Flatness-Based Control of a Closed Circuit Hydraulic Press*

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Abstract

Hydraulic drive technology is well known for its high force density and, hence, basically qualified for press applications. While systems with constant pressure supply suffer from a bad energy efficiency due to resistance control and, furthermore, an inferior controllability in particular at fast movement operation, the presented hydraulic press concept overcomes this drawbacks by using a synchronized cylinder in a closed circuit displacement control for rapid movement and in press mode. In this paper a flatness-based control of the position of the main press actuator is presented. The derivation of the controller design is followed by simulation experiments and, furthermore, by a discussion of measurements on a two axes prototype with a load capacity of 50 tons.

1 Introduction

Conventional hydraulic presses (see for instance [1, 2]) use a hydraulic load-sensing for the press mode and a controlled falling of the upper press tool for a fast movement downwards. Since in the load-sensing mode and, furthermore, during the fast motion operation the movement is controlled by resistance control a certain loss of energy occurs. The resulting heating of the oil must be reduced by additional coolers, at least in some cases. Another, drawback of such systems is the

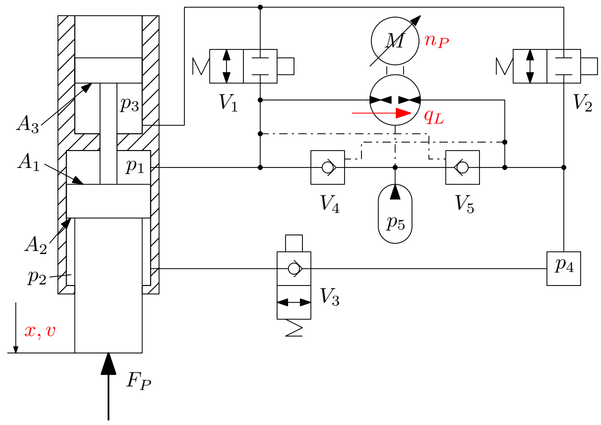


Fig. 1: Functional scheme of a closed circuit hydraulic press

permanent operation of the constant pressure supply, which results in additional energy losses and unnecessary noise (see for instance [3, 4]). The mentioned concept has a single hydraulic supply for one machine, which needs besides pump, motor also transmission lines from a tank and auxiliary components, which inhibit a strict modular and compact design of the machine, where each axis would be completely individual and realized in a compact manner. Furthermore, during the assembling process numerous hydraulic connections must be installed, which constitute potential danger of external leakage, hose breaking and replacement by spare parts which results in additional maintenance work.

In contrast to conventional hydraulic presses with the presented concept according to Fig. 1, the mentioned drawbacks can be prevented. The main actuator is a hydraulic cylinder, which is controlled by a pump, i.e. the rotational speed of the pump corresponds to the velocity of the piston. The cylinder has three hydraulic chambers, where the cross-section areas follow the relation

$$A_1 = A_2 + A_3, \tag{1}$$

which enables a different transmission ratio be-

*This is a preprint of the following chapter: H. Kogler et al., Flatness-Based Control of a Closed-Ciruit Hydraulic Press, published in Dynamics and Control of Advanced Structures and Machines, edited by H. Irschik et al., 2022, Springer Nature, reproduced with permission of Springer Nature Switzerland AG. The final authenticated version is available at: https://doi.org/10.1007/978-3-030-79325-8_10 tween the rotational speed of the pump and the velocity of the piston. In particular, if the valve V_1 is switched, then the effective cross-section area $A_2 = A_1 - A_3$ is active. Thus, a rapid motion at low forces can be achieved. If the valve V_2 is switched, then A_1 is the effective cross-section area and high forces can be produced at low velocities. The valve V_3 represents an emergency valve and during a normal operation of the press this valve is always open. In case of an emergency case the valve V_3 must be shut quickly in order to prevent severe injuries of human operators. The accumulator keeps the whole configuration at a minimum pressure level. In combination with the unlockable check valves V_4 and V_5 the compressibility of the fluid and certain leakage flows can be compensated.

In this paper the application of a flatness-based controller according to the established literature (see for instance [5, 6, 7, 8]) for the position of the piston of the press is presented. In contrast to common PI-controllers, which are only capable to stabilize a rest position, the flatness-based approach stabilizes the trajectory of the piston. Thus, with a properly designed flatness-based controller multiple press axes can be easily realized without an additional synchronizing controller.

2 Modeling

The mechanical part of the press, respectively the movement of the piston, follows the differential equations according to Eq. (2) which is basically represented by the momentum equation. In this model a static friction model considering a stick-slip effect in the motion of the piston.

The hydraulic model is represented by Eq. (3) with the flow rates through the unlockable check valves

$$\begin{aligned} Q_{V_4} &= \begin{cases} K_{V_4} \sqrt[\wedge]{p_5 - p_1} & 0.45 p_4 > p_1 \\ K_{V_4} \mathrm{cv}\left(p_5 - p_1\right) & \text{otherwise} \end{cases} \\ Q_{V_5} &= \begin{cases} K_{V_5} \sqrt[\wedge]{p_5 - p_4} & 0.45 p_1 > p_4 \\ K_{V_5} \mathrm{cv}\left(p_5 - p_4\right) & \text{otherwise} \end{cases} \\ \mathrm{cv}\left(\Delta p\right) &= \begin{cases} \sqrt[\wedge]{\Delta p} & \Delta p > 0 \\ 0 & \text{otherwise.} \end{cases} \end{aligned} \tag{4}$$

The function $\sqrt[\wedge]{\cdot}$ represents the square root characteristics[1] of a hydraulic valve considering a flow in both directions according to

$$\sqrt[\wedge]{\Delta p} = \mathrm{sign}\left(\Delta p\right) \sqrt{|\Delta p|}. \tag{5}$$

The resulting complete model of the press consisting of Equations (2) and (3) is nonlinear and suffers from numerous square root characteristics due to the valve's flow equations and, furthermore, the switching conditions of the check valves and the friction model. Thus, the model is evaluated as too complicated for an efficient controller design, but will be used for simulation experiments.

3 Synthesis

For an efficient controller design a suitable dynamic model for the motion of the piston must be found. Therefore, in a first step the compressibility of the fluid is completely neglected. Then the continuity equation reduces to

$$0 = V_D n_P - A_E v - q_L, \tag{6}$$

with the displacement volume V_D of the pump, the effective cross-section area of the piston A_E and a certain leakage flow rate q_L depending on the pressure difference over the pump. Furthermore, the leakage flow q_L represents a certain correction term for the compressibility of the fluid, at least in a simplified manner. A re-arrangement of Eq. (6) leads to

$$\dot{x} = \underbrace{-\frac{q_L}{A_E}}_{f} + \underbrace{\frac{V_D}{A_E}}_{g} n_P, \tag{7}$$

which represents the basic model used for the following controller design. The model (7) is nonlinear and input-state-linearizable according to literature (see for instance [9]). With the feedback of the input transformation

$$n_P = \frac{v_B - L_f x}{L_g x} \tag{8}$$

the model is linearized and now represented by so called Brunovsky canonical form

$$\dot{x} = -\frac{q_L}{A_E} + \frac{V_D}{A_E} \underbrace{\left(\frac{\left(v_B + \frac{q_L}{A_E}\right) A_E}{V_D} \right)}_{\text{Eq. (8)}} = v_B \tag{9}$$

with the new input v_B. From Eq. (9) it is clear that the actual velocity of the piston corresponds

[1] Around the origin the square root function is approximated by a polynomial.

$$\begin{bmatrix} \dot{x} \\ \dot{v} \end{bmatrix} = \begin{bmatrix} v \\ \frac{1}{m}\left(p_2 A_2 - p_1 A_1 + p_3 A_3 - mg - F_P - \underbrace{\left(\operatorname{sign}(v)\left(\left(F_S - d_c\right) e^{-\left|\frac{v}{v_0}\right|} + d_c\right) + d_v v\right)}_{\text{static friction model}}\right) \end{bmatrix} \tag{2}$$

$$\begin{bmatrix} \dot{p}_1 \\ \dot{p}_2 \\ \dot{p}_3 \\ \dot{p}_4 \\ \dot{p}_5 \end{bmatrix} = \begin{bmatrix} \frac{E_{\text{oil}}}{V_{0_1} - A_1 x}\left(u_{V_1} K_{V_1} \sqrt[\wedge]{p_3 - p_1} - V_S n_P + A_1 v - \frac{q_{\text{leak}}}{p_{\text{leak}}}\left(p_1 - p_4\right) + Q_{V_4}\right) \\ \frac{E_{\text{oil}}}{V_{0_2} + A_2 x}\left(-A_2 v + u_{V_3} K_{V_3} \sqrt[\wedge]{p_4 - p_2}\right) \\ \frac{E_{\text{oil}}}{V_{0_3} + A_3 x}\left(-A_3 v - u_{V_1} K_{V_1} \sqrt[\wedge]{p_3 - p_1} + u_{V_2} K_{V_2} \sqrt[\wedge]{p_4 - p_3}\right) \\ \frac{E_{\text{oil}}}{V_{0_4}}\left(-u_{V_3} K_{V_3} \sqrt[\wedge]{p_4 - p_2} - u_{V_2} K_{V_2} \sqrt[\wedge]{p_4 - p_3} + V_S n_P - \frac{q_{\text{leak}}}{p_{\text{leak}}}\left(p_4 - p_1\right) + Q_{V_5}\right) \\ \frac{\kappa p_5\left(-Q_{V_4} - Q_{V_5}\right)}{V_{0_S}\left(\frac{p_0}{p_5}\right)^{\frac{1}{\kappa}}} \end{bmatrix} \tag{3}$$

exactly to the new input v_B of the system. Thus, defining a control error $e = x - x_d$ with the desired piston position x_d a flatness-based controller is derived by

$$0 = \underbrace{(v_B - v_d)}_{\dot{e}} + \underbrace{(x - x_d)}_{e}\gamma_C, \tag{10}$$

which represents a linear differential equation for the trajectory error e. With the correct choice of the controller parameter γ_C for asymptotic stability Eq. (10) can be solved for v_B and the trajectory error e is forced to decay with time. However, the controller only works properly, if the leakage flow rate q_L is known. Assuming that the dynamics of the process force F_P is low compared to the desired dynamics of the trajectory of the piston, then the leakage flow can be modeled like

$$\dot{q}_L = 0. \tag{11}$$

With the new state q_L the extended system calculates to

$$\begin{bmatrix} \dot{x} \\ \dot{q}_L \end{bmatrix} = \underbrace{\begin{bmatrix} 0 & -\frac{1}{A_E} \\ 0 & 0 \end{bmatrix}}_{\mathbf{A}} \begin{bmatrix} x \\ q_L \end{bmatrix} + \underbrace{\begin{bmatrix} \frac{V_D}{A_E} \\ 0 \end{bmatrix}}_{\mathbf{b}} n_P$$
$$y = \underbrace{\begin{bmatrix} 1 & 0 \end{bmatrix}}_{\mathbf{c}^\mathsf{T}} \begin{bmatrix} x \\ q_L \end{bmatrix}, \tag{12}$$

which is linear and even time invariant. Thus, for the system (12) now a complete observer can be designed. For this purpose the observability must be checked by the calculation of the observability matrix

$$\mathbf{Q} = \begin{bmatrix} \mathbf{c}^\mathsf{T} \\ \mathbf{c}^\mathsf{T}\mathbf{A} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ 0 & -\frac{1}{A_E} \end{bmatrix}, \tag{13}$$

which is regular for every point in time and, thus, the system (12) is observable. In order to simplify the observer design the system is transformed into observer canonical coordinates by using

$$\mathbf{z} = \mathbf{\Theta}^{-1}\mathbf{x} \tag{14}$$

and the transformation matrix [10]

$$\mathbf{\Theta}^{-1} = \left[\mathbf{Q}^{-1}\begin{bmatrix} 0 \\ 1 \end{bmatrix}, \mathbf{A}\mathbf{Q}^{-1}\begin{bmatrix} 0 \\ 1 \end{bmatrix}\right] = \begin{bmatrix} 0 & -\frac{1}{A_E} \\ 1 & 0 \end{bmatrix}. \tag{15}$$

The system (12) in canonical coordinates calculates to

$$\mathbf{A}^\star = \mathbf{\Theta}\mathbf{A}\mathbf{\Theta}^{-1} = \begin{bmatrix} 0 & 0 \\ 1 & 0 \end{bmatrix}$$
$$\mathbf{c}^{\star\mathsf{T}} = \begin{bmatrix} 0 & 1 \end{bmatrix}, \tag{16}$$

where the dynamics of the observer can be easily designed

$$\mathbf{A}_\mathcal{O} = \mathbf{A}^\star + \begin{bmatrix} -\alpha_0 \\ -\alpha_1 \end{bmatrix}\mathbf{c}^{\star\mathsf{T}} = \begin{bmatrix} 0 & -\alpha_0 \\ 1 & -\alpha_1 \end{bmatrix}. \tag{17}$$

After a transformation back into original coordinates the complete observer reads

$$\begin{bmatrix} \dot{\hat{x}} \\ \dot{\hat{q}}_L \end{bmatrix} = \begin{bmatrix} -\frac{\hat{q}_L}{A_E} + \frac{V_D}{A_E} n_P + \alpha_1\left(x - \hat{x}\right) \\ A_E \alpha_0 \left(x - \hat{x}\right) \end{bmatrix}. \tag{18}$$

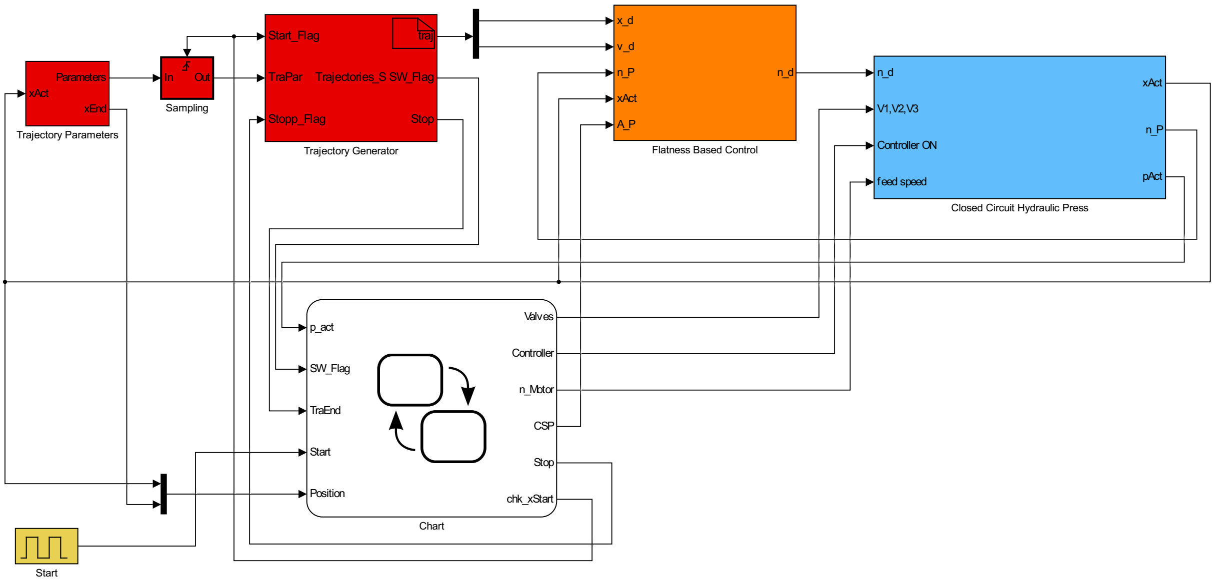


Fig. 2: Simulation model

4 Simulations

The controller and the observer derivated in the previous section were tested by simulation in *Matlab/Simulink*. The corresponding block diagram is depicted in Fig. 2. The blue colored block on the right hand side contains the dynamic model of the press according to Equations (2) and (3). In the orange block left to the press model the flatness-based controller and the load observer are located. The block has 5 inputs, as two for the desired position and velocity of the piston, the actual rotational speed of the pump, the actual piston position and, finally, the effective cross-section area of the piston according to the switching states of the valves V_1 and V_2 in Fig. 1. The decisions for the valve switching and, thus, the selection of fast or press mode are calculated in the state machine depicted at the bottom of the block diagram in Fig. 2. The remaining red blocks in the upper left corner of the simulation diagram represent the parameterization of the piston trajectory, which must be at least 2 times differentiable with respect to time according to the requirements for the flatness-based control.

The simulation experiments were carried out for a spring load according to Fig. 3, where the load was determined by the compressive force due to the displacement $x - x_{\text{contact}}$ of a linear spring.

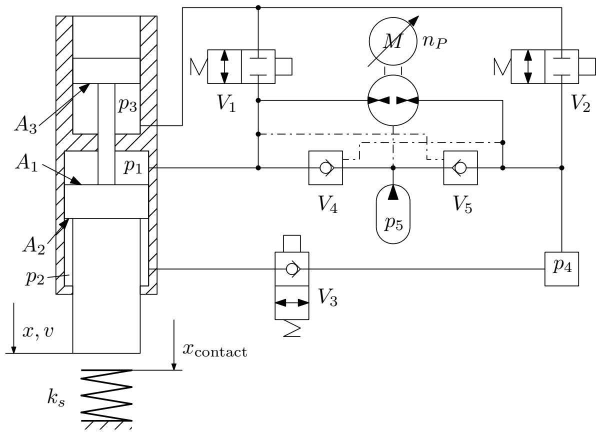
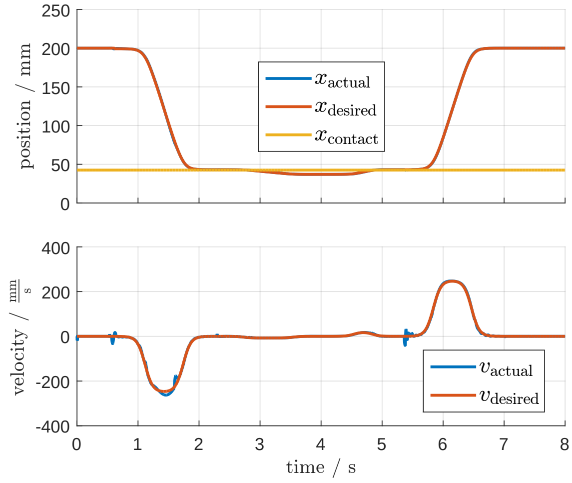
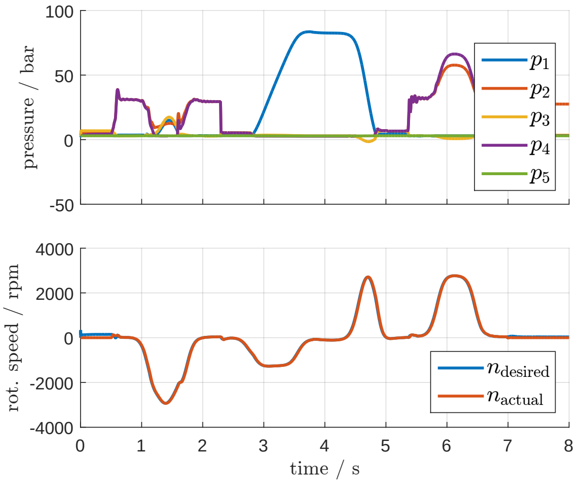


Fig. 3: Load model

In Fig. 4 the simulation results for a full exemplary press cycle are depicted. In Fig. 4a the position and the velocity of the piston are illustrated. The press cycle starts at an initial position of 200 mm with a rapid motion downwards close to the position x_{contact} with a small effective cross-section area A_2. During this fast motion downwards some fluctuation in the velocity are present, which result from the switching of the unlockable check valves V_4 and V_5 according to the internal pressure states presented in the upper diagram of Fig. 4b. Then the configuration is switched to the large effective cross-section area A_1 for the press mode and the motion against the spring load is executed, which results in a pressure build up of p_1 in the main cylinder chamber. A certain holding phase in the desired target press position is followed by a de-

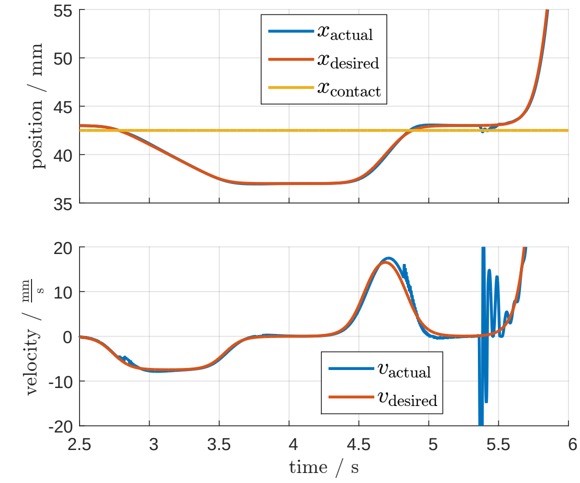


(a) Position and velocity

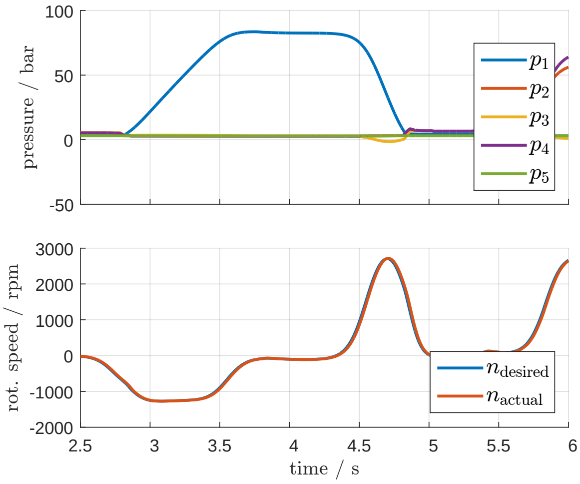


(b) Pressure and rotational speed

Fig. 4: Simulation of an exemplary press cycle



(a) Position and velocity



(b) Pressure and rotational speed

Fig. 5: Zoom into Press mode

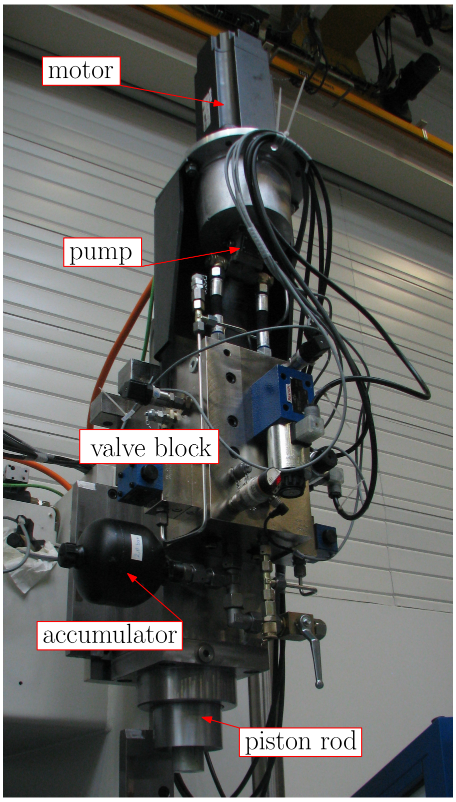
compression phase until the piston is losing contact with the load spring. Then the cross-section area is switched to fast mode again and a rapid motion to the initial position is performed. In beginning of this phase again fluctuations in the velocity can be observed, which result from switching to the small effective cross-section area A_2 and, furthermore, again from the switching of unlock able check valves V_4 and V_5.

Figure 5 presents a close-up of the press mode, which shows a satisfying control performance of the piston trajectory. Only at the decompression at higher velocity compared to the press motion and at switching to rapid motion configuration some fluctuations in the trajectory occur. However, the controller satisfyingly copes with such disturbances resulting in a nice decay of the trajectory error.

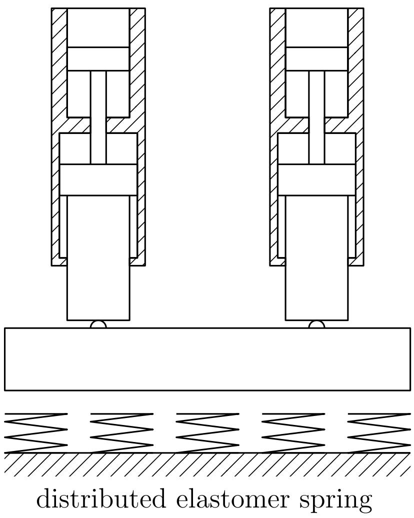
5 Measurements

In Fig. 6a a picture of a single axis prototype with a force capacity of 25 tons is presented and the major parts with regard to Fig. 1 are indicated. In Fig. 6b a parallel arrangement of two single axes is depicted, which was considered for the measurements. By the use of a beam the two individual press axes act simultaneously against the load, which is constituted by a distributed elastomer spring.

In Fig. 7 the measurements of an exemplary press cycle with the 50 tons two axes press are illustrated. In the diagrams of Fig. 7a the position and the velocity of the piston are depicted. Like in the

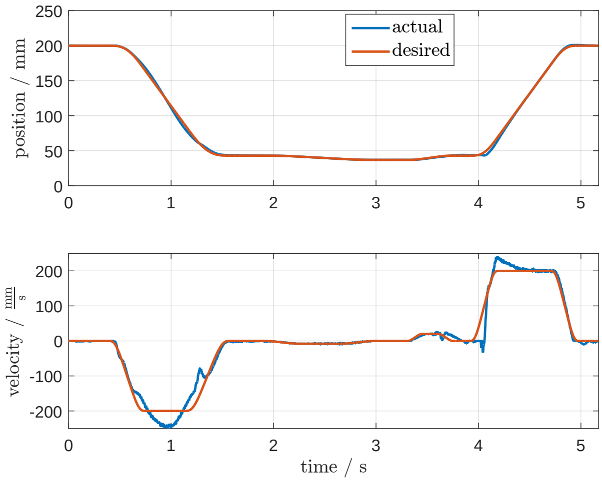


(a) Prototype with 25 tons capacity

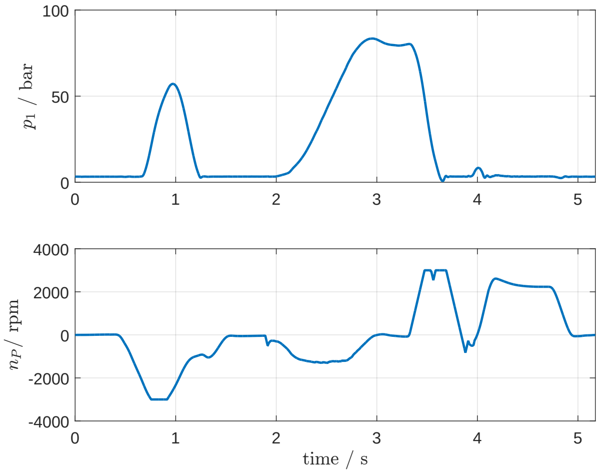


(b) double axes press with 50 tons capacity

Fig. 6: Testrig for measurements



(a) Position and velocity

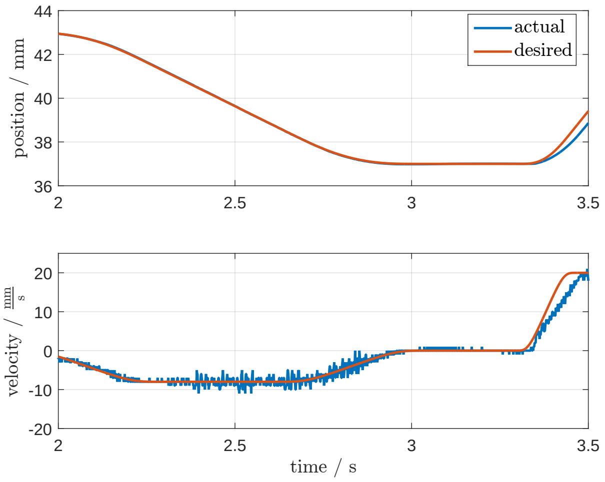


(b) Pressure and rotational speed

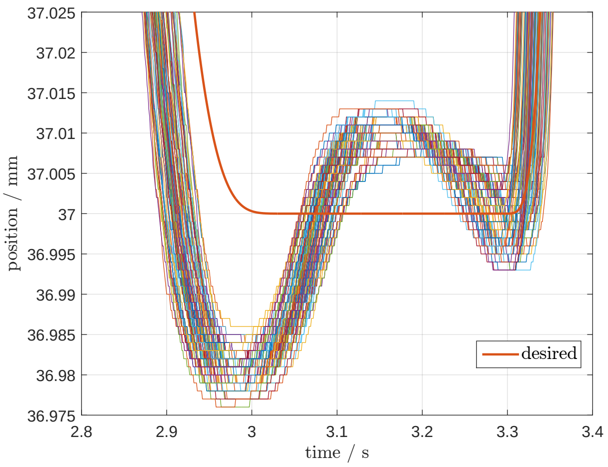
Fig. 7: Measurements of a full exemplary press cycle

simulation results during the fast motion downwards fluctuations occur in the measurements as well. Here, additionally to the effect of the switching of the unlockable check valves a saturation effect in the rotational speed of the motor comes into play, which can be seen in the lower diagram of Fig. 7b.

In Fig. 8a a zoom into the press mode phase is illustrated. The fluctuations in the actual velocity result from the numeric differentiation of the position signal. In Fig. 8b a closer view to the target position of 37 mm shows 60 consecutive periods of one press cycle. In those measurements an accuracy of approximately 40 μm at a repeatability in the range of 10 μm can be achieved. It must be remarked that the results could be achieved without



(a) Position and velocity



(b) Accuracy and Repeatability

Fig. 8: Zoom into Press mode

any parameter identification. Due to the simple structure of the dynamic model for the synthesis, only a few parameters from data sheets and the mechanical design were necessary. Furthermore, the measurements were carried out with a two axes prototype. Both axes respond a little bit different due to their individual friction behavior and, thus, each axis represents a certain disturbance for the other press axis. Consequently, with a single axis press even a better accuracy can be achieved.

6 Conclusion and Outlook

In this paper the design and application of a flatness-based controller for a double axes closed circuit hydraulic press prototype was presented. The press concept is realized as displacement control, where the rotational speed of the pump corresponds to the velocity of the piston, thus, no resistance control is applied and no proportional valves are used. The transmission ratio between force and velocity of the piston with regard to the rotational speed of the pump is controlled by on/off switching valves. According to the displacement control of the piston an excellent energy efficiency can be achieved; no additional cooler is needed. The closed circuit system only needs a small amount of oil, thus, a compact design can be realized in order to completely hide the hydraulics from the operator; only electricity must be provided to the black box system. For the controller design a simple dynamic model could be found, which was used for the design of a flatness-based controller. This control strategy achieves a stabilization of the trajectory, which means that multiple individual press axes can be arranged and operated in parallel without any additional synchronizing controller. With the presented controller satisfying control performance could be achieved on a two axes press prototype. Next steps in development are focused on a more compact integration of the motor-pump section.

Acknowledgment

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