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Estimation of electropneumatic clutch actuator load characteristics

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Abstract:
This paper propose a dynamic model of an electropneumatic clutch actuator system for heavy duty trucks. The focus is set on modeling of the clutch load characteristic which should be parameter affine, and the main purpose of the modeling task in this paper is to prepare for on-line adaption of this load characteristics. The knowledge of the clutch load characteristic is important for being able to estimate the pressure in the system online, when this is not measurable. In this paper off-line estimation is used to find parameters such that the model is a good representation of the system. The resulting 5^{th} order model is verified by comparison to actual truck measurements.

1. INTRODUCTION

Automated clutch actuation makes it easier for the driver, particularly in stop and go- traffic, and have especially seen a recent growth in the European automotive industry. This have raised the interest for automated manual transmission (AMT) systems. An AMT system consist of a manual transmission through the clutch disc, and an automated actuated clutch during gear shifts, and one of the AMT's largest advantages is low cost, high efficiency, reduced clutch wear and improved fuel consumption.

Most attention in this trend has been given to hydraulic clutch system for cars. Lucente et al. [2007] provides a nonlinear model for dry clutches with electrohydraulic actuator, in addition to models of the gear system, while Montanari et al. [2004] and Horn et al. [2003] concentrate on control and trajectory position tracking of similar electrohydraulic systems. In the paper by Zhang et al. [2002] we find a dynamic model for clutch engagement using hydraulic actuators, and Moon et al. [2004] presents a dynamic model for an electromechanical clutch system in his paper on clutch-by-wire system.

In this paper we will deal with the modeling of an electropneumatic clutch actuator for AMTs and clutch-by-wire systems for heavy duty trucks. There is a significant difference between the clutches designed for cars and ones designed for heavy duty trucks. In a truck, a much higher level of torque is required to be transmitted from the motor through the clutch disc, and hence are these discs needed to be of much larger radius than the ones for cars. Another difference is that in trucks pressurized air is available. This is the reason why it will be preferred to use pneumatics to actuate the clutch system, even though this complicates the modeling task due to the compressibility of air.

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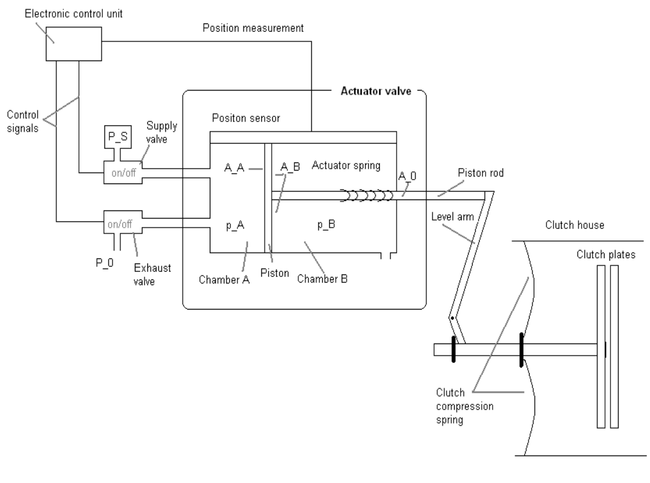

Fig. 1. Drawing of the electropneumatic clutch system

The considered clutch system is the same as the one treated by Kaasa in his Ph.D. thesis, Kaasa [2006]. The allocation of air to the clutch actuator, which were controlled by a three-way proportional valve, is in this paper assumed to be controlled by a set of on/off-valves. These have a dynamic response that is harder to model accurately, but they are desired because of space, cost and robustness advantages.

Figure 1 show a drawing of the main components of the clutch system, which is a pull-type clutch. In addition to the on/off valves, the system consist of an electronic control unit, a position sensor and an actuator valve. The state of the clutch plates (engaged/disengaged/slipping) is decided by the position of the clutch actuator piston, which is a result of the forces acting on the piston. These forces are mainly pressure, friction and clutch load. The clutch load characteristic is a lumped force of the clutch compression spring and a counteracting, much weaker, actuator spring. The compression spring is a stiff and highly nonlinear diaphragm spring, while the actuator spring on the other hand is a linear coil spring.

It is desired to be able to estimate the clutch load characteristic during the lifetime of the clutch, as the characteristic changes as the clutch wears. These changes have a large influence on the system, and will introduce large errors if not recalculated during the clutch's lifetime. We will therefore find a model for the clutch load characteristics which is suitable for parameter estimation, and estimate these parameters off-line. It is also desired that position is the only measured state of the system. This is another reason why we are so interested in the clutch load. If this is know, it is possible to calculate the pressure in the system from a dynamic model, as velocity can be derived from the position measurement.

This paper is organized as follows. In Section 2 we give a description of the system and present the proposed model for the system. Section 3 contains the results from the parameter estimations as well as the simulations that verifies our model. Concluding remarks are given in Section 4.

2. CLUTCH ACTUATOR MODEL

2.1 Motion dynamics

The motion of the clutch actuator piston can be described by

$$M\frac{d^2y}{dt^2} = A_A p_A - A_B p_B - A_0 P_0 - f_l - f_f \tag{1}$$

where y is the position, f_f is the friction force, f_l is the clutch load force and P_0, p_A and p_B are the atmospheric pressure and the pressures in chamber A and B. The areas A_A, A_B and $A_0 = A_A - A_B$ are the areas in chamber A and B, and the difference in area of the piston in chamber A and B, and M is the mass of the piston.

2.2 Friction between piston and cylinder

The friction between the piston and the cylinder is modeled by a dynamical friction model, a LuGre model,

$$f_f = D_v v + K_z z + D_{\dot{z}}\dot{z} \tag{2}$$

where z describes a pre-sliding seal deflection, D_v is the viscous effect, K_z is the deflection stiffness and $D_{\dot{z}}$ is the deflection damping coefficient. The dynamic pre-sliding can be given by

$$\dot{z} = v - \frac{K_z}{F_C}|v|\,z \tag{3}$$

where F_C is the Coulomb friction. The parameter values are found from the work of Kaasa [2006].

2.3 Pressure

We use a simple model of the pressure dynamics

$$\dot{p}_A = -\frac{A_A v}{V_A(y)}p_A + \frac{RT_0}{V_A(y)}w_v \tag{4}$$

$$\dot{p}_B = \frac{A_B v}{V_B(y)}p_B + \frac{RT_0}{V_B(y)}w_r \tag{5}$$

where w_v and w_r are the resulting flow to/from the two chambers A and B. This model is based on 6 assumptions

- constant chamber temperature, i.e isothermal conditions
- constant supply pressure and that the exhaust pressure is equal to the atmospheric pressure
- air behaves as an ideal gas obeying the ideal gas equation of state
- the energy change in the fluid due to elevation is negligible
- the thermodynamic properties are uniformly distributed in the volume
- frictionless flow is assumed, i.e. isentropic flow

2.4 Valve dynamics

From Kaasa [2006] we have that flow through an orifice can be described by

$$\psi(r, B) = \Omega_0(r) + B\Omega_1(r, sgn(B)) \tag{6}$$

where

$$\Omega_0 = \begin{cases} \sqrt{1-r^2}, & r \in [0,1] \\ 0, & r > 1 \end{cases} \tag{7}$$

$$\Omega_1(r,+1) = -\Omega_0 + \begin{cases} 1, & r \in [0, B_0] \\ \sqrt{1-(\frac{r-B_0}{1-B_0})^2}, & r \in \langle B_0, 1] \\ 0, & r > 1 \end{cases} \tag{8}$$

$$\Omega_1(r,-1) = \Omega_0 - \begin{cases} 1-r, & r \in [0,1] \\ 0, & r > 1 \end{cases} \tag{9}$$

and where $r = \frac{p_l}{p_h}$ is the relation between the low and the high pressure at the sides of the orifice. The basis functions in (8) and (9) can be characterized as approximately isentropic or incompressible laminar flow, and the value of B decides which of these should be used.

The air flow to/from chamber A is $w_v = w_{vs} - w_{ve}$, where w_{vs} is the flow through the on/off valve controlling the supply and w_{ve} is the flow through the valve controlling exhaust. These flows can be described by

$$w_{vx} = g_{vx}(p_h, p_l, C_{vx}, B_{vx})y_{vx}(u_{vx}, R_{0,vx}, R_{1,vx})$$

where

$$g_{vx} = \rho_0 C_{vx}\psi(\frac{p_{l,vx}}{p_{h,vx}}, B_{vx})p_{h,vx} \tag{10}$$

and the valve opening degree is described

$$y_{vx} = sat_{[0,1]}(\frac{1}{R_{1,vx} - R_{0,vx}}(u_{vx} - R_{0,vx})), u_{vx} \in [0,1] \tag{11}$$

The subscript vx stands for vs and vem, $R_{0,vx}$ and $R_{1,vx}$ are valve opening constants and the parameter u_{vx} is the duty cycle of the valve's PWM (Pulse Width Modulation) input. The pressures is $p_{l,vs} = p_A$, $p_{l,ve} = P_0$, $p_{h,vs} = P_S$, $p_{h,ve} = P_B$ where P_S is the supply pressure.

The flow to/from chamber B, $w_r = w_{in} - w_{out}$ can be modeled as a fixed orifice,

$$w_{in} = \rho_0 C_r \psi(\frac{p_B}{P_0}, B_{in}) \tag{12}$$

$$w_{out} = \rho_0 C_r \psi(\frac{p_B}{P_0}, B_{out}) \tag{13}$$

2.5 Clutch load

To prepare for easy adaption of the clutch load characteristics, the model should be parameter affine and the number of parameters should be small. We propose a simple model for the load characteristic

$$f_l(y) = \Theta^T \phi(y) \tag{14}$$

where $\Theta^T = [\theta_1, \theta_2, \theta_3]^T$ and ϕ are three B-splines curves defined by the following polynomials and slope conditions.

$$\phi_1(y) = \begin{cases} 0, & y < t_1 \\ y - t_1, & t_1 \leq y < t_2 \\ a_1 y^2 + b_1 y + c_1, & t_2 \leq y < t_3 \\ a_1 t_3^2 + b_1 t_3 + c_1, & y \geq t_3 \end{cases} \tag{15}$$

- Transition between linear and quadratic part shall be smooth: $a_1 t_2^2 + b_1 t_2 + c_1 = t_2 - t_1$
- Derivative in t_2 shall be equal to one: $2a_1 t_2 + b_1 = 1$
- Derivative in t_3 shall be equal to zero: $2a_1 t_3 + b_1 = 0$

$$\phi_2(y) = \begin{cases} 0, & y < t_2 \\ a_2 y^2 + b_2 y + c_2, & t_2 \leq y < t_3 \\ y + d_2, & y \geq t_3 \end{cases} \tag{16}$$

- The quadratic part shall start at zero in t_2: $a_2 t_2^2 + b_2 t_2 + c_2 = 0$
- The quadratic part shall have a gradient equal to zero in t_2: $2a_2 t_2 + b_2 = 0$
- The spline shall be continuous in t_3: $a_2 t_3^2 + b_2 t_3 + c_2 = t_3 + d_2$
- The derivative in t_3 shall be equal to one: $2a_2 t_3 + b_2 = 1$

$$\phi_3(y) = \begin{cases} 0, & y < t_2 \\ a_3 y^3 + b_3 y^2 + c_3 y + d_3, & t_2 \leq y < t_4 \\ e_3 y^3 + f_3 y^2 + g_3 y + h_3, & t_4 \leq y < t_5 \\ 0, & y \geq t_5 \end{cases} \tag{17}$$

where $t_5 = 2t_4 - t_2$.

- The first cubic part shall start at zero in t_2: $a_3 t_2^3 + b_3 t_2^2 + c_3 t_2 + d_3 = 0$
- The first cubic part shall have a gradient equal to zero in t_2: $3a_3 t_2^2 + 2b_3 t_2 + c_3 = 0$
- The first cubic part shall have a value equal to one millimeter in t_4: $a_3 t_3^4 + b_3 t_4^2 + c_3 t_4 + d_3 = 0.001$
- The first cubic part shall have a gradient equal to zero in t_4: $3a_3 t_4^2 + 2b_3 t_4 + c_3 = 0$
- The second cubic part shall start at zero in $2t_4 - t_2$: $e_3(2t_4 - t2)^3 + f_3(2t_4 - t2)^2 + g_3(2t_4 - t2) + h_3 = 0$
- The second cubic part shall have a gradient equal to zero in $2t_4 - t_2$: $3e_3(2t_4 - t2)^2 + 2f_3(2t_4 - t2) + g_3 = 0$
- The second cubic part shall have a value equal to one millimeter in t_4: $e_3 t_4^3 + f_3 t_4^2 + g_3 t_4 + h_3 = 0.001$
- The second cubic part shall have a gradient equal to zero in t_4: $3e_3 t_4^2 + 2f_3 t_4 + g_3 = 0$

Rewritten we have

$$\xi_i = A_i^{-1} B_i \qquad i = 1, 2, 3 \tag{18}$$

where

$$A_1 = \begin{bmatrix} t_2^2 & t_2 & 1 \\ 2t_2 & 1 & 0 \\ 2t_3 & 1 & 0 \end{bmatrix}$$

$$A_2 = \begin{bmatrix} t_2^2 & t_2 & 1 & 0 \\ 2t_2 & 1 & 0 & 0 \\ t_3^2 & t_3 & 1 & -1 \\ 2t_3 & 1 & 0 & 0 \end{bmatrix}, \tag{19}$$

$$A_3 = \begin{bmatrix} t_2^3 & t_2^3 & t_2 & 1 & 0 & 0 & 0 & 0 \\ 3t_2^2 & 2t_2 & 1 & 0 & 0 & 0 & 0 & 0 \\ t_4^3 & t_4^2 & t_4 & 1 & 0 & 0 & 0 & 0 \\ 3t_4^2 & 2t_4 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & (2t_4 - t_2)^3 & (2t_4 - t_2)^2 & (2t_4 - t_2) & 1 \\ 0 & 0 & 0 & 0 & 3(2t_4 - t_2)^2 & 2(2t_4 - t_2) & 1 & 0 \\ 0 & 0 & 0 & 0 & t_4^3 & t_4^2 & t_4 & 1 \\ 0 & 0 & 0 & 0 & 3t_4^2 & 2t_4 & 1 & 0 \end{bmatrix}$$

$$B_1 = \begin{bmatrix} t_2 - t_1 \\ 1 \\ 0 \end{bmatrix}, B_2 = \begin{bmatrix} 0 \\ 0 \\ t_3 \\ 1 \end{bmatrix}, B_3 = \begin{bmatrix} 0 \\ 0 \\ 0.001 \\ 0 \\ 0 \\ 0 \\ 0.001 \\ 0 \end{bmatrix} \tag{20}$$

and the ξ_i's are the collection of parameters a_i, b_i.... The knots are chosen, based on knowledge of the considered clutch, to be in the positions $t = [0\ 2\ 8.5\ 11.5]^T\ mm$, and we get the following parameters

$$\xi_1 = \begin{bmatrix} -76.92308 \\ 1.3077 \\ -3.0769 \cdot 10^{-4} \end{bmatrix}$$

$$\xi_2 = \begin{bmatrix} 7.6923 \cdot 10^{-2} \\ -3.0769 \cdot 10^{-4} \\ 3.0769 \cdot 10^{-7} \\ -8.4968 \cdot 10^{-3} \end{bmatrix} \tag{21}$$

$$\xi_3 = \begin{bmatrix} 47.2372 \\ -2.3327 \cdot 10^3 \\ -0.1610 \\ 1.6900 \\ 1.5163 \cdot 10^{-4} \\ -6.9439 \cdot 10^{-3} \\ -113.7192 \\ 2.3327 \cdot 10^3 \end{bmatrix}$$

The resulting B-spline curves are shown in Figure 2.

2.6 Model

Summarized we get the 5^{th} order model

$$\begin{aligned} \dot{y} &= v \\ \dot{v} &= \frac{1}{M}(A_A p_A - A_B p_B - A_0 P_0 - f_l - f_f) \\ \dot{p}_A &= -\frac{A_A v}{V_A(y)} p_A + \frac{RT_0}{V_A(y)} w_v \\ \dot{p}_B &= \frac{A_B v}{V_B(y)} p_B + \frac{RT_0}{V_B(y)} w_r \end{aligned} \tag{22}$$

$$\dot{z} = v - \frac{K_z}{F_C} |v|\, z \tag{23}$$

and all the parameter values can be found in the Appendix.

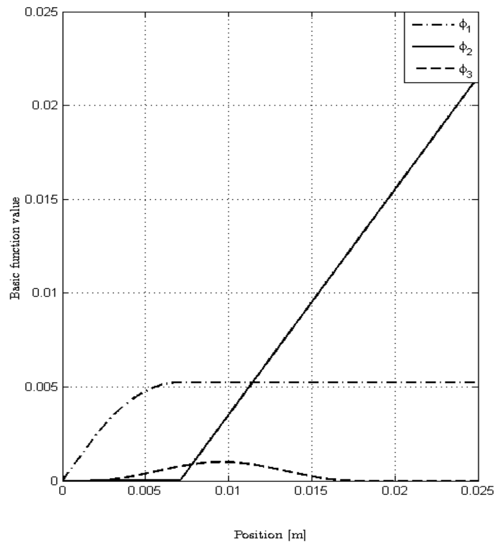

Fig. 2. The B-spline basis functions

3. EXPERIMENTAL RESULTS

3.1 Parameter estimation

The parameter estimation for the load characteristic is conducted by use of the *lsqnonlin* algorithm in MatLab, where the cost function is the sum of the position errors. The position error vector is found from either ballistic or partitioned ballistic simulations. In partitioned ballistic simulation, the simulation is initialized by the position measurement after each N^{th} simulation step. As we have no measurement of the other state variables, we initialize these by the values of the former simulation step. For the first simulation the initial values

$$\begin{aligned} y_0 &= y_{mes} \\ v_0 &= 0 \\ p_{A0} &= P_0 \\ p_{B0} &= P_0 \\ z_0 &= 0 \end{aligned} \tag{24}$$

are used. In ballistic simulation no new initialization is done during the simulation. All simulations are conducted with variable step length and *ode45* as the chosen solver.

The measurement used as reference is position and pressure measurement obtained from the truck Gamal at Kongsberg Automotive ASA. The operation range for the clutch is $y = [0, 25]$ mm, and the model is needed to be most precise in the region around engaging/disengaging of the clutch, in this case, the region $y = [5 - 8]$ mm.

The optimization routine for finding the parameters is conducted twice. The first one takes large steps and finds good initial values for further optimization. The second takes smaller optimization steps and finds the best parameters starting from these initial values.

Est. method	θ_1	θ_2	θ_3
Ballistic	$9.3850 \cdot 10^5$	$-0.5817 \cdot 10^5$	$0.3909 \cdot 10^5$
Part. ball., N=2	$8.4554 \cdot 10^5$	$-0.1678 \cdot 10^5$	$0.0971 \cdot 10^5$
Part. ball., N=4	$8.5581 \cdot 10^5$	$-0.2233 \cdot 10^5$	0
Part. ball., N=8	$9.8251 \cdot 10^5$	$-0.3468 \cdot 10^5$	$0.5386 \cdot 10^5$
Part. ball., N=16	$8.9178 \cdot 10^5$	$-0.3527 \cdot 10^5$	0
Part. ball., N=32	$8.8767 \cdot 10^5$	$-0.5476 \cdot 10^5$	0

Table 1. Estimated clutch load parameters

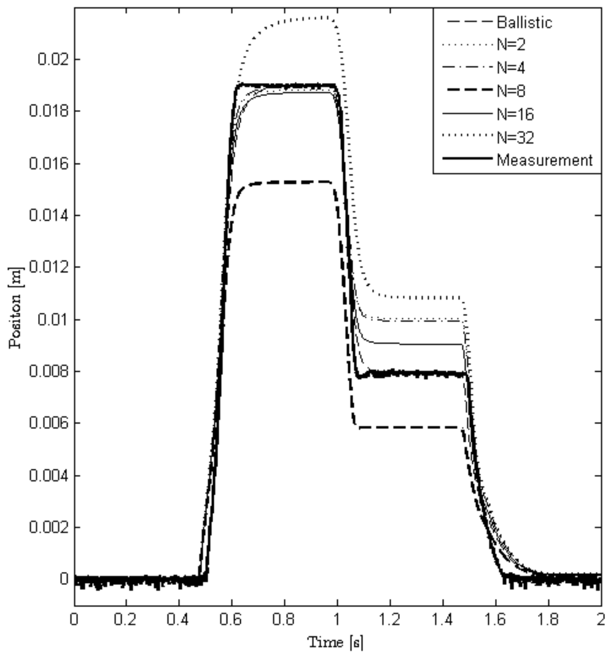

Fig. 3. Simulated positions

3.2 Results

Estimated parameters from ballistic and five partitioned ballistic simulations are presented in the Table 1 below. The number N stands for the number of partitions of the data set. Simulations results with these parameters are shown in Figure 3 and 4. Figure 5 show all the resulting clutch load characteristics. An estimation of the clutch load characteristic in the system is calculated from

$$f_l = A_A(p_A - p_0) \tag{25}$$

which is derived from motion dynamics equation (1) by assuming $v = 0$ and $p_B = P_0$.

The average position and pressure errors are presented in Table 2. From these and the simulation results we see that the parameters obtained from ballistic simulations gives the best results in position and the partitioned ballistic simulations with $N = 32$ gives the best results if considering the pressure of chamber A.

To verify that the estimated parameters gives a good model of the system, simulation with another data set from the truck have been done. Figure 6 and 7 show all the states, in addition to the measured position and pressure, from simulations with Θ obtained from ballistic simulations and from partitioned ballistic simulations with

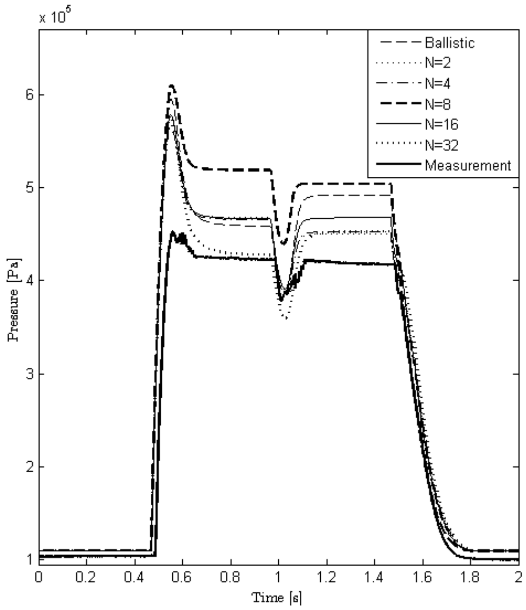

Fig. 4. Simulated pressure in chamber A

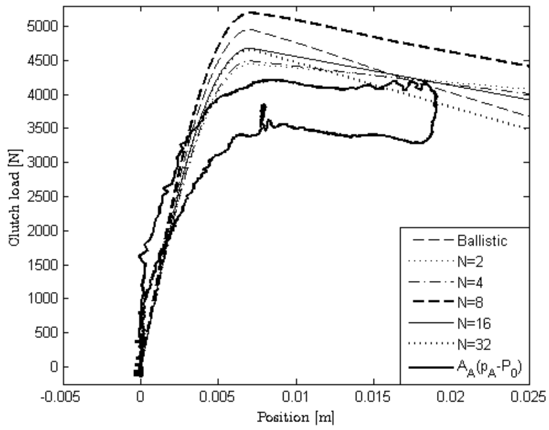

Fig. 5. Clutch load characteristic

Estimation method	Position error	Pressure error
Ballistic	$0.2678 \cdot 10^{-4}$	$2.7602 \cdot 10^{12}$
Part. ballistic, N=2	$8.8691 \cdot 10^{-4}$	$1.7203 \cdot 10^{12}$
Part. ballistic, N=4	$8.7327 \cdot 10^{-4}$	$1.7330 \cdot 10^{12}$
Part. ballistic, N=8	$3.3932 \cdot 10^{-4}$	$5.3149 \cdot 10^{12}$
Part. ballistic, N=16	$2.5684 \cdot 10^{-4}$	$2.1805 \cdot 10^{12}$
Part. ballistic, N=32	$4.1361 \cdot 10^{-3}$	$1.020 \cdot 10^{12}$

Table 2. Position and pressure errors

$N = 32$. The main goal of the clutch load estimation, is to be able to calculate the pressure of chamber A. By, as above, assuming that $V = 0$ and $p_B = P_0$ we can calculate this from

$$p_A = \frac{1}{A_A}(f_l + A_A * P_0). \tag{26}$$

Figure 8 show measured pressure together with pressures estimated this way, and, as expected, estimation from simulations with Θ from partitioned ballistic simulations with $N = 32$ give the best results.

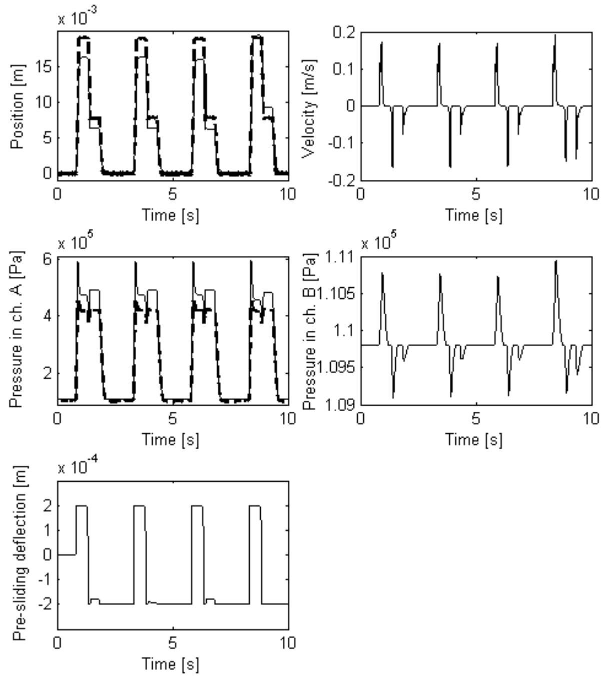

Fig. 6. All states from simulations with Θ from ballistic simulations

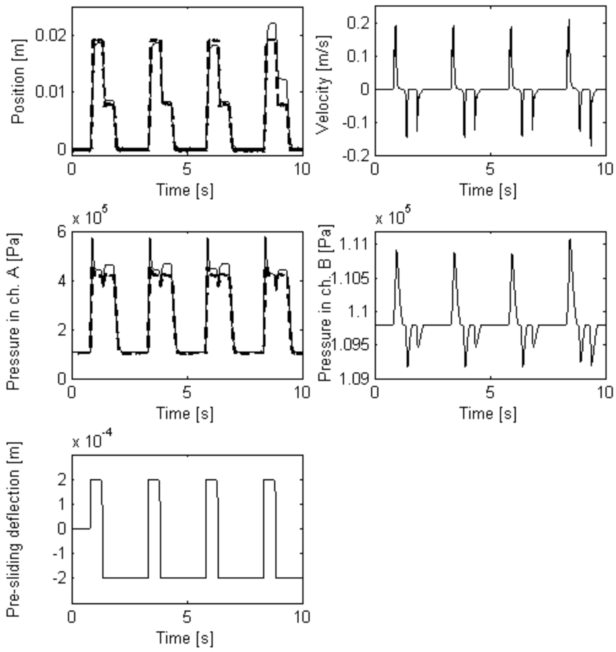

Fig. 7. All states from simulations with Θ from part. ballistic simulations with $N = 32$

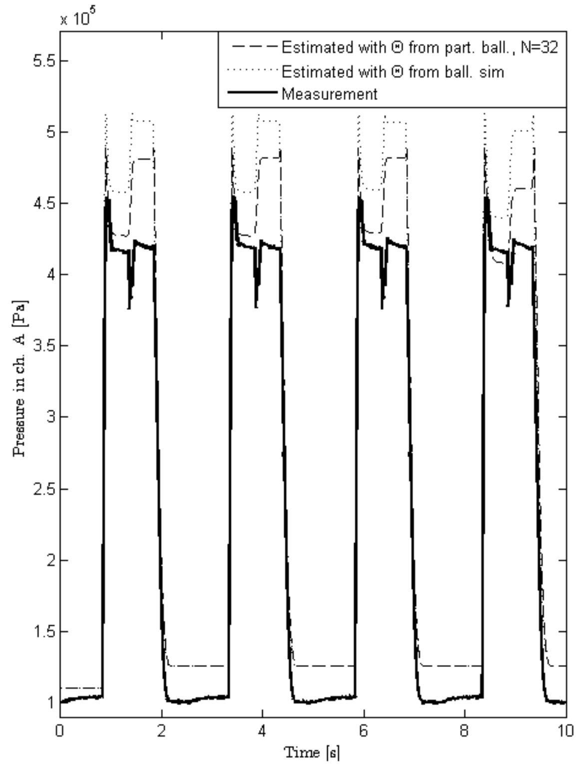

Fig. 8. Pressure in chamber A, estimated from $\frac{1}{A_A}(f_l + A_A P_0)$

Simulated position and the simulated pressure in chamber A deviate some from the measurement from the truck, which is not surprising as the proposed model is rather simple. From the clutch load characteristics in Figure 5, it is clear that the proposed model do represent the system completely. There is also some uncertainty associate with the value of the supply pressure, which also may result in simulation deviations. The estimated pressure also suffer from these uncertainties. These results show the importance of an on-line adaption of the clutch load, especially since also wear of the clutch and temperature variations will change the truck's clutch load characteristics.

4. CONCLUDING REMARKS

The main purpose of this paper was to provide a rather simple model for the clutch actuator system, which is suited for on-line adaption of clutch load characteristic parameters. A 5^{th} order model of an electropneumatic clutch actuator have been presented, and parameters for clutch load characteristic are estimated off-line. The simulations verifies that the proposed model is a fair representation of the system, and that the model is a good basis for further development of the model, where emphasis should be placed on on-line adaption of clutch load characteristics and construction of a nonlinear observer.

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Appendix A. PARAMETERS

Par.	**Value**	**Unit**	**Description**
A_A	$12.3 \cdot 10^{-3}$	m^2	Area of chamber A
A_B	$11.848 \cdot 10^{-3}$	m^2	Area of chamber B
$V_{0,A}$	$0.148 \cdot 10^{-3}$	m^3	Vol. of ch. A, $y = 0$
$V_{0,B}$	$1.2 \cdot 10^{-4}$	m^3	Vol. of ch. B, $y = 0$
P_0	$1.095 \cdot 10^5$	Pa	Ambient pressure
T_0	293	K	Temperature
R	288	$\frac{J}{kgK}$	Gas constant of air
M	10	kg	Mass of piston
P_S	$9 \cdot 10^5$	Pa	Supply pressure
D_v	5000	$[Ns/m]$	Viscous effect
K_z	$1 \cdot 10^6$	$[N/m]$	Deflect. stiffness
$D_{\dot{z}}$	5000	$[Ns/m]$	Deflect. damp. coeff.
F_C	200	$[N$	Coloumb friction
$R_{0,vs}$	0.07	-	Valveop. const.
$R_{0,ve}$	0.07	-	Valveop. const.
$R_{1,vs}$	0.93	-	Valveop. const.
$R_{1,ve}$	0.93	-	Valveop. const.
B_{vs}	0.1	-	Flow parameter
B_{ve}	-0.7	-	Flow parameter
C_{vs}	$19 \cdot 10^{-9}$	$[\frac{m^3}{Pa \cdot s}]$	Conductance
C_{ve}	$21.4 \cdot 10^{-9}$	$[\frac{m^3}{Pa \cdot s}]$	Conductance
C_r	$5 \cdot 10^{-8}$	$[\frac{m^3}{Pa \cdot s}]$	Conductance
B_0	0.582	-	Critical flow for air
B_{in}	0.5	-	Flow parameter
B_{out}	0.5	-	Flow parameter