IMPROVING THE DYNAMIC RESPONSE OF A LINEAR HYDRAULIC DRIVE USING A DIGITAL VALVE CONTROL

Helmut Kogler[1], Rainer Haas[1], Georg Keintzel[2]
[1]Linz Center of Mechatronics
Altenberger Straße 69, 4040 Linz, Austria
[2]Primetals Technologies
Turmstrasse 44, 4030 Linz, Austria
E-mail: helmut.kogler@lcm.at

ABSTRACT

Hydraulic cylinders are mostly used for linear drive actuation of heavy loads because of their high force density. Furthermore, servo valves are often used in a closed loop control in order to achieve a high dynamic movement of the drive system. In case of huge loads the natural frequency of the drive is rather low, which in turn slows down the closed loop response in spite of the use of fast servo valves. This problem is shown by simulations for the step control of wrapper rolls in a down coiler in hot rolling mills. In order to overcome the limited bandwidth of the closed loop servo valve control in this paper a digital valve control is presented, which achieves the maximum physical dynamic response of a linear hydraulic drive. It is a type of feedforward control, which is designed to the natural frequency of the drive such that almost no oscillations of the wrapper roll will remain. The original problem of the wrapper roll was transferred to a linear drive axis with similar dynamics. Measurements on a laboratory test rig show a significant improvement in the dynamic response with the high dynamic digital control compared to a digital single pulse control. The results and relevant effects are discussed and, finally, a conclusion and an outlook for next steps in development are provided.

KEYWORDS: high dynamic, digital, switching, hydraulics, optimal, input shaping

1. INTRODUCTION

Hydraulic drive technology is well known for achieving high forces with simple and robust actuators. In particular hydraulic cylinders are often the first choice to realize linear motion or pivoting of heavy loads. In industrial plants the power is often provided by a common constant pressure supply system and the individual actuators are controlled by valves. If a fast and precise drive performance is required, then hydraulic servo valves are often preferred because of their fast response dynamics within the range of a few milliseconds. In such a case the hydraulic drive is mostly closed loop controlled in order to achieve the desired dynamic performance. However, huge mechanical loads in combination with the compressible fluid in the drive cylinder often result in a dynamic system with low natural frequency, which constitute a limit for the overall dynamic performance of the drive. Today in modern industrial plants advanced controllers are used for hydraulic drive applications, but still often simple PID controllers are a common and simple control strategy for certain hydraulic drives. Furthermore, in case of the use of a digital CPU the parameters of the proportional controller are still often tuned by hand according to the meanwhile historic but simple design rules presented in [1]. However, in such cases the high response dynamics of the servo valves cannot be exploited for stability reasons of the closed loop system. An opportunity to overcome the limited dynamics of the closed loop control of heavy loads would be a feedforward control, which is designed to the dynamic behavior of the drive system. Furthermore, the intention to achieve a time optimal behavior suggests a so called bang-bang feedforward control, which is based on input-shaping techniques as, for instance, according to [2]. Such a feedforward control is designed to the natural frequency of a flexible drive system and yields to a digital control signal for a movement with reduced vibrations in the end position. Since the feedforward control results in a digital valve actuation now digital hydraulic valves can be used instead of expensive and sensitive servo valves.

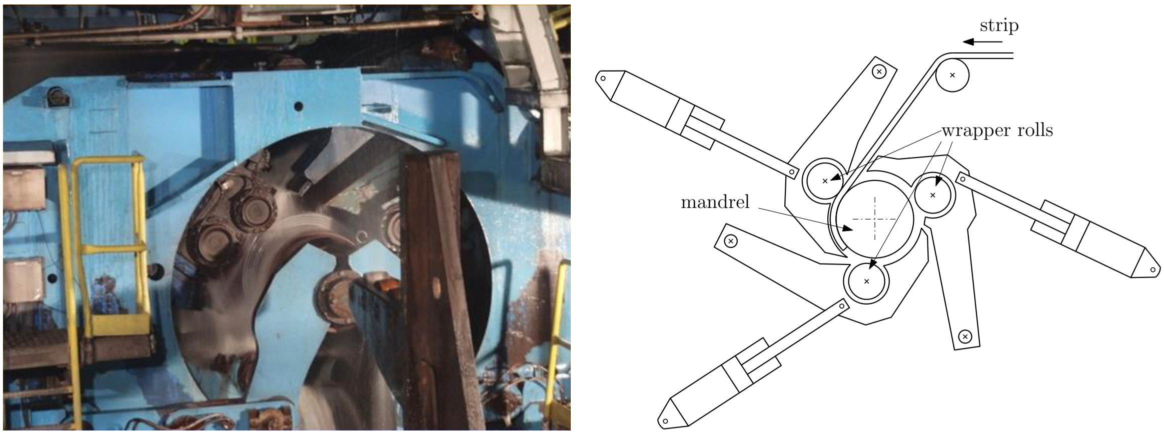
In this paper the problem of limited dynamics at a closed loop servo hydraulic drive is demonstrated for a wrapper roll in a rolling mill. Furthermore, the digital hydraulic switching concept based on input-shaping techniques is presented. In a next step the initial problem of the wrapper roll is transformed to linear hydraulic laboratory drive axis with similar dynamics of the wrapper roll. After a discussion of the experimental results the contribution ends up with concluding remarks and an outlook in further steps in development.

2. STEP CONTROL IN A DOWN COILER

In hot rolling mills the inserted hot steel slabs are rolled to long flat steel products with a length in the range of several hundred meters or, in case of thin strips, even up to a few kilometers. At the end of the mill the hot steel strip is formed to coils in a so called down coiler like depicted in Fig. 1 in order to be prepared for further production steps.

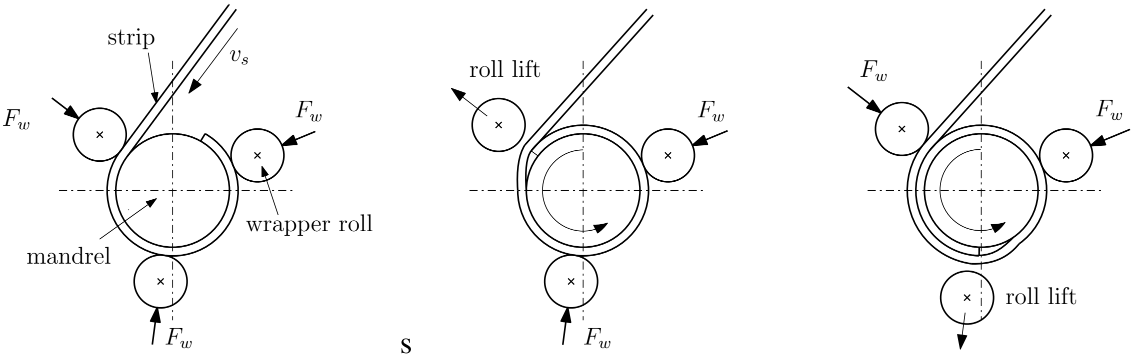
2.1. The Wrapping Process

After a ready wound up coil is separated from the mandrel the next steel strip must be threaded into the down coiler. For this purpose so called wrapper rolls are moved against the mandrel and the head end of the strip is directed by certain guide plates around the mandrel. Today, in common down coilers typically 3 or 4 hydraulically actuated wrapper rolls are arranged around the mandrel, which are used to apply a certain wrapping force against the strip such that the mandrel drive is able to build up a certain strip tension for a proper coiling quality, which is illustrated in Fig. 2a. After the first winding, when the head end of the strip passes the first wrapper roll for the second time the wrapping force must be released and the wrapper roll must be lifted; otherwise the head end would create



(a) Coiler in operation (b) Functional scheme

Figure 1: Down coiler in a hot strip mill



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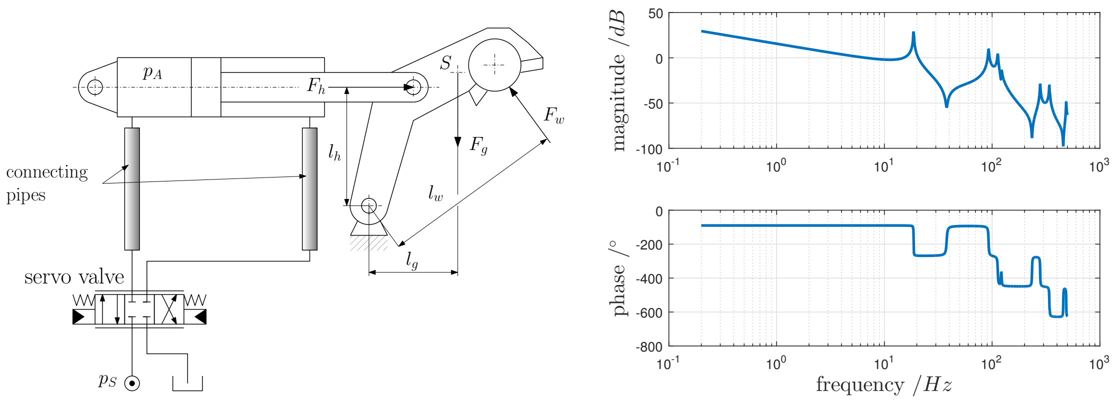
(a) Threading the strip (b) Step of 1st wrapper roll (c) Step of 2nd wrapper roll

Figure 2: Principle of the step control

hit marks in the coil, which would result in reduced product quality and increased waste material. This process is called step control and is depicted in Fig. 2b. When the strip head end has passed the wrapper roll then the roll is moved against the strip in order to build up the wrapping force again (see Fig. 2c). The control performance of the step control of all wrapper rolls must ensure that at least one wrapper roll is applying the desired wrapping force. After a few windings, when the mandrel has built up full strip tension the wrapper rolls are pivoted to the outer rest position. More information about coiling of hot rolled flat materials can be found, for instance, in [3].

2.2. Simulation of the Hydraulic Step Control

In Fig. 3a the functional scheme of one exemplary hydraulic wrapper roll actuation system is illustrated. The mechanical part of the wrapper roll is pivoted by a hydraulic differential cylinder, which is controlled by a servo valve. In Fig. 3a also the connecting pipe lines are depicted, because they also have influence on the dynamic behavior of the wrapper roll. The associated transfer function of the roll position with regard to the valve opening is depicted in Fig. 3b according to the parameter set of Tab. 1. In fact, the dynamics of the wrapper roll is nonlinear, but the steps in the wrapping process are sufficiently small in order to justify a linear consideration. The first resonance peak shows a frequency of



(a) Functional scheme

(b) Transfer function

Figure 3: Servo hydraulically actuated wrapper roll

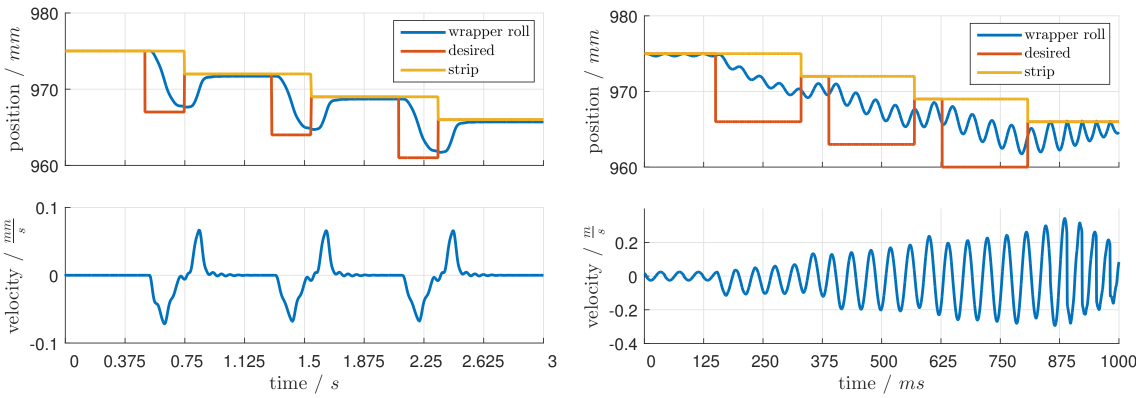
Parameter	Value	Parameter	Value
fluid compressibility	$E_o = 12000\,\text{bar}$	piston diameter	$d_p = 260\,\text{mm}$
mass of the pivoting arm	$m = 4600\,\text{kg}$	rod diameter	$d_r = 180\,\text{mm}$
rotary inertia of pivoting arm	$\Theta = 7800\,\text{kg}\cdot\text{m}^2$	cylinder length	$l_C = 1.1\,\text{m}$
leverage of gravity	$l_g = 0\,\text{m}$	pipe length	$l_P = 4\,\text{m}$
hydraulic leverage	$l_h = 1\,\text{m}$	pipe diameter	$d_P = 49\,\text{mm}$
leverage of wrapping force	$l_w = 0.6\,\text{m}$		

Table 1: Parameters of a wrapper roll

approximately 18 Hz according to the mechanical inertia and the hydraulic stiffness of the cylinder in the wrapping position pivoted to the mandrel. All higher pole/zero dynamics in this transfer function result from the pressure build up equation in the rod sided chamber and the dynamics of both connected pipes.

In fact, in modern hot rolling mills advanced controller concepts are applied, but for the sake of simplicity in this study the responses of the wrapper rolls are compared to the closed loop control with a simple proportional controller. Furthermore, in reality the strip speed corresponds to the desired strip reduction mainly in order to maintain the necessary temperatures during the rolling process, i.e. the thinner the strip the higher the strip speed at the output of the last mill stand. However, in order to show the basic operating principle of a propper stepping with the simple P-controller a lower strip speed of $v_S = 3\,{}^{m}\!/\!{}_{s}$ for a strip thickness of $h_S = 3\,\text{mm}$ was considered in the first simulations depicted in Fig. 4a. In the upper diagram the incoming strip, the desired and the actual position of the wrapper roll related to the cylinder stroke are depicted. In the lower diagram the piston speed is illustrated. The simulation study is focused on the stepping process, which means that no wrapping force is considered.

In Fig. 4b the simulation results of a stepping process for a strip thickness of $h_S = 3\,\text{mm}$ at a strip speed of $v_S = 10\,{}^{m}\!/\!{}_{s}$ is presented. At such high rolling speeds the step control can often not be realized properly with simple control strategies. It must be remarked that in the simulation model the damping was reduced to a minimum in order to consider a worst case scenario. For a better control performance at high rolling speeds model based concepts must be considered. Of course, a state space controller or even nonlinear



(a) Step control at $v_S = 3 m/s$ (b) Step control at $v_S = 10 m/s$

Figure 4: Simulation results of a servo hydraulically actuated wrapper roll

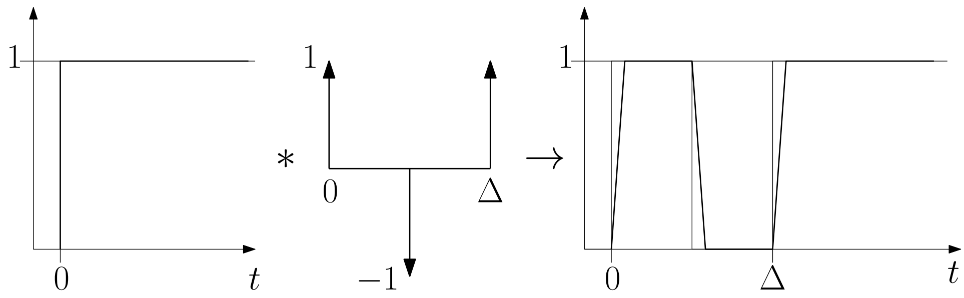


Figure 5: Unity magnitude zero vibration input shaper (UM-ZV)

control concepts like for instance [4] would yield an improved dynamic response for the servo actuated wrapper rolls. However, it is clear that the maximum dynamic response is limited by the first natural frequency according to spring-mass oscillator consisting of the wrapper roll mechanics and the hydraulic stiffness in the cylinder. According to literature regarding optimal control theory like, for instance, [5] it can be shown that a time optimal response and, thus, the maximum dynamic response can be achieved with a so called bang-bang feedforward control. This means that the hydraulic valves are used only in digital mode; they are either fully open or closed. Such bang-bang controls can be easily designed by input shaping methods like proposed in, for instance [6], where the control of cranes with reduced sway is presented. With regard to the wrapper roll such a digital control concept suggests to replace the expensive and sensitive servo valves by cheap and robust digital hydraulic valves as considered in the following section.

3. DIGITAL SWITCHING CONCEPT

The digital control for time optimal system performance is designed to the natural frequency of a flexible system. In case of the wrapper roll the dominating eigen frequency ω_d results from the mechanical part of the roll in combination with the compressibility of the hydraulic fluid in the cylinder. According to theory and in order to avoid remaining oscillations during and right after the intended movement a unity magnitude zero vibration (UM-ZV) input shaper according to [6] as depicted in Fig. 5 is applied to the system. The trapezoidal signal on the right hand side indicates an approximation of the switching

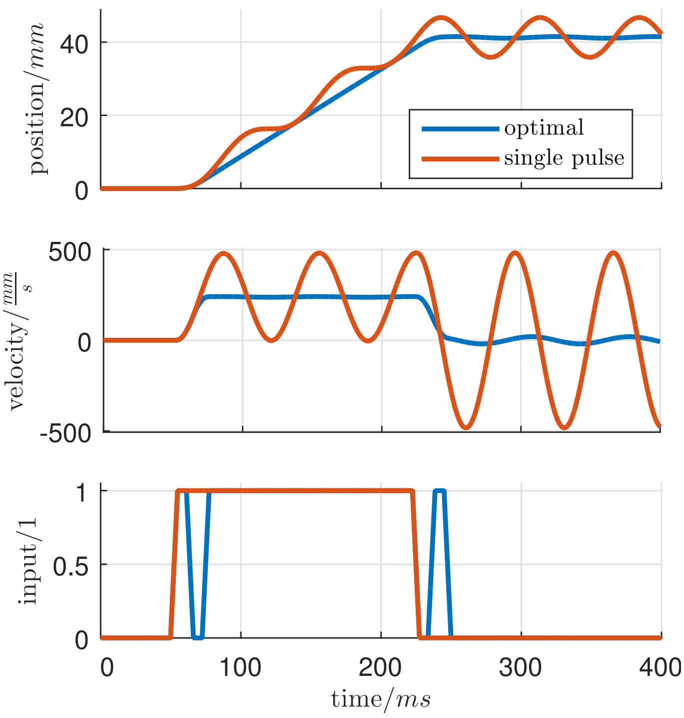


Figure 6: Single pulse vs. optimal control

dynamics of the digital valve, which is assumed to be sufficiently low in the range of a few milliseconds. In Fig. 6 the basic performance of the optimal digital valve control compared to a single pulse valve actuation is depicted for an exemplary ramp movement of a simple cylinder drive. The mechanical load is accelerated due to the first switching pulse with the duration according to the UM-ZV input shaper; then the digital valve is closed and from time Δ the valve is switched on again in order to move the load with minimum oscillations at maximum velocity, which is determined by the flow rate through the valve and the cross-section area of the piston. For deceleration of the drive another characteristic switching pattern, i.e. a brake pulse is applied at a certain distance before the desired position is reached. For demonstration of the concept in this simulation result the viscous damping was completely neglected, therefore the oscillations in the single pulse response do not decay with time. Since real digital hydraulic valves have limited dynamics small oscillations are present in the velocity diagram.

The basic functional scheme of the digital hydraulic concept for the wrapper roll is illustrated in Fig. 7. The servo valve was replaced by two directional switching valves; both are connected to the piston sided chamber and the rod sided chamber is connected to supply pressure constantly. The associated transfer function of the digital concept regarding the piston related to the input flow rate through the digital valves is depicted in Fig. 8a, which shows an improved dynamic frequency behavior in the higher resonances compared to Fig. 3b due to the constant pressure supply in the rod sided chamber. In fact, besides the first natural frequency all other resonances are located below the 0 dB magnitude line, which means that this resonances have reduced influence on the desired behavior even in spite of the broad band excitation of the digital switching of the valves. Furthermore, in contrast to the servo configuration the operating pressure p_A in the piston sided chamber of the cylinder is well defined. Assuming that pressure fluctuations in the cylinder due to acceleration, respectively, deceleration are small the flow through the digital valves is nearly constant. The simulation results of the optimal digital valve control

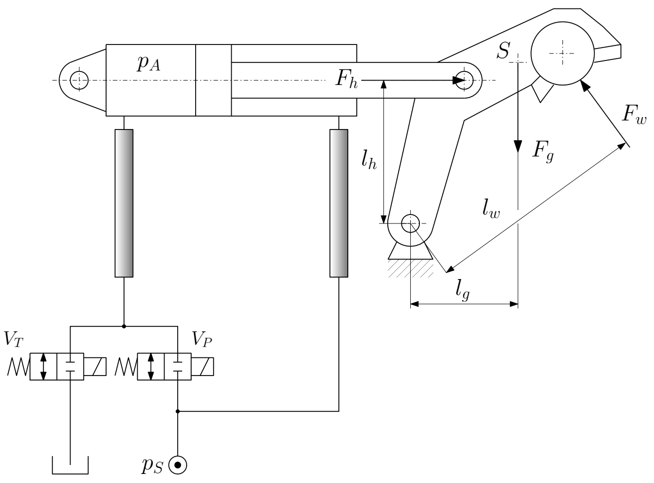
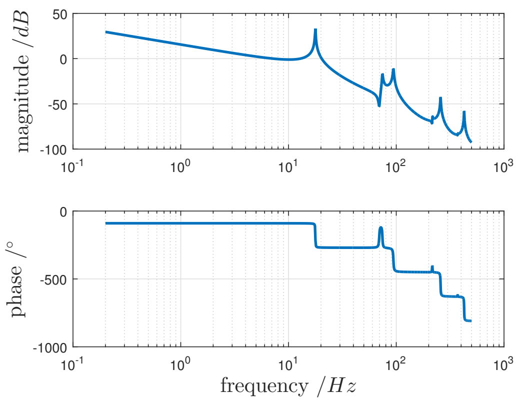
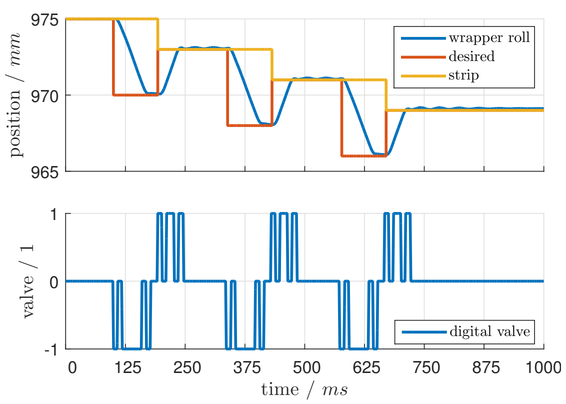


Figure 7: Schematics of a digital hydraulic wrapper roll actuation



(a) Transfer function



(b) Simulation results for $v_s = 10\,{}^{m}\!/\!{}_{s}$

Figure 8: Dynamics and response of the digital wrapper roll actuation

for the wrapper roll at a strip speed of $10\,{}^{m}\!/\!{}_{s}$ is depicted in Fig. 8b. In the lower diagram the opening of the switching valves is shown; a positive sign indicates a switching of the supply sided valve and, respectively, the negative sign is related to the tank sided valve. The UM-ZV shaper is designed to the first natural frequency of $\omega_d \approx 2\pi \cdot 18\,\mathrm{Hz}$ with regard to the frequency respronse in Fig. 8a. Figure 8b shows not only a satisfying tracking of the wrapper roll, moreover the maximum physical dynamic response of the considered drive system is achieved. It must be remarked that for a proper realization of the digital control a sufficiently fast digital hydraulic valve must be used.

4. EXPERIMENTS ON A LABORATORY TEST AXIS

Before the high dynamic digital valve control is applied to a real wrapper roll the concept must be verified by experiments in the laboratory. For this purpose the original problem of the wrapper roll is transferred to a down scaled linear hydraulic drive with similar dynamics. In Fig. 9 the scheme of a linear test axis controlled by digital hydraulic valves can be examined. Similar to the wrapper roll the rod sided chamber is connected to supply pressure constantly. In order to avoid cavitation due to the switching of the tank sided

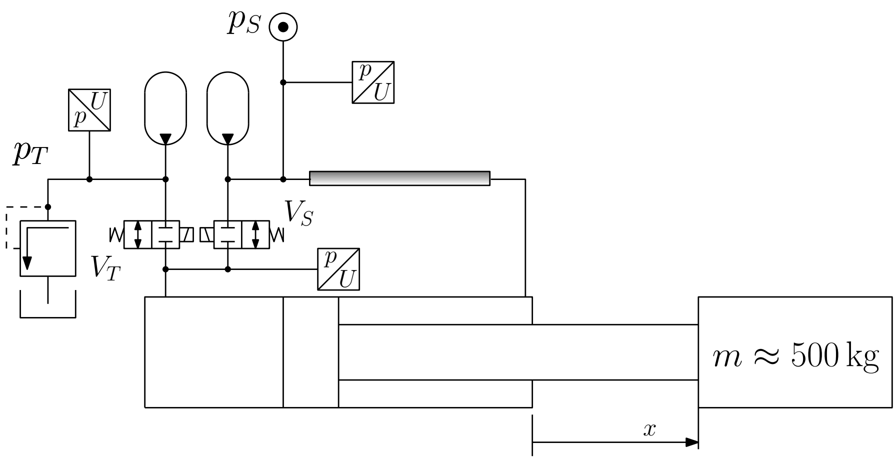


Figure 9: Scheme of the digital hydraulic test bench

parameter	value	parameter	value
piston diameter	$d_P = 53\,\mathrm{mm}$	dead load	$m = 500\,\mathrm{kg}$
rod diameter	$d_R = 36\,\mathrm{mm}$	digital valves	$Q_N \approx 10\,{}^{l}/_{\mathrm{min}}@5\mathrm{bar}$
cylinder stroke	$l_C = 500\,\mathrm{mm}$	supply pressure	$p_S = 160\,\mathrm{bar}$
dead volume	$V_0 = 0.1\,\mathrm{dm}^3$	tank pressure	$p_T = 20\,\mathrm{bar}$

Table 2: Parameters of the test axis

valve the tank is pre-pressurized with a pressure relief valve and a gas loaded accumulator. Furthermore, in order to stabilize the supply pressure and, thus, to avoid unwanted dynamic influence of the supply lines a gas loaded high pressure accumulator is located directly at the supply sided switching valve. With the parameters of the configuration listed in Tab. 2 the natural frequency can be assessed by

$$\omega_d = \sqrt{\frac{E_o A_P^2}{m\,(V_0 + A_P x)}} \approx 2\pi \cdot 18\,\mathrm{Hz}, \tag{1}$$

at a piston position $x = 0.35\,\mathrm{m}$ and with the piston sided cross-section area $A_P = \frac{d_P^2 \pi}{4}$ and the fluid compressibility E_o from Tab. 1. Thus, the characteristic switching time is rather the same as for the original wrapper roll.

In order to confirm the simulation results of the wrapper roll corresponding experiments were carried out. For this purpose a test rig in accordance with Fig. 9 was set up in the laboratory. The resulting test stand is illustrated in Fig. 10. The operating position of the piston was chosen in order to achieve the same natural frequency of the hydro-mechanical spring mass oscillator from the simulation results and, furthermore, close to the natural frequency of a real wrapper roll mechanism. For the experiments digital 2/2 cartridge seat valves from BUCHER Hydraulics were used.

In Fig. 11 measurements of the different digital valve control concepts - the single pulse, respectively, the optimal switching concept - are presented. In Fig. 11a the position tracking for the single pulse control are depicted. In the upper diagram the actual and the desired position of the piston are shown. The constant slope in the desired position represents the velocity of the piston according to the constant flow rate through the digital

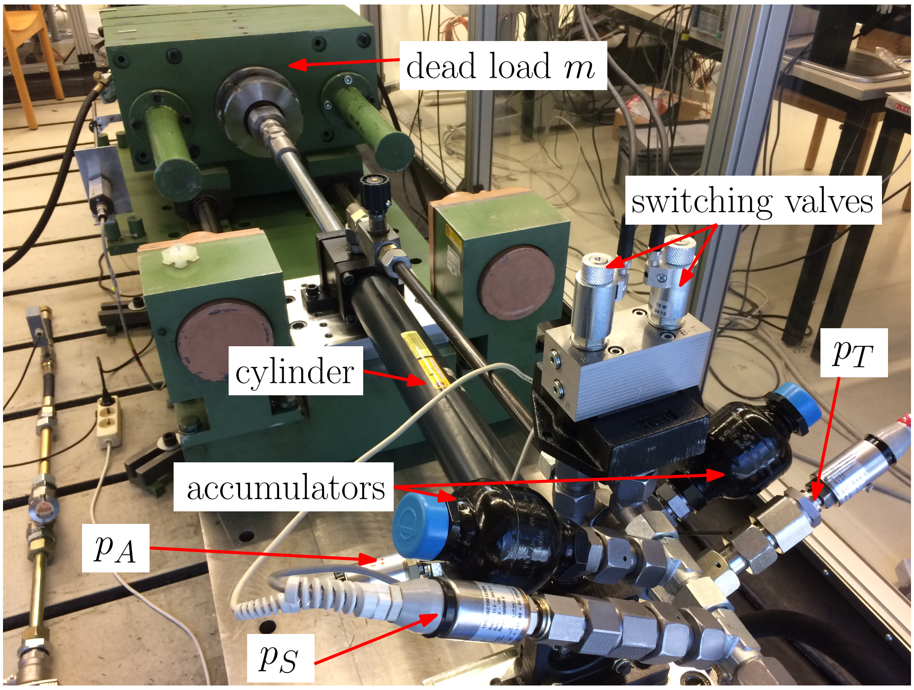
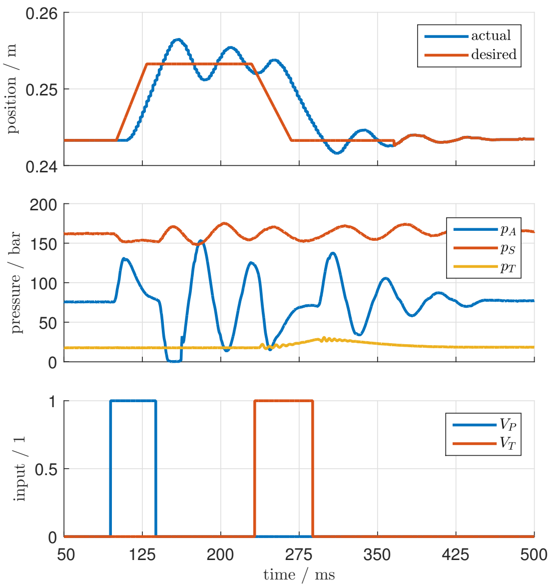
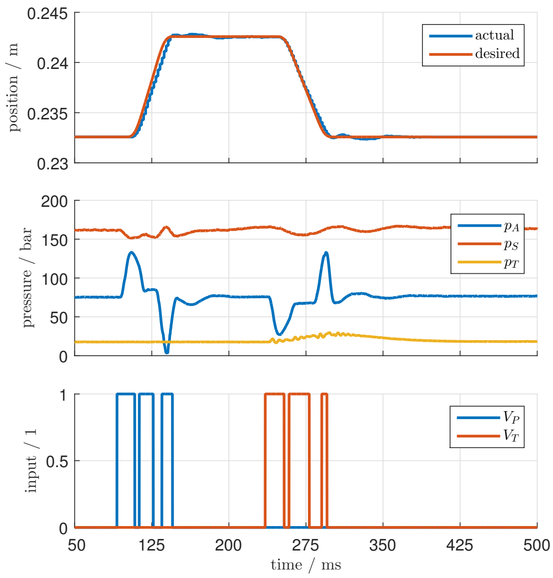


Figure 10: Hydraulic test bench in the laboratory

valve at the constant pressure p_A in a rest position of the drive. In fact the equilibrium pressure p_A is approximately one half of the supply pressure p_S according to the relation of the cross-section areas of piston sided and annulus chamber. But, since the tank is pre-pressurized to approximately 20 bar the pressure difference between the piston sided chamber and tank is lower, which leads to a reduced velocity for the retracting movement compared to the extending velocity of the cylinder. The response, i.e. the actual piston position, shows large unintended oscillations around the desired rest position with the natural frequency of the drive configuration, which were excited by the single pulse valve actuation. Furthermore, focusing on the pressure p_A in the middle diagram even cavitation occurs for an extending movement after the supply sided valve is shut quickly. Such a dynamic response of the drive would not be acceptable for a wrapper roll, since the oscillations would provoke strip collision and cavitation would be harmful for the hydraulic components. In Fig. 11b the drive performance with the optimal digital valve control is presented. In the upper diagram the actual and the desired piston position agree well. Moreover, the cavitation effect in the piston sided chamber could be prevented with the optimal switching. The actual control signals of the digital valves are depicted in the lower diagram. In contrast to the expected symmetric switching according to the UM-ZV an asymmetric switching is necessary for the optimal performance. On the laboratory test rig some parameters, like the cross-section area of the piston in combination with the dead load, result in a large pressure peak for the acceleration of the drive. Due to this pressure peak in the cylinder chamber the flow through the digital valve is not constant during the switching process. Furthermore, hydraulic switching valves do not provide the actual position of the spool, respectively, the poppet. This means that it is not possible to detect whether the valve is fully open or closed. Moreover, effects like the build up of the current in the solenoid or even oil stiction complicate an appropriate valve actuation. For these reasons the switching pulses for acceleration, respectively deceleration must be optimized any way for a specific application in order to achieve the desired behavior of



(a) Single Pulse Control



(b) Optimal high dynamic digital control

Figure 11: Measurements on a linear test axis

the drive system.

5. CONCLUSIONS AND OUTLOOK

In this paper a digital switching concept for the time optimal digital control of a linear hydraulic drive configuration was presented and the successful implementation on a laboratory test rig could be shown. Based on optimal control theory, respectively, on input shaping techniques the control is designed to the first natural frequency of the drive system. The time optimal high dynamic digital valve control shows the maximum physical dynamic response of the considered drive system. However, in hydraulics due to several effects like the unknown switching state of a digital valve or unsteady and even nonlinear characteristics of the flow rate through the digital valves the actual switching of the valves must be adapted on real drive systems. The presented control strategy is a type of feedforward control. Thus, for an industrial application the accuracy, the repeatability and the robustness must be in an acceptable range. Then expensive and sensitive servo hydraulic valves can be replaced by cheaper and robust digital valves. Moreover, with digital valves the efficiency of hydraulic drives can be improved, since digital valves do not have significant leakage. So far, the wrapping force for building up a certain strip tension was not considered in this study. But basically it is also possible to feedforward the necessary flow rate into the cylinder achieving the necessary pressure for wrapping. The final wrapping force is intended to be controlled by a PWM closed loop control strategy.

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