hydraulic circuit of the developed pressure controlled system using a new torque sensor design.

Since the observed parameters react very sensitively to many interferences, they are qualified to be used for diagnostic purposes. Mechanical problems show very early in the characteristic curves of the ζ -value, for example.

CONTROL OF THE RATE OF RATIO CHANGE

During transient operating conditions of the CVT, for example when raising the engine speed in order to get more power for vehicle acceleration, it is favorable to control the rate of ratio change.

Fig. 11 shows a simplified model, which is often used to describe the characteristic effects within the CVT drive train. The model consists of a moment of inertia J_1 rotating at the angular speed ω_1 of the engine and represents the inertia of all rotating parts running at the engine side of the power train (e.g. crank shaft and primary pulley). Vehicle mass and inertia of the rotating parts at the wheel side of the drive train are represented by the moment of inertia J_2 running at wheel speed ω_2 . The two masses are coupled by the CVT running at a certain speed ratio i (ω_1/ω_2) and inducing a change in speed ratio di/dt .

During normal operation the superior power train control basically calculates the desired engine speed and torque which is determined according to its operation line in the characteristic engine map (e.g. eco, sport) and driver's demand (accelerator pedal deflection) [11]. Engine speed is then achieved by the help of the CVT controller (input speed control of the CVT) and torque is set by the engine controller. Since running in an economic operating mode using a fuel saving operation line, leads

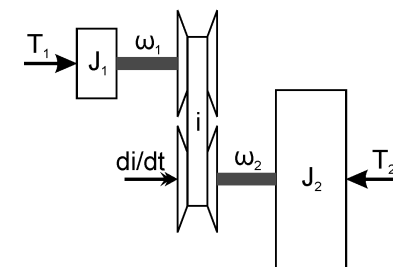


Fig. 11: Model of a CVT drive train

to rather low engine speeds and high torque near the maximum load line (area of low break specific fuel consumption). An increase in power demand consequently leads to higher engine speed which is achieved by changing the ratio i , the CVT is running at:

Any change in speed ratio of the CVT predominantly leads to an acceleration or deceleration of the rotational mass J_1 (due to the relatively low moment of inertia at the engine side compared to the inertia at the wheel side) and therefore leads to a change in engine speed. Thus available power at the wheel is lowered or raised according to the power needed for changing engine speed, the vehicle might even react converse to the drivers demand when the change in engine speed (respectively change of speed ratio at the CVT) happens too quickly: In extreme circumstances the vehicle would slow down in the first moment even when the kick down detent is being engaged. Therefore the possibility of controlling the rate of speed ratio change di/dt is getting essential [5].

Concerning the top level control of a CVT power train there are several publications known from the past [3, 5, 12]. At the level of hydraulic variator control a short structured overview on different approaches is given in [13]. Mostly they are concerned with VDT-based belt drives and its clamping system. Detailed newer papers on the control of chain drive CVTs and its special PIV-based clamping system with torque sensor(s) and clamping cylinders, typically equal in area, can rarely be found. In [14] an estimation of the shift rate is realized using neural networks in order to compensate the expected power loss at the wheel with the electric motor of the hybrid drive line. Most approaches for belt drives use steering or control of the oil flow into the primary clamping cylinder in order to control the transmission ratio or the rate of ratio change [15]. Other concepts use electrically driven servo-pumps to control the pressures and the flow (shift rate) within the oil circuit of the CVT [16].

The results presented here for chain transmissions are related to some extent to the findings stated earlier for belt drives. Investigations on the rate of ratio change already were carried out in [17].

The use of gathered ζ -curves by the new controller layout, as it was presented above, leads to better performance in ratio-based control strategies. The control of the rate of ratio change di/dt becomes possible by a simple modification: Instead of closed loop controlling the speed ratio, an open loop control using the proportional behavior between shift speed and the difference between actual clamping forces and steady state clamping forces can be realized. Since the steady state forces are known from the gathered ζ -curves in conjunction with the minimum clamping force requirements, this difference can easily be computed according to desired rate of speed ratio change. The modified control structure can be seen from **Fig. 12**.

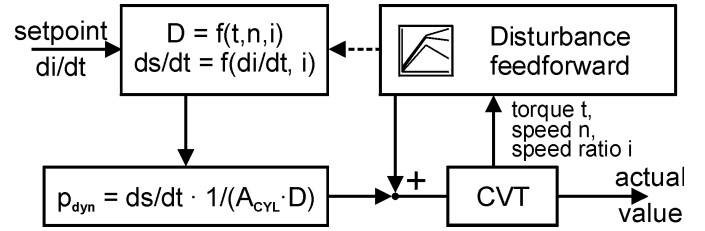


Fig. 12: Structure of the shift speed controller

Instead of the manipulated variable of the linear controller, which usually raises one of the two clamping pressures, an additional force F_{dyn} that leads to a change in speed ratio is added to the clamping forces (pressures) derived from the disturbance feedforward block in order to sustain the actual speed ratio. With respect of the damping coefficient D , the shift rate ds/dt is proportional to the force F_{dyn} :

$$ds/dt = F_{dyn} \cdot D \quad (1)$$

The geometrical characteristics of the variator are used to compute the required shift speed ds/dt (axial velocity of the pulley) according to the desired rate of speed ratio change di/dt , **Fig 13**.

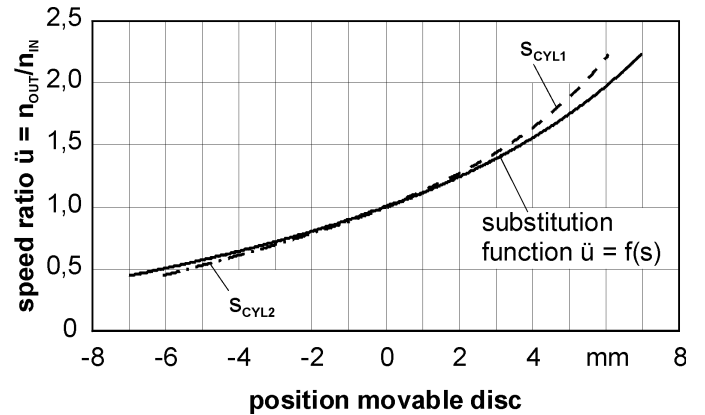


Fig. 13: Correlation between axial position of pulley and speed ratio [9]

Since the centrifugal effects in the clamping cylinders are already compensated by the feedforward branch, the force F_{dyn} can be expressed by the product of the area of one clamping cylinder A and the hydraulic pressure p_{dyn} . Thus the required pressure p_{dyn} that needs to be set according to the desired shift speed can easily be computed:

$$p_{dyn} = ds/dt \cdot (1/A \cdot D) \quad (2)$$

As can be seen from Fig. 14, the damping coefficient D is mainly a speed dependent value. Shift speed increases proportionally according to dynamic force F_{dyn} , Fig. 15. The influence of torque is rather low., Fig. 16.

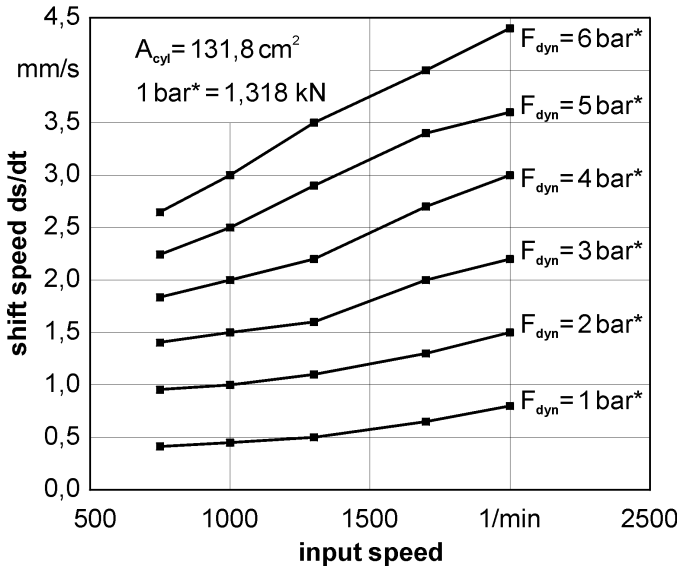


Fig. 14: Correlation between shift speed ds/dt , input speed and dynamic force F_{dyn} .

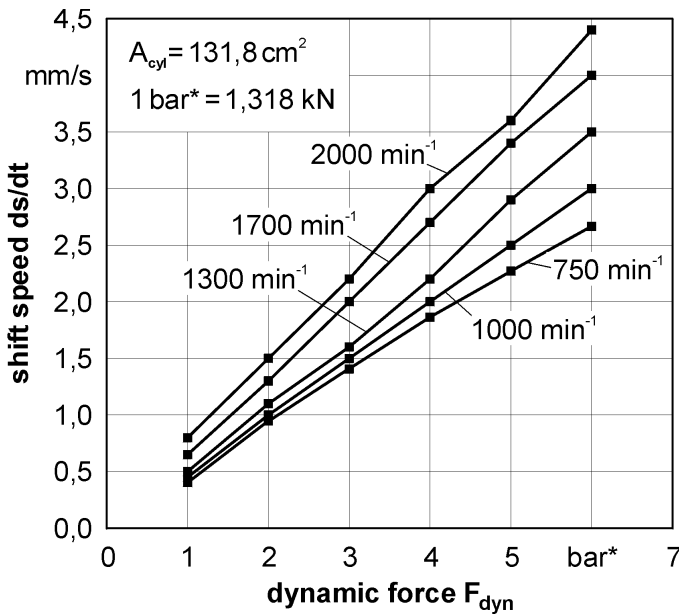


Fig. 15: Correlation between shift speed ds/dt and dynamic force F_{dyn} .

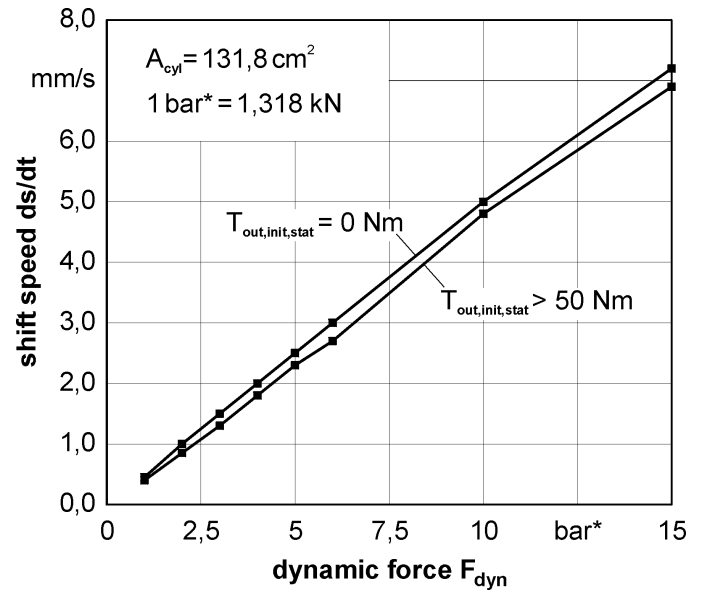


Fig. 16: Correlation between shift speed ds/dt , input torque and dynamic force F_{dyn} .

Fig. 17 shows several measurements of shift speed using the new controller layout. Starting in underdrive at a point in time of 1 second, the control structure is switched over to the open loop control of shift speed, applying constantly different dynamic forces F_{dyn} . The delay in rise time that can be seen from the measurement is due to a running average (200 ms) that was used for determination of shift speed ds/dt . Near overdrive ratio, the control switches back to the classical ratio control structure to maintain the actual speed ratio.

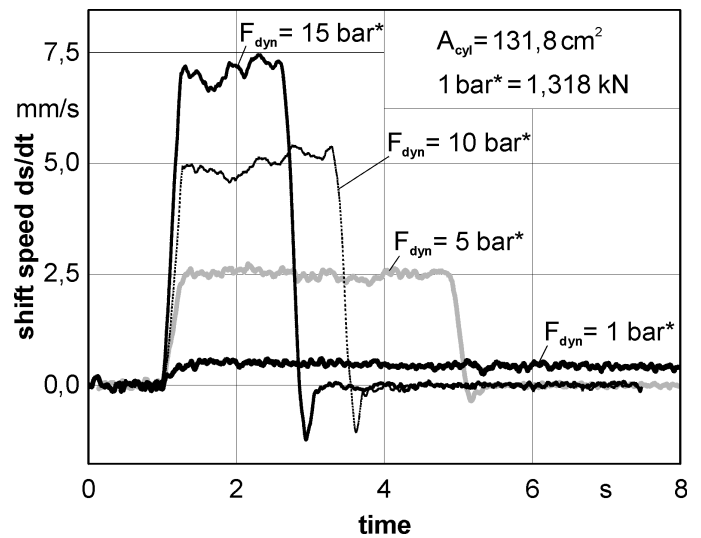


Fig. 17: Results of an open loop control of the shift speed ds/dt (rise time due to a running average of 200 ms)

DEVELOPMENT TOOLS AND CONTROL HARDWARE

The dynamic and stationary behavior that was found and implemented into the control structure and proofed for control purposes (simple description with low requirements on computing power) is also used as a model description with high accuracy for the CVT for further simulation purposes. Although the development of the control structure was carried out using a Rapid-Control-Prototyping-System running under Matlab/Simulink® in conjunction with xPC-Target for the realtime environment, the aim to develop a control structure that can be used with typical hardware for electronic control units in mobile machinery was fully reached. The control unit (freely programmable controller "ESX" by Sensor-Technik Wiedemann, Kaufbeuren, Germany) chosen for running in a test vehicle was programmed in the classical way using C-routines in a first step. Moreover an ESX-library for use with an automatic code generator (TargetLink 2.0) within Matlab was developed. Thus easy reuse of the Simulink models already developed before for programming of the control unit is getting possible.

CONCLUSION

In a CVT drive train it is necessary to control as well quasi-stationary conditions (speed and speed ratio) as well as the transient behavior of the variator (di/dt). A control layout based on disturbance feedforward was used. In this regard using the ratio ζ of clamping forces for steady state in all working conditions and an automatic adaptation of the used ζ -map during steady state conditions, gives the opportunity not only to improve performance, but also to realize further functions (diagnostics, efficiency optimization). Moreover the control of shift rate is getting possible in a very simple way using the results of the investigated operation behavior for chain drives. The findings correspond to a certain extent with former approaches and results for belt drives, concerning the use of the difference of actual clamping forces and steady state clamping forces in conjunction with a damping coefficient in order to realize a certain dynamic behavior.

The work resulted in a fast and simple, nevertheless accurate model description and control structure for chain drive CVTs. The latter accepts setpoints for speed ratio i as well as for the rate of ratio change di/dt . The approach is also suitable for application in products of series-production.

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