Adaptive observer-based switched control for electropneumatic clutch actuator with position sensor

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Abstract: In this paper we extend previous control design applied to an electropneumatic clutch actuator. A dual-mode switched nonlinear controller design and an adaptive nonlinear observer design are combined to achieve position control of the clutch actuator with only position measurements available. The observer provides estimates of position, velocity, chamber pressures, friction state and parameters, and clutch load characteristic parameters. The paper focus on design for implementation in a heavy duty truck system, and focus on performance evalution using simulations. Performance is tested with a simulation model which is validated against experimental results from a heavy duty test truck, showing no significant loss of performance due to lack of a pressure sensor.

1. INTRODUCTION

The considered electropneumatic clutch actuator is intended for automation of the manual transmission in Automated Manual Transmissions (AMT) or Clutch-By-Wire (CBW) applications in heavy duty trucks. Pressurized air is already present in trucks, and due to economical reasons, pneumatics are also chosen for the clutch system. Pneumatic actuators are used in many industrial applications, because of cleanliness, low cost, high ratio of power to weight and easy maintenance, Ali et al. [2009], but they also hold highly nonlinear characteristics due to compressibility of air, stiction and high friction forces which affects control accuracy.

Cost, combined with space advantages and better robustness properties, have lead to that on/off solenoid valves are chosen as control valves over proportional valves with are the other common options. From a control point of view, on/off solenoid valves are more challenging because of the valve's discrete on/off nature, limitations in response time and a dynamical response which is hard to model accurately. Pulse Width Modulation (PWM) is usually applied when on/off solenoid valves are used for position control of pneumatic actuators, see Ahn and Yokota [2005] and references therein. Sliding mode techniques have become a common approach for designing controllers, especially for pneumatic actuators with on/off solenoid valves since the valves discrete nature can be exploited. Nguyen et al. [2007] uses a sliding mode approach to construct a control signal which can be directly applied to the solenoid valves, and Paul et al. [1994] do the same for position control of a pneumatic cylinder. Switched control design for the electropneumatic clutch actuator without using PWM, exploiting the solenoid valves discrete nature, is considered in earlier work Sande et al. [2007], Langjord et al. [2008, 2009], Langjord and Johansen [2010].

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Use of pressure feedback is essential for precise and robust control performance for pneumatic actuators, Pandian et al. [2002]. State and parameter estimations are desired as a position sensor is the only sensor present in the production clutch actuator system. Bigras and Khayati [2002] presents a nonlinear observer for pressure estimation in a pneumatic cylinder, ensuring exponential stability of the estimation error. Gulati and Barth [2009] propose an energy-based pressure observer which is combined with a sliding mode controller for control of a pneumatic servo actuation system controlled by a 4-way proportional valve. Szabo et al. [2010] recently presented a feedback linearization based observer for an electropneumatic clutch system. The theses by Kaasa [2006] and Vallevik [2006] consider observer designs for the same clutch actuator system as us, but with a three-way proportional valve as control valve. The thesis by Kaasa [2006] also consider nonlinear output feedback control utilizing the observer design. Adaptive nonlinear observer design for the electropneumatic clutch actuated by on/off solenoid valve have been treated in earlier work Langjord et al. [2010, 2011].

In this paper we combined an adaptive nonlinear observer, Langjord et al. [2011], and dual-mode switched controller designs, Langjord and Johansen [2010], with focus on practical aspects for implementation in a heavy duty truck. In Langjord and Johansen [2010] the nonlinear control design uses both position and pressure sensors, while we aim to avoid the pressure sensor and introduce adaptation of the clutch load characteristic and friction model which is known to be essential, Langjord et al. [2009]. The on/off solenoid valveset considered in this paper is faster than the one considered in Langjord et al. [2011], and the pulse width modulation in the observer model can be omitted as the switched controller provides the control signals.

2. SYSTEM AND MODELS

In Figure 1, a schematic of the considered clutch actuator system is shown. The electronic control unit (ECU) calculates and send control signals to the supply and exhaust valves. These valves control the resulting flow into/out of the actuator chamber. The position of the piston is a result of the friction, the pressure, the spring forces, and determines the state of the clutch plates. The piston acts on the clutch through the piston rod. At the initial piston position the clutch plates are engaged and as the piston moves to the right (air added to the actuator) the clutch plates gets pulled apart, first they will be slipping and then fully disengaged. Due to wear, the characteristics of the clutch compression spring changes, and the clutch load force which is the lumped force of this spring and the counteracting, much weaker, actuator spring changes. The force increases with wear, especially for lower piston positions. We present 3 models of the system, with different level of complexity fit for its application. Variables are described in Table 1.

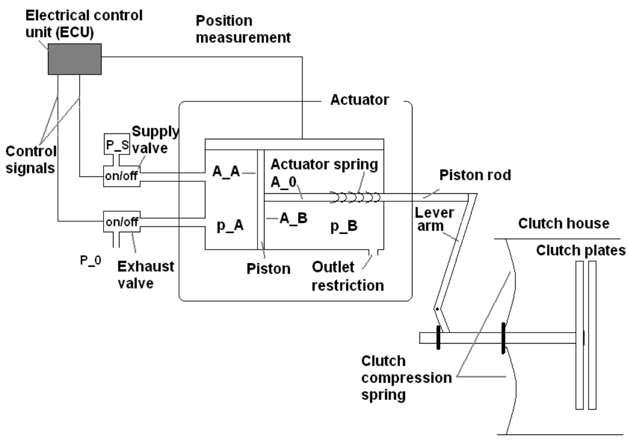


Fig. 1. A schematic of the system

2.1 Simulation model

We present a 5th order model as simulation model

$$\dot{y} = v \tag{1a}$$

$$\begin{aligned} M\dot{v} = &-\phi^T(y)\theta - Dv - Fz \\ &+A_A p_A - A_B p_B - A_0 P_0 \end{aligned} \tag{1b}$$

$$\frac{F}{K_z}\dot{z} = v - |v|z \tag{1c}$$

$$\dot{p_A} = -\frac{A_A v}{V_A(y)}p_A + \frac{RT_0}{V_A(y)}w_v(p_A, u) \tag{1d}$$

$$\dot{p_B} = \frac{A_B v}{V_B(y)}p_B + \frac{RT_0}{V_B(y)}w_r(p_B) \tag{1e}$$

where details of the modeling can be found in Langjord et al. [2011]. The input signal is delayed by 2 ms and 3.5 ms which, respectively, correspond to the average response time of the on/off supply and exhaust solenoid valves in the truck set up. The simulation model representing the actual system, is validated against experimental data from the test truck, see Figure 4, and is shown to capture the main behavior of the clutch actuator. White noise with variance $1.7*10^{-12}$ m^2 and 10 Pa, both sample time 1 ms, is added to the position and the pressure output of the simulation model, respectively, corresponding to the sensor noise present in measurements obtained in the truck.

Table 1. Parameters of the clutch actuator model

y, x_1	Piston position
v, x_2	Piston velocity
x_3	Accumulated air
z	Pre-sliding deflection
p_A, p_B	Pressure in chamber A,B
m_A, m_B	Mass of air in chamber A,B
u	Normalized control input to valve set
w_c	Normalized control input to valve set
A_A, A_B	Area of chamber A,B
A_0	Area of piston rod
P_0	Ambient pressure
P_S	Supply pressure
D	Viscous damping coefficient
K_z	Deflection stiffness coefficient
$F_{\dot{z}}$	Deflection damping coefficient
F	Coulomb friction
T_0	Ambient temperature
R	Gas constant of air
M	Mass of piston
$V_{A,0}, V_{B,0}$	Volume of ch. A,B at $y = 0$
θ	Gain of clutch load ch. basis functions

2.2 Observer design model

For observer design it is convenient to express the system dynamics with the masses of air, m_A and m_B, as states rather than the pressures p_A and p_B,

$$m_A(p_A, y) = \frac{V_A(y)}{RT_0}p_A, \tag{2}$$

$$m_B(p_B, y) = \frac{V_B(y)}{RT_0}p_B, \tag{3}$$

where

$$V_A(y) = V_{A,0} + A_A y \tag{4}$$

$$V_B(y) = V_{B,0} - A_B y. \tag{5}$$

We get the following 5th order observer model

$$\dot{y} = v \tag{6a}$$

$$\begin{aligned} M\dot{v} = &-\phi^T(y)\theta - Dv - Fz + \frac{A_A RT_0}{V_A(y)}m_A \\ &-\frac{A_B RT_0}{V_B(y)}m_B - A_0 P_0 \end{aligned} \tag{6b}$$

$$\frac{F}{K_z}\dot{z} = v - |v|z \tag{6c}$$

$$\dot{m}_A = w_v(p_A, u) \tag{6d}$$

$$\dot{m}_B = w_r(p_B) \tag{6e}$$

as proposed in Langjord et al. [2011] and we refer to this paper for more details. The same input delays as in 2.1 are included to represent valve response times.

2.3 Control design model

It is shown in Langjord and Johansen [2010] that valve dynamics, friction dynamics, and pressure dynamics in chamber B can be neglected, and still sufficiently good control performance is achieved. Hence we use the simpler 3rd order model of the clutch actuator for control design.

$$\dot{x}_1 = x_2 \tag{7a}$$

$$M\dot{x}_2 = \phi^T(x_1)\theta - Dx_2 + \frac{A_A x_3}{V_A(x_1)} - A_A P_0 \tag{7b}$$

$$\dot{x}_3 = RT_0 w_c \tag{7c}$$

where $x_3 = p_A V_A(x_1)$ is accumulated air in chamber A. The model is presented in Langjord and Johansen [2010] and we refer to this paper for more details.

3. ADAPTIVE NONLINEAR OBSERVER

A full-order adaptive nonlinear observer for the clutch actuator system were proposed in Langjord et al. [2011] based on a Lyapunov-design

$$\dot{\hat{y}} = \hat{v} + l_y(y - \hat{y}) \tag{8a}$$

$$M\dot{\hat{v}} = -\phi^T(\hat{y})\hat{\theta} - \hat{D}\hat{v} - F\hat{z} + \frac{A_A RT_0}{V_A(y)}\hat{m}_A - \frac{A_B RT_0}{V_B(y)}\hat{m}_B - A_0 P_0 + l_v(\dot{y} - \hat{v}) \tag{8b}$$

$$\frac{F}{K_z}\dot{\hat{z}} = \hat{v} - |\dot{y}|\hat{z} \tag{8c}$$

$$\dot{\hat{m}}_A = w_v(\hat{p}_A, u) + \frac{l_m}{V_A(y)}(\dot{y} - \hat{v}) \tag{8d}$$

$$\dot{\hat{m}}_B = w_r(\hat{p}_B), \tag{8e}$$

where $l_y, l_v, l_m \geq 0$ are observer injection gains and the adaptation laws are given by

$$\dot{\hat{D}} = -\gamma_D \hat{v}(\dot{y} - \hat{v}) \tag{9}$$

$$\dot{\hat{\theta}} = -\Gamma\phi(y)(\dot{y} - \hat{v}), \tag{10}$$

with $\Gamma = \Gamma^T > 0, \gamma_D > 0$. Let $\tilde{v} = v - \hat{v}$, $\tilde{y} = y - \hat{y}$, $\tilde{z} = z - \hat{z}$, $\tilde{m_A} = m_A - \hat{m_A}$ and $\tilde{m_B} = m_b - \hat{m_B}$.

From Langjord et al. [2011] we know that the observer presented in (8) with the adaptation laws (10) and (9), where $l_v, l_m \geq 0$, $l_y \geq \frac{(\alpha + K\theta_{max})^2}{2\alpha l_v}$, $\alpha > 0$ and $\Gamma = \Gamma^T > 0, \gamma_D > 0$, ensures that for any physically meaningful initial conditions and system trajectories

(1) the error dynamics are stable and all estimates are bounded
(2) $\tilde{v}$, $\tilde{m}_B$ and $\tilde{y}$ converges to zero
(3) if v and u are persistently exciting (PE) then also $\tilde{m}_A$ and $\tilde{z}$ converges to zero

4. DUAL-MODE SWITCHED CONTROLLER

A dual-mode switched controller for the clutch actuator system was proposed in Langjord and Johansen [2010] based on a backstepping control-Lyapunov function design where ultimately the discrete control input w_c is chosen to make the time-derivative of the Lyapunov function as small as possible. The controller is

$$w_c = \begin{cases} w_2, \ \xi \in \Omega_2 \\ w_1, \ \xi \in \bar{\Omega}_1 \end{cases} \tag{11}$$

where $\Omega_2 = O \backslash \bar{\Omega}_1$, $\bar{\Omega}_1$ is the stability region of w_1, $O = \{x | x_1 \in [0, 0.0025], x_2 \in \mathbb{R}, x_3 \in \langle 0, \infty]\}$ are the theoretical region of operation for the clutch, and the state error are written as $\tilde{x}_i = x_i - x_i^*$ where x_i^* are reference points, $x^* = [x_1^*, 0, x_3^*]$. The switched controllers are

$$w_1 = \begin{cases} -U_{\max} sgn(\xi_3) & \text{if } \xi_3 \neq 0 \\ 0 & \text{if } \xi_3 = 0 \end{cases} \tag{12}$$

$$w_2 = \begin{cases} -U_{\max} sgn(s(\tilde{x}_1, \tilde{x}_3)) & \text{if } s(\tilde{x}_1, \tilde{x}_3) \neq 0 \\ 0 & \text{if } s(\tilde{x}_1, \tilde{x}_3) = 0 \end{cases} \tag{13}$$

where $s(\tilde{x}_1, \tilde{x}_3) = \lambda_2 \tilde{x}_3 - \frac{\alpha_2}{M} ln\left(\frac{V_A(x_1)}{V_A(x_1^*)}\right)$ and ξ is found from the coordinate transformation

$$\xi_1 = x_1 - x_1^* \tag{14a}$$

$$\xi_2 = x_2 + k\xi_1 \tag{14b}$$

$$\xi_3 = x_3 - \frac{V(x_1)}{A}(\phi^T(x_1)\theta - Dk\xi_1 + AP_0 - Mk\xi_2)$$

where k is a backstepping gain.

From Langjord and Johansen [2010] we have that the equilibrium point x^* of the system (7b) with the combined controller given in (11) is asymptotically stable in the largest invariant region in O.

The controller is implemented with feedback from the estimated states $\hat{v}$ and $\hat{p_A}$, and the parameter estimates $\hat{\theta}$ and $\hat{D}$. The stability result holds if the true states and parameters are used for feedback, while we are actually violating this assumption by using estimated states and parameters.

5. SIMULATION RESULTS

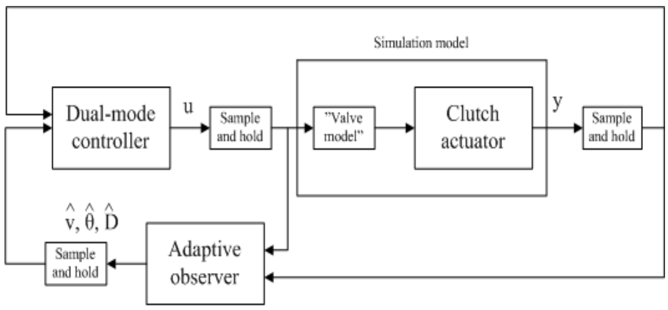


Fig. 2. Connection of controller and adaptive observer

5.1 *Considerations for simulation*

The combined system is tested by simulations in Simulink. The simulated system, the adaptive nonlinear observer and controller are implemented using explicit Euler discretization, where the Euler integration step is set to 0.1 ms. The sampling rate of the position measurement in the actual truck and simulation is 1 ms, and the control signals to the on/off solenoid valves are set at the same rate. The position from the simulation model and the calculated input signal is therefore generated by a sample-and-hold.

The performance of the combined system is tested against a clutch reference shown in Figure 3. In the engage/disengage area the reference position should be reached within 0.1 s and with a steady state error of less

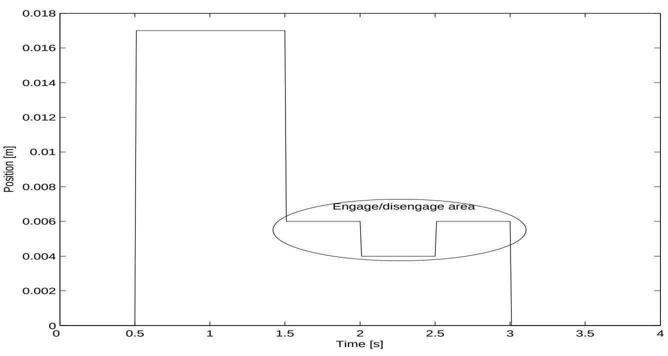


Fig. 3. Clutch sequence

than 0.2 mm. Outside this area these requriments can be somewhat relaxed. The controller is switched off whenever the steady state requirement is fulfilled.

The observer and adaptation gains have been set based on the tuning in Langjord et al. [2011], $l_y = 10$, $l_v = 2000$, $\Gamma = I * 0.1$ and $\gamma_D = 1 * 10^6$. The gain Γ is set 30 times higher in the region $3 - 6$ mm as the clutch load characteristics is especially important in this area due to its steep curve, and since this region is visited only for a short transient periods with a typical clutch sequence. The adaptation is switched off whenever the position is not changing, due to lack of PE which might lead to drift or divergence of the estimates.

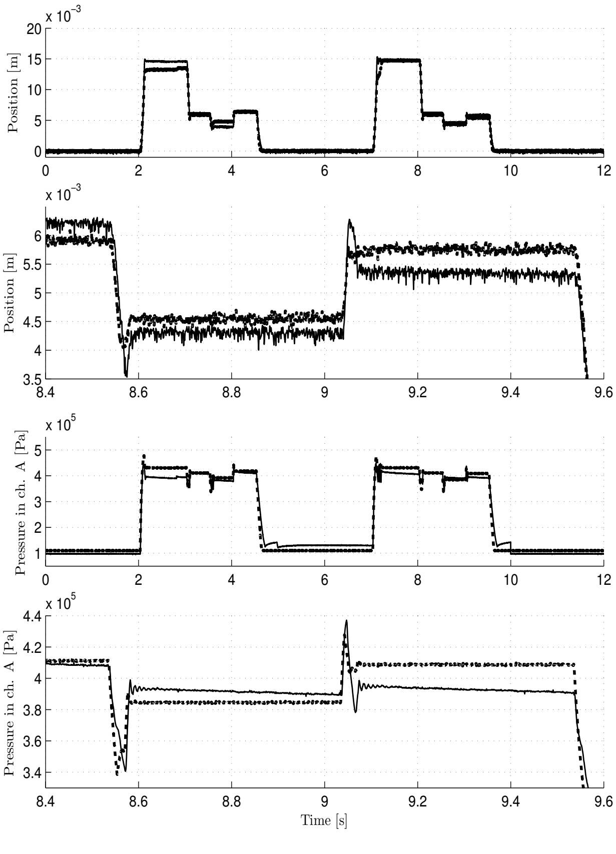
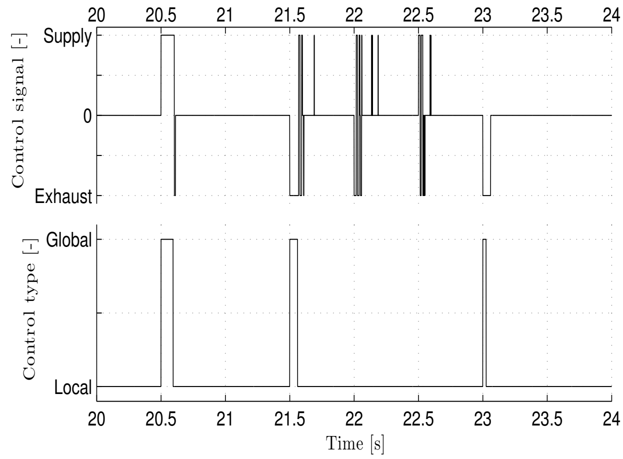


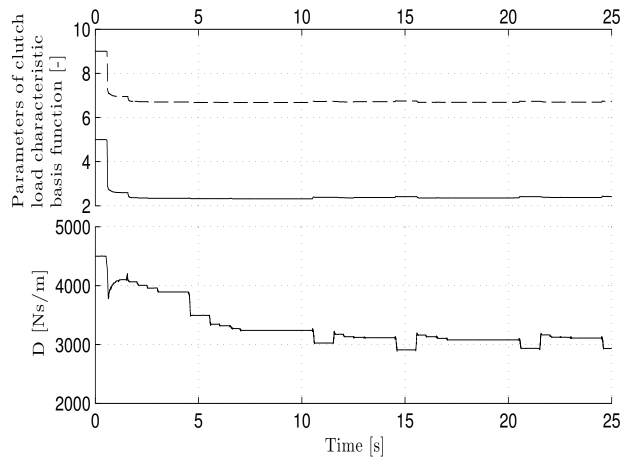
Fig. 4. Experimental measurements from truck (solid) and results from the simulation model (dashed)

5.2 Results

Simulations with nominal values, $\theta_0 = [3, 6]$ and $D_0 = 2500$, show good results of controller, and the requirements are met. In Figure 5 results with $\theta_0 = [5, 9]$ and $D_0 = 4500$ are shown. It is clear that the controller combined with the adaptive observer still manages to meet the requirements. Figure 6(b) show that the estimated parameters converge, but not to the nominal values. The total force of the clutch load and the friction, on the other hand, converges to the same value for the two simulations. This is in agreement with the PE analysis in Langjord et al. [2011] which prove that the parameter error converges, but not necessarily to zero. One of the benefits using a switched controller, is that this type of controller may not be as sensitive to of noise as a continuous controller.



(a) Controller signal and used controller.



(b) Estimated parameters

Fig. 6. Simulations of dual-mode controller and full-order adaptive observer with nominal values $\theta_0 = [5, 9]$ and $D_0 = 4500$.

Simulations with the dual-mode switched controller and

- position from simulation model including noise
- velocity filtrated from position
- pressure from simulation model including noise

have been conducted for comparison. The performance is similar to the performance with the observer-based

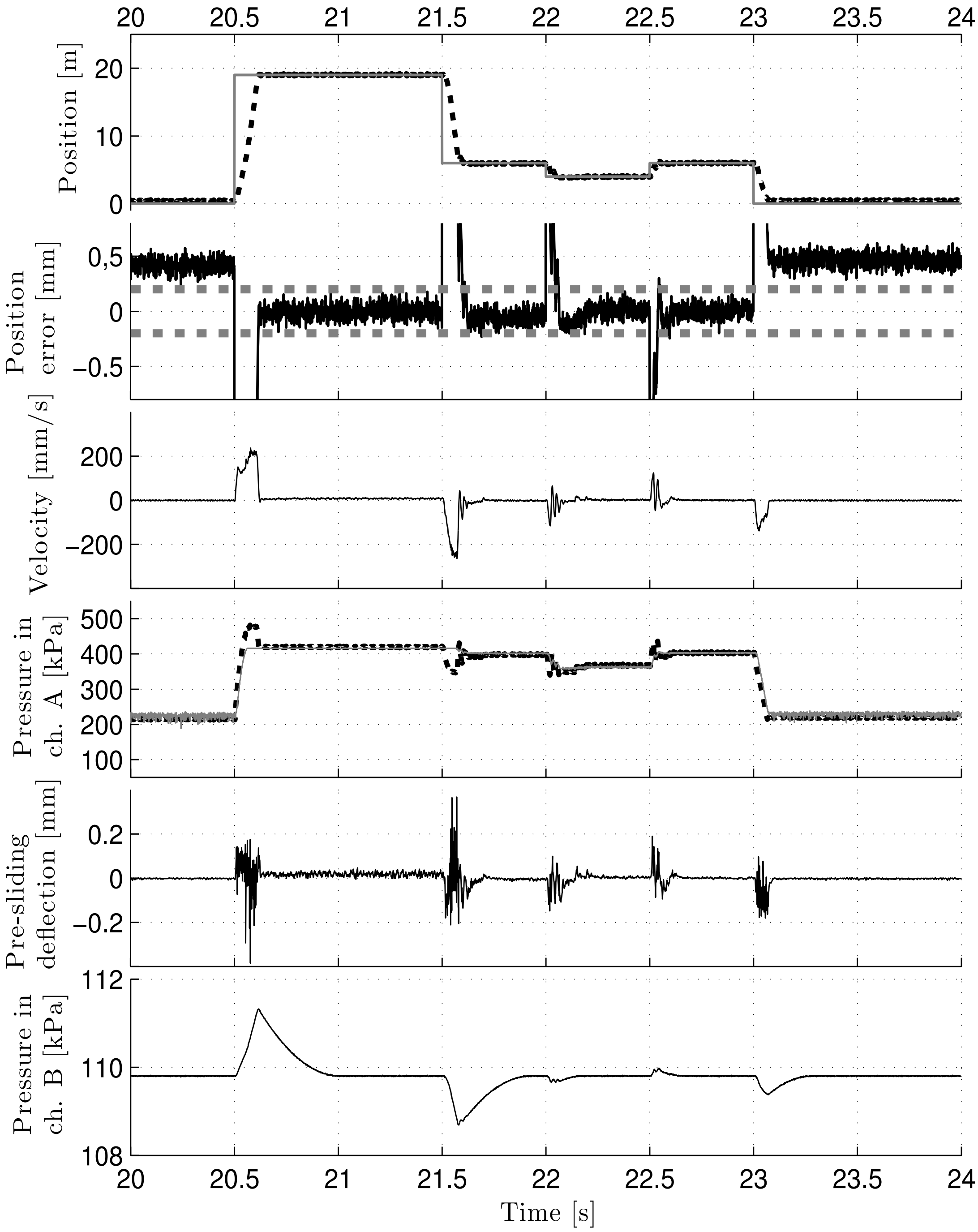


Fig. 5. Simulations of dual-mode controller and full-order observer with $\theta_0 = [5, 9]$ and $D_0 = 4500$. For position, the simulated state (dashed) and the reference (solid) are shown, and for pressure in ch. A, the estimated reference (solid) and the observer state (dashed) are shown.

controller, see Figure 7. Given that the simulation model captures the real system well enough we can state that exchanging the pressure sensor with an observer does not result in any loss of performance.

Simulations with the same state feedback sensors characteristics, filtering, and controller configuration, as the experimental test in Langjord et al. [2010], have also been conducted. Comparing these results we get that they are quite similar, with a bit less chattering in the simulated results. This was expected as additional noise can be present in the truck due to motor vibration etc. The similarity in performance confirms that the simulation model captures the main behavior of the actual system.

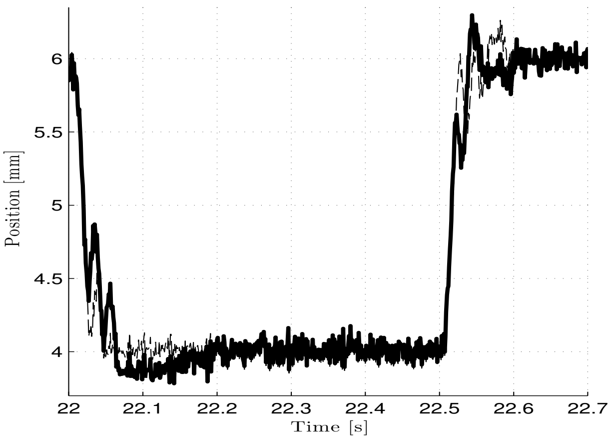


Fig. 7. Segment of the resulting position for simulations of the controller with pressure feedback from measurement (dashed) and from observer (solid).

6. CONCLUSIONS

This paper present position control for an electropneumatic clutch actuator, using a dual-mode switched controller design and utilizing a full-order adaptive nonlinear observer. Tests using the validated simulation model show that the combined design makes the piston position follow the position reference with sufficient precision, even when starting with large offsets in friction and clutch load parameters in the observer model. The results show that the pressure sensor can be exchanged by an observer without significant loss of performance. Theoretical stability analysis of adaptive observer-based feedback is hard to make, since a nonlinear seperation principle is known only for special classes of nonlinear observer-based control systems. Still, no special considerations were needed when combinding the given controller and the given adaptive observer.

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