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Simulation and control of an electro-hydraulic actuated clutch

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ABSTRACT

The basic function of any type of automotive transmission is to transfer the engine torque to the vehicle with the desired ratio smoothly and efficiently and the most common control devices inside the transmission are clutches and hydraulic pistons. The automatic control of the clutch engagement plays a crucial role in Automatic Manual Transmission (AMT) vehicles, being seen as an increasingly important enabling technology for the automotive industry. It has a major role in automatic gear shifting and traction control for improved safety, drivability and comfort and, at the same time, for fuel economy. In this paper, a model for a wet clutch actuated by an electro-hydraulic valve used by Volkswagen for automatic transmissions is presented. Starting from the developed model, a simulator was implemented in Matlab/Simulink and the model was validated against data obtained from a test-bench provided by Continental Automotive Romania, which includes the Volkswagen wet clutch actuated by the electro-hydraulic valve. Then, a predictive control strategy is applied to the model of the electro-hydraulic actuated clutch with the aims of controlling the clutch piston displacement and decreasing the influence of the network-induced delays on the control performances. The simulation results obtained with the proposed method are compared with the ones obtained with different networked controllers and it is shown that the strategy proposed in this paper can indeed improve the performances of the control system.

1. Introduction

Nowadays, clutch pedals as well as automatic transmissions, double-clutch transmissions, hybrid drive concepts and chassis control systems increasingly require open-loop and closed-loop controlled actuators. The introduction of a new actuator opens up new opportunities for controlling the engine and drive-line, and new strategies that can improve the drive-line performance are predictable. In the last decades, the use of control systems for automated clutch and transmission actuation has been constantly increasing; a clear trend is that automatic clutch systems will be introduced and used in a wider variety of applications, which would benefit from advanced clutch control. For example, start and stop strategies can be employed and in addition the clutch control can be utilised in automated manual transmissions to reduce the time for gear changes. Furthermore, clutch control is also a factor in look-ahead control.

Recent attention has focused on modelling different valve types used as actuators in automotive control systems: physics-based nonlinear model for an exhausting valve [1], nonlinear state-space model description of the actuator that is derived based on physical principles and parameter identification [2,3], nonlinear physical model for programmable

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valves [4], nonlinear model of an electro-magnetic actuator used in brake system based on system identification [5], mathematical model obtained using identification methods for a valve actuation system of an electro-hydraulic engine [6], linear model constructed based on a grey-box approach which combines mathematical modelling and system identification for an electro-magnetic control valve [7], input–output simplified mathematical model and state-space mathematical model for an electro-hydraulic valve actuator, both based on parameter identification and physical laws [8].

Also, during the last years, the automated clutch actuators have been actively researched and different models and control strategies have been developed: a model for an electro-hydraulic valve used as actuator for a wet clutch [9], dynamic modelling and control of electro-hydraulic wet clutches [10], PID control for a wet plate clutch actuated by a pressure reducing valve [11], predictive and piecewise LQ control of a dry clutch engagement [12], switched control of an electro-pneumatic clutch actuator [13], Model Predictive Control of a two stage actuation system using piezoelectric actuators for controllable industrial and automotive brakes and clutches [14], explicit Model Predictive Control of an electro-pneumatic clutch actuator [15].

All of the above control solutions assume that the sensors, controllers and actuators are directly connected, which is not realistic. Rather, in modern vehicles, the control signals from the controllers and the measurements from the sensors are exchanged using a communication network, e.g., Controller Area Network (CAN) or Flexray, among control system components, yielding a so called Networked Control System (NCS). Although these NCSs brought many attractive advantages, which include: low cost, simple installation and maintenance, increased system agility, higher reliability and greater flexibility, this also brings up a new challenge on how to deal with the effects of the network-induced delays and packet losses in the control loop. The delays may be unknown and time-varying and may degrade the performances of control systems designed without considering them and can even destabilise the closed-loop system.

The predictive control techniques were introduced mainly in order to deal with plants that have complex dynamics (unstable inverse systems, time-varying delay, etc.) and plant model mismatch. These strategies are of a particular interest from the point of view of both broad applicability and implementation simplicity, being applied on large scale in industry processes, having good performances and being robust at the same time. They were initially utilised for slow processes: oil refineries, petrochemicals, pulp and paper, primary metal industries, gas plant [16], but starting with the evolution of hardware components and algorithms, the possibility to implement these types of control algorithms to fast processes, which have reduced sampling periods, appeared: vehicle engine and traction control, aero-spatial applications, autonomous vehicles, power generation and distribution [17].

Some predictive control algorithms were already proposed for different vehicle subsystems: Anti-lock Braking System (ABS) control [18], Vehicle Dynamics Control (VDC) [19], vehicle magnetic actuators [20], middle-layer control [21], but without taking into account the delays that can appear in a networked environment.

As such, in this paper, an input–output model for a wet clutch actuated by an electro-hydraulic valve used by Volkswagen for automatic transmission is developed. The electro-hydraulic valve used to control the wet clutch is not a typical one, being especially designed for Volkswagen vehicles. This type of valve was not modelled in literature and a proper model was needed in order to control the wet clutch with the aim of lowering emissions, reducing fuel consumption and increasing comfort. The designed electro-hydraulic clutch actuator model has as input the supply voltage and as outputs the clutch pressure and the clutch piston displacement. Starting from the developed model, a simulator was implemented in Matlab/Simulink and the model was validated against data obtained from a test-bench provided by Continental Automotive Romania, which includes the Volkswagen DQ250 wet clutch actuated by the electro-hydraulic valve DQ500. The simulations are very similar to the experimental data proving that the modelling approach is suitable to this kind of clutch actuated by an electro-hydraulic valve.

Using the developed input–output model for the valve–clutch system, an appropriate networked controller based on a predictive strategy is proposed in order to control the clutch piston displacement, while decreasing the influence of the variable time delay induced in the CAN-based NCS on the control performance. The plant is a subsystem of the automatic transmission of a Volkswagen vehicle and the main control goal is to make the clutch plates position track a given external reference, while taking into account the delays that appear in the NCS. The results obtained with the proposed method are compared with the ones obtained with different networked controllers and it is shown that the strategy presented in this paper can indeed improve the performances of the control system.

2. Modelling of the actuator–clutch system

In this section an input–output mathematical model based on physical principles for flow and fluid dynamics for the electro-hydraulic actuated clutch is presented.

2.1. Structure and functional operation

Schematics of the electro-hydraulic valve actuator and wet multi-plate clutch layout are given in Fig. 1. A pump produces the line pressure P_S used as input for the electro-hydraulic actuator represented by a pressure reducing valve. This valve realises a pressure P_R on the clutch side, depending on the current i in the solenoid, which will have as consequence the magnetic force F_{mag} exerted on the valve plunger, which moves linearly within a bounded region under the effect of this force. Such a force is generated by a solenoid placed at one boundary of the region. The magnetic force is a

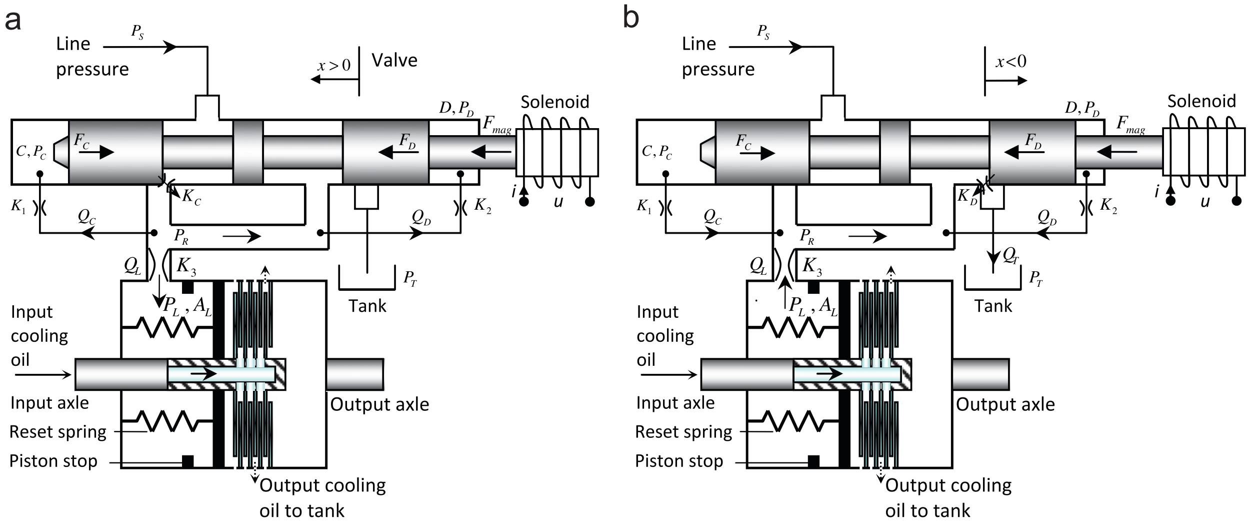

Fig. 1. Actuator and clutch system: (a) charging phase and (b) discharging phase.

function of the solenoid current i and the displacement x, defined by

$$F_{mag} = f(i) = \frac{k_a i^2}{2(k_b + x)^2}; \quad L\frac{di}{dt} + Ri = u,$$

where k_a and $k_b$are constants, L is the solenoid induction, R the resistance and u is the supply voltage.

The pressure to be controlled P_R is sensed on the plunger end areas C and D and compared with the magnetic force F_{mag}. The two forces F_C and F_D from the two sensed pressure chambers generate the feedback force $F_{feed} = F_C - F_D$. Depending on the valve plunger position, there are two phases: the charging phase, when the magnetic force is greater than the feedback force and the valve plunger is moved to the left, connecting the source with the hydraulic actuated clutch (Fig. 1a), and the discharging phase, when the magnetic force is switched off or has a lower value than the feedback force so that the valve plunger is moved to the right, connecting the hydraulic actuated clutch to the tank (Fig. 1b).

The wet clutch is a chamber with a piston as represented in Fig. 1. In the charging phase when the valve plunger is moved to the left and the displacement x is considered positive, the oil flows from the source through the valve to the clutch and the piston in the clutch moves towards the clutch plates compressing them. In the discharging phase, when the valve plunger is moved to the right and the displacement x is negative, the clutch piston moves to the left and the oil flows from the clutch chamber through the valve to the tank.

2.2. Input–output model of the actuator–clutch system

In [8] two models for an electro-hydraulic actuator were developed: an input–output model, where simplifications were made in order to obtain a suitable transfer function to be implemented in Matlab–Simulink, and a state-space model. Starting from the equations that describe the actuator input–output model, an input–output model for a wet clutch actuated by an electro-hydraulic valve used by Volkswagen for automatic transmission was developed in [22] and it is presented in this subsection.

Comparing the magnetic force and the feedback force it results a force balance, which describes the plunger motion and the output pressure. This equation of force balance is the same for both positive and negative displacements of the plunger

$$F_{mag} - CP_C + DP_D = M_v s^2 x + K_e x, \tag{1}$$

where P_C represents the pressure of the left sensed chamber, P_D the pressure of the right sensed chamber, M_v is the plunger mass, $K_e = 0.43w(P_{S_0} - P_{R_0})$ represents the flow force spring rate, P_S is the supply pressure, P_R is the reduced pressure, $P_{S_0}, P_{R_0}$are the nominal values of the pressures, w represents the area gradient of main orifice (from the source to the wet clutch), x is the plunger displacement and s represents the Laplace operator.

In order to capture the dynamical behaviour of the system, physical laws were applied for the charging phase of the pressure reducing valve, illustrated in Fig. 1a. The linearised continuity equations which describe the dynamics from the sensed pressure chambers are

$$Q_C = K_1(P_R - P_C) = \frac{V_C}{\beta_e} sP_C - Csx, \tag{2}$$

$$Q_D = K_2(P_R - P_D) = \frac{V_D}{\beta_e} sP_D + Dsx, \tag{3}$$

where K_1, K_2 are the flow-pressure coefficients of restrictors, V_C, V_D are the sensed chambers volumes and β_e represents the effective bulk modulus.

Using the flow through the left and right sensed chambers, the flow through the main orifice from the source to the wet clutch and the clutch flow, the linearised continuity equation at the chamber of the pressure being controlled is

$$K_C(P_S-P_R)-Q_L-K_lP_R-K_1(P_R-P_C)-K_2(P_R-P_D)+K_qx=\frac{V_t}{\beta_e}sP_R, \tag{4}$$

where Q_L is the clutch flow, K_C is the flow-pressure coefficient of main orifice, K_q is the flow gain of main orifice, K_l is the leakage coefficient and V_t represents the total volume where the pressure is being controlled.

These equations define the actuator dynamics and combining them into a more useful form, solving (2) and (3) with respect to P_C and P_D and substituting into (4) yields after some manipulations

$$(K_CP_S-Q_L)\left(\frac{s}{\omega_1}+1\right)\left(\frac{s}{\omega_2}+1\right)+K_qx\left[1+\left(\frac{1}{\omega_1}+\frac{1}{\omega_2}+\frac{C}{K_q}-\frac{D}{K_q}\right)s\right.$$
$$\left.+\left(\frac{1}{\omega_1\omega_2}+\frac{C}{K_q\omega_2}-\frac{D}{K_q\omega_1}\right)s^2\right]=P_RK_{ce}\left(\frac{s}{\omega_1}+1\right)\left(\frac{s}{\omega_2}+1\right)\left(\frac{s}{\omega_3}+1\right), \tag{5}$$

where $\omega_1=\beta_eK_1/V_C$, $\omega_2=\beta_eK_2/V_D$ are the break frequencies of the left and right sensed chambers, respectively, $\omega_3=\beta_eK_{ce}/V_t$ is the break frequency of the main volume and $K_{ce}=K_C+K_l$ represents the equivalent flow-pressure coefficient. The model (5) was obtained considering that $V_C/V_t \ll 1$, $V_D/V_t \ll 1$.

In the discharging phase of the pressure reducing valve, illustrated in Fig. 1b, the dynamical behaviour of the system is obtained by applying physical laws to the oil flow. The linearised continuity equations at the sensed pressure chambers are similar with (2) and (3), but having changed sign. Using the flow through the left and right sensed chambers, the flow through the main orifice from the wet clutch to the tank and the clutch flow, the linearised continuity equation obtained for the chamber of the pressure being controlled is

$$Q_L+K_1(P_C-P_R)+K_2(P_D-P_R)-K_D(P_R-P_T)-K_lP_R+K_qx=\frac{V_t}{\beta_e}sP_R, \tag{6}$$

where K_D is the flow-pressure coefficient of main orifice and P_T represents the tank pressure.

In an entire analogue manner like demonstrated for the charging phase, again making the assumptions that $V_C/V_t \ll 1$, $V_D/V_t \ll 1$ and considering that the flow-pressure coefficients of the main orifices are equal, $K_D=K_C$, the final form for the reducing valve model in the discharging phase was obtained

$$(K_DP_T+Q_L)\left(\tfrac{s}{\omega_1}+1\right)\left(\tfrac{s}{\omega_2}+1\right)+K_qx\left[1+\left(\tfrac{1}{\omega_1}+\tfrac{1}{\omega_2}+\tfrac{C}{K_q}-\tfrac{D}{K_q}\right)s+\left(\tfrac{1}{\omega_1\omega_2}+\tfrac{C}{K_q\omega_2}-\tfrac{D}{K_q\omega_1}\right)s^2\right]=P_RK_{ce}\left(\tfrac{s}{\omega_1}+1\right)\left(\tfrac{s}{\omega_2}+1\right)\left(\tfrac{s}{\omega_3}+1\right), \tag{7}$$

In order to obtain the actuator model the pressures P_C and P_D from (2) and (3) are replaced in (1) resulting after some manipulations

$$F_1-sx\left[\frac{C^2}{K_1}\Big/\left(\frac{s}{\omega_1}+1\right)+\frac{D^2}{K_2}\Big/\left(\frac{s}{\omega_2}+1\right)\right]=xK_e\left(\frac{s^2}{\omega_m^2}+1\right),$$

where$\omega_m=\sqrt{K_e/M_v}$ notes the mechanical natural frequency of the actuator plunger and

$$F_1=F_{mag}-C\frac{P_R}{((s/\omega_1)+1)}+D\frac{P_R}{((s/\omega_2)+1)},$$

For the clutch model, the first equation arises by applying Newton's second law to the forces on the piston, resulting

$$A_LP_L=M_ps^2x_p+B_fsx_p+Kx_p, \tag{8}$$

where A_L is the area of piston, P_L the pressure from the piston chamber, x_p the piston displacement, M_p the total mass of the piston, K the load spring gradient and B_f is the viscous damping coefficient of the piston.

Applying the continuity equation to the piston chamber yields

$$Q_L=K_3(P_R-P_L)=\frac{V_L}{\beta_e}sP_L+A_Lsx_p, \tag{9}$$

where K_3 is the flow-pressure coefficient of the pipe from valve actuator to the clutch and V_L is the piston chamber volume.

Eqs. (1) and (5), for the charging phase of the actuator, and (1) and (7) for the discharging phase of the actuator together with the piston dynamics given by (8) and (9), define the actuator–clutch system dynamic model.

Starting from these equations, a schematic diagram of the transfer functions for the actuator–clutch system was created and represented in Fig. 2. It can be seen that a switch block was used in order to commutate between the two phases that describe the functionality of the actuator, the charging and the discharging phase. The sign of the displacement of the plunger was used as the switch parameter, with the threshold α, thus selecting different perturbations for positive or negative displacements of the plunger.

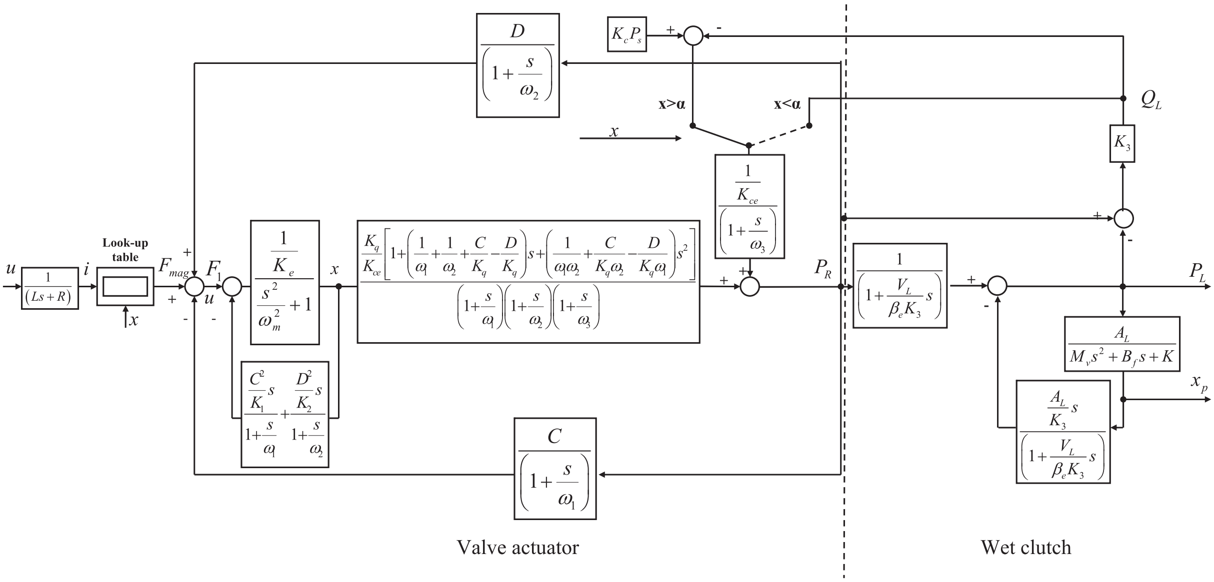

Fig. 2. Transfer function block diagram of valve–clutch system.

3. Networked predictive control strategy

The actuator–clutch system, which is a subsystem of the automatic transmission from a Volkswagen vehicle, is controlled over a communication network (i.e., CAN), so the control signals from the controller, which is implemented in an Electronic Control Unit (ECU), have to be sent to the actuator (valve) through a communication network; also, the values of the measurements from the sensor have to be sent to the controller through the same network, resulting a NCS. This NCS introduces a variable time delay in the control loop, which can degrade the performances of the control system; that is the reason why the delays have to be considered in the controller design, in order to be compensated.

To this end, in this section, firstly, an equivalent ARX actuator–clutch model is identified, secondly, the predictive control problem is solved for a given time delay and, thirdly, three methods on how to deal with the time-varying network-induced delays are proposed. Also, at the end of this section, the control architecture is described.

3.1. CARIMA model of the actuator–clutch system

In order to apply the predictive control strategy that will be described in Section 3.2, a CARIMA model for the valve–clutch system was developed

$$A(z^{-1})y(k) = z^{-d}B(z^{-1})u(k-1) + \frac{e(k)C(z^{-1})}{D(z^{-1})}, \tag{10}$$

considering as input the supply voltage and as output the clutch piston displacement. In (10) d is the delay of the plant, $e(k)$ is white noise with zero mean value, $A(z^{-1})$, $B(z^{-1})$ are the system polynomials with the degrees n_A and n_B and $C(z^{-1})$, $D(z^{-1})$ are the disturbances polynomials.

The system was identified with an ARX equivalent model employing the simulation model, utilising as input a PseudoRandom Binary Sequence (PRBS) signal. These sequences are successions of squared pulses, width modulated, that approximate a discrete white noise and their richness in frequencies helps capturing the dynamical behaviour of the system.

The ARX model is given by the following system polynomials:

$$A(z^{-1}) = 1 - 1.781z^{-1} + 0.8039z^{-2}$$

$$B(z^{-1}) = 0.00003312z^{-1} + 0.0001122z^{-2}, \tag{11}$$

For disturbances model, the polynomial C is considered to equal one and $D(z^{-1}) = 1 - z^{-1}$ for obtaining a zero steady-state error.

3.2. Control strategy

In the predictive control strategies, the control action is obtained by solving a minimisation problem at each sampling period. Usually, a sequence of future predicted control actions are calculated, but only the first is, in fact, sent to the actuator. At the next sampling period, the optimisation problem is solved again with the new measurements coming from the sensors and the control action is recalculated.

Considering the plant described by the Controlled AutoRegressive Integrated Moving Average (CARIMA) model (10), the prediction model is given by

$$\hat{y}(k+j|k)=G_{j-d}(z^{-1})D(z^{-1})z^{-d-1}u(k+j)+\frac{H_{j-d}(z^{-1})D(z^{-1})}{C(z^{-1})}u(k-1)+\frac{F_j(z^{-1})}{C(z^{-1})}y(k),$$

with $j=\overline{hi,hp}$, where hi is the minimum prediction horizon and hp is the prediction horizon. $u(k+j-1\big|k)$, $j=\overline{1,hc}$ is the future control, computed at time k and $\hat{y}(k+j|k)$ are the predicted values of the output, hc being the control horizon.

The two Diophantine equations presented in [23] are used to determine the polynomials $F_j(z^{-1})$, $G_j(z^{-1})$ and $H_j(z^{-1})$.

Now, considering as inputs $D(z^{-1})u(k)$ and collecting the j-step predictors in a matrix notation, the prediction model can be written as

$$\hat{\mathbf{y}}=\mathbf{G}\mathbf{u}_d+\hat{\mathbf{y}}_0,$$

where $\hat{\mathbf{y}}_0$ represents the free response and the matrix G is given in [23].

The objective function is based on the minimisation of the tracking error and on the minimisation of the controller output, the control weighting factor λ being introduced in order to make a trade-off between these objectives

$$J=(\mathbf{G}\mathbf{u}_d+\hat{\mathbf{y}}_0-\mathbf{w})^T(\mathbf{G}\mathbf{u}_d+\hat{\mathbf{y}}_0-\mathbf{w})+\lambda\mathbf{u}_d^T\mathbf{u}_d,$$

subject to $D(z^{-1})u(k+i)=0$ for $i\in[hc,hp-d-1]$, where $\mathbf{w}$ is the reference trajectory vector with the components $w(k+j\big|k)$, $j=\overline{hi,hp}$. Minimising the objective function ($\partial J/\partial\mathbf{u}_d=0$), the optimal control sequence yields as

$$\mathbf{u}_d^*=(\mathbf{G}^T\mathbf{G}+\lambda\mathbf{I}_{hc})^{-1}\mathbf{G}^T[\mathbf{w}-\hat{\mathbf{y}}_0],.$$

Using the receding horizon principle and considering that $\gamma_j, j=[hi,hp]$ are the elements of the first row of the matrix $\left(\mathbf{G}^T\mathbf{G}+\lambda\mathbf{I}_{hc}\right)^{-1}\mathbf{G}^T$, the following control algorithm results:

$$D(z^{-1})u(k)=\sum_{j=hi}^{hp}\gamma_j[w(k+j|k)-\hat{y}_0(k+j|k)],$$

3.3. Time-varying delays

In order to deal with the variable-time network-induced delays, firstly, a method to determine the upper bound of the delays is presented, and, secondly, three methods of considering the delays by the predictive control algorithm are proposed.

The equation developed in [26] is used to determine the upper bound of the communication delays that appear on Controller Area Network (CAN) in automotive applications

$$d_j\le\frac{(j+2)\cdot l}{R-\sum_{i=0}^{j-1}(l/c_i)},\tag{12}$$

where l=136 bits denote the maximum frame length including the 6 bit CS time, R=500 kbps is the rate of a high-speed CAN, c_i is the cycle length of the ith priority message and j is the priority of the node for which the upper bound is being calculated. Using typical values for the parameters, it yields that the sum of the delays from sensor-to-controller and from controller-to-actuator is randomly distributed in the interval $[0,\ 12T]$, where T=1 ms is the sampling period of the system.

Knowing the upper bound of the time-varying communication delays, in this paper, three methods of considering the communication delay proposed to be used by the predictive algorithm are discussed starting from the results presented in [24,25]:

(i) average delay value method: the delay considered by the prediction model is calculated using

$$d=\frac{d_m+d_M}{2},$$

where d_m is the minimum delay and d_M is the maximum delay that can appear in the communication network;

(ii) identification method: the delay is considered equal to the minimum delay that can appear in the communication network

$$d=d_m,$$

and instead of the polynomial B, another polynomial $\tilde{B}$, identified in order to model the system including the delays between d_m and d_M was introduced

$$\tilde{B}(z^{-1})=\tilde{b}_0+\tilde{b}_1z^{-1}+\cdots+\tilde{b}_{n_{\tilde{B}}}z^{-n_{\tilde{B}}},$$

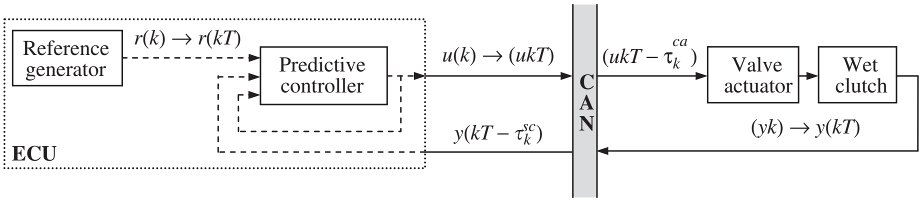

Fig. 3. Schematic diagram of the NCS.

with

$$n_{\tilde{B}} = n_B + d_M - d_m$$
$$\tilde{b}_0 = \tilde{b}_1 = \cdots = \tilde{b}_{n_{\tilde{B}}} = \frac{b_0 + b_1 + \cdots + b_{n_B}}{n_{\tilde{B}} + 1}; \quad B(1) = \tilde{B}(1),$$

(iii) gain-scheduling method: this method is applied in order to adapt the control algorithm to the variable time delays that appear in the communication network. The adaptation algorithm is derived with assumption that the average communication delays from sensor-to-controller and from controller-to-actuator are equal, both being variables. So, for every delay between d_m and d_M, the coefficients of the previous control actions, of the previous outputs and those of the future references were a priori explicitly calculated and then a look-up table was created with these coefficients. At each sampling time new coefficients are selected using as selection variable

$$d = 2\frac{\sum_{i=1}^{N} \tau^{sc_i}}{N}, \tag{13}$$

where τ^{sc_i} is the communication delay from sensor-to-controller in ith step. Therefore, the average value of N previous delays is calculated in this way. Unlike for the other two predictive methods (average delay value method and identification method), in the case of this method the upper bound of the delays can be unknown and only an estimation of it can be used in order to apply the strategy.

3.4. Control architecture

After applying the control strategy described in Section 3.2 to the CARIMA model described by (10), the structure was implemented in Matlab/Simulink and a schematic diagram of the NCS is represented in Fig. 3. The reference generator block computes and sends the reference trajectory vector to the predictive controller, which has the main control goal of making the clutch plates position track the given external reference. The valve actuator and the wet clutch blocks are illustrated in detail in Fig. 2. The valve actuator has as input the control signal send by the predictive controller through the network, the voltage u, and as output the reduced pressure P_R, which is the input of the wet clutch block. The outputs of the last block are the clutch pressure P_L and the clutch piston displacement x_p, whose value is sent as a measurement signal back through the network to the predictive controller. The predictive controller block implements the control algorithm described in Section 3.2. The delays that are induced by the communication network from sensor-to-controller τ^{sc} and from controller-to-actuator τ^{ca} can also be seen in Fig. 3.

4. Simulations and experimental results

Based on the mathematical model presented in Section 2, in this section a simulator was implemented for the actuator–clutch system and the simulation results are discussed. Also, a networked predictive controller is designed for the system while taking into account the delays that appear in the NCS.

4.1. Model validation

In order to validate the model obtained for the electro-hydraulic actuated clutch, a simulator was designed in Matlab/Simulink starting from the mathematical model described in Section 2 and the results obtained with the simulator were compared with the ones obtained on a real test-bench. The test-bench STAH—50.100, which includes the Volkswagen DQ250 wet clutch actuated by the electro-hydraulic valve DQ500, was provided by Continental Automotive Romania. The test-bench can be used to test the hydraulic command, distribution and control devices. In order to simulate more precisely the real work conditions from the plant (equipment), where the devices will be installed, the test-bench has the option to tune the three functional parameters (pressure, flow and temperature) to the real values. The tuned values can be predefined and they are hold into specific limits by a programmable logic controller (PLC).

The parameters used in simulations for the electro-hydraulic actuator, which are summarised in Table 1: the volumes of the actuator chambers, the left and right areas of the valve plunger, the flow-pressure coefficients, were obtained by measurements made on the test-bench. The test-bench also provides measurements for the outputs of the system, represented by the valve plunger displacement and the clutch pressure which are used in order to validate the implemented simulator.

Following experiments made on the test-bench from Continental Automotive Romania, using as input the supply voltage, the real-time clutch pressure response P_L from Fig. 4b was obtained. Although the input of the plant is the supply voltage, only the solenoid current i, respectively, the magnetic force F_{mag}, which are represented in Fig. 4a, were measured

Table 1
Parameters values.

Symbol	Value	Unit	Symbol	Value	Unit
K_e	1000	(N/m)	V_C	7.53e-8	(m^3)
M_v	25e-3	(kg)	V_D	1.04e-7	(m^3)
β_e	1.6e+9	(N/m^2)	V_t	3.2e-4	(m^3)
ω_1	1.17e+6	(rad/s)	C	3.66e-5	(m^2)
ω_2	5.42e+7	(rad/s)	D	2.94e-5	(m^2)
ω_3	1.04e+6	(rad/s)	α	2e-5	(m)
K_C=K_D	7.58e-11	((m^3/s)/(N/m^2))	M_p	0.5	(kg)
K_1	5.50e-10	((m^3/s)/(N/m^2))	V_L	2.51e-5	(m^3)
K_2	3.52e-9	((m^3/s)/(N/m^2))	A_l	7.75e-4	(m^2)
K_q	5.3418	((m^3/s)/(N/m^2))	K	900	(N/m)
w	3e-3	(m)	B_f	0	(N s/m)
P_S	1e+6	(N/m^2)	k_a	0.005	(N m^2/A^2)
P_T	0	(N/m^2)	k_b	0.01	(m)
K_l	2e-9	((m^3/s)/(N/m^2))	L	0.01	(H)
K_3	1.26e-8	((m^3/s)/(N/m^2))	R	0.2	(Ω)

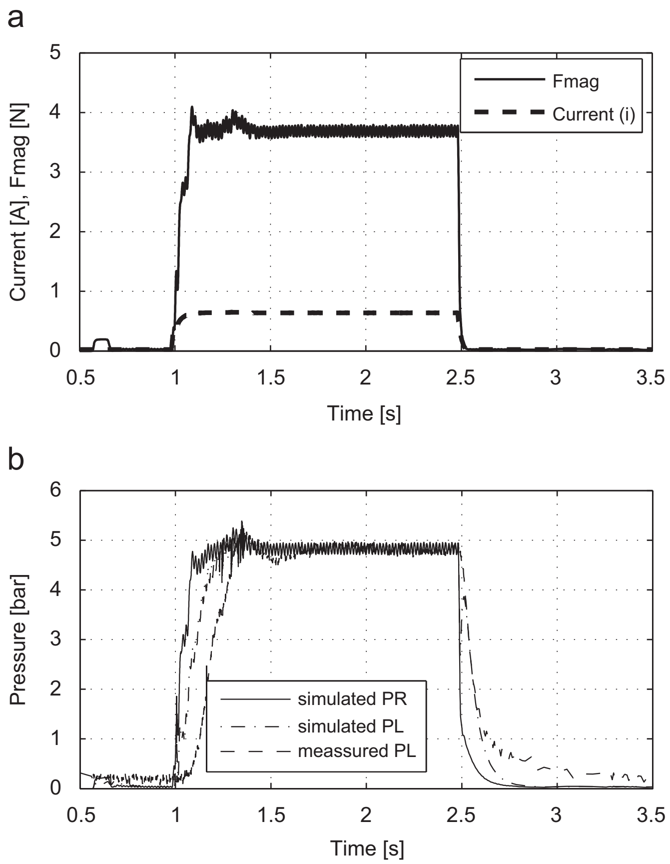

Fig. 4. Model validation: (a) input signals and (b) pressures behaviour.

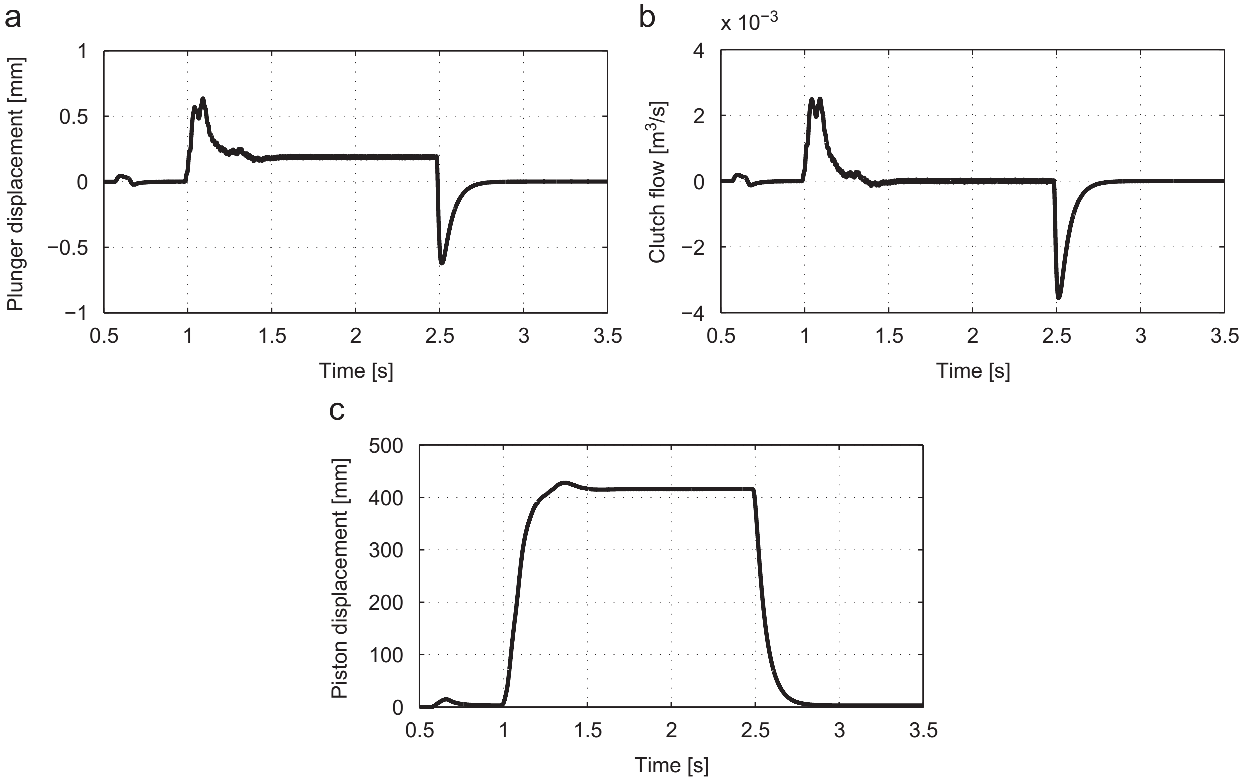

Fig. 5. Simulation results: (a) valve plunger displacement; (b) clutch flow and (c) clutch piston displacement.

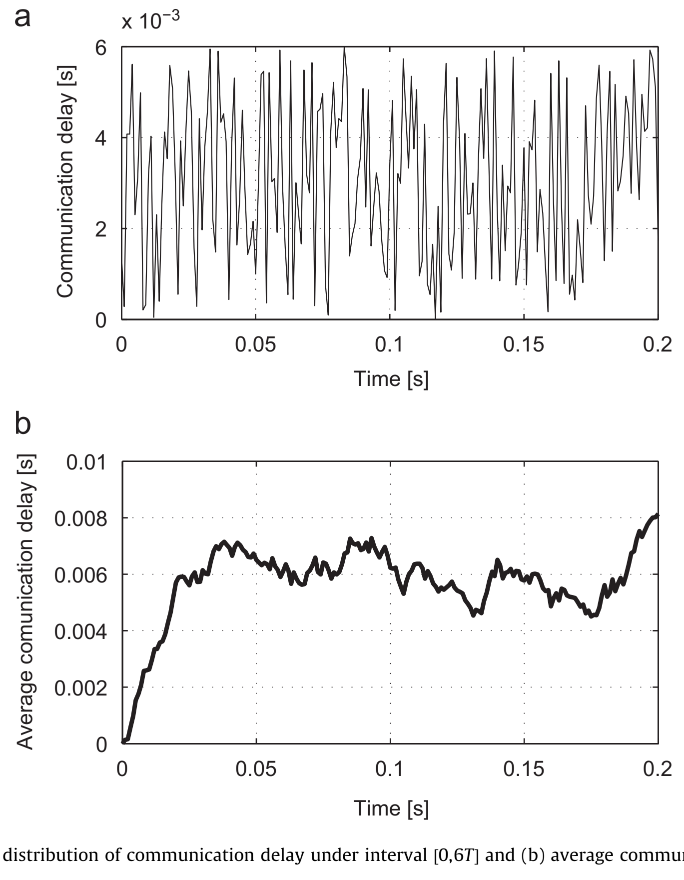

Fig. 6. Time delays: (a) distribution of communication delay under interval [0,6T] and (b) average communication delay for N=10.

on the test-bench. The simulation results are validated due to similar behaviour obtained for the pressure in the clutch chamber. Using as input signal the same electric current i from the experiments made on the test-bench, Fig. 4b shows the comparison between measurements and simulations for the pressure obtained in the clutch chamber, along with the simulation results of the reduced pressure. Good agreement between the real-time and simulation results proves that the model captures the essential dynamics of the system. The behaviour of the simulated pressure from the piston chamber follows the measured pressure behaviour, having the same steady-state value, but with a difference in the high frequency due to the approximation of the very small time constants. These approximations result in a transient behaviour of the model a little bit different than that of the real response in the two phases. Thus, in the charging phase this difference appears as a time delay and in the discharging phase as a slow response. These modelling errors demand the use of an advanced control technique as the predictive approach, which is capable of compensating them.

The behaviour obtained for the valve displacement represented in Fig. 5a, is similar with the behaviour obtained in [8] where a linearised input–output model and a state-space model were developed for the electro-hydraulic pressure reducing valve used as actuator for the wet clutch. In the simulator of the actuator–clutch system, the clutch flow, which is illustrated in Fig. 5b, was obtained as a difference between the pressure from the valve and the clutch pressure. The simulation results obtained for the clutch piston displacements are illustrated in Fig. 5c, results obtained with the same input signal from Fig. 4a.

It can be seen that for a positive clutch flow, there are positive displacements both for the valve plunger and the clutch piston, illustrating the charging phase of the valve when the clutch chamber is filled with oil, while for a negative clutch flow, there are negative displacements, illustrating the discharging phase of the valve and the oil going from the clutch chamber through the valve to the tank.

The value obtained for the valve piston displacement is in the range of $[-1,+1]$ mm, like it is supposed to be, because the actuator is a closed loop system and the plunger displacement is restricted by the balance in forces. On the other hand, the clutch is an open loop system, with no feedback, so it results a high value of the piston displacement which can be further limited by applying a proper control strategy for the electro-hydraulic actuated wet clutch.

4.2. Controller performances

This section presents the validations of the proposed predictive control strategies investigated on the valve–clutch system model using the Matlab/Simulink programme.

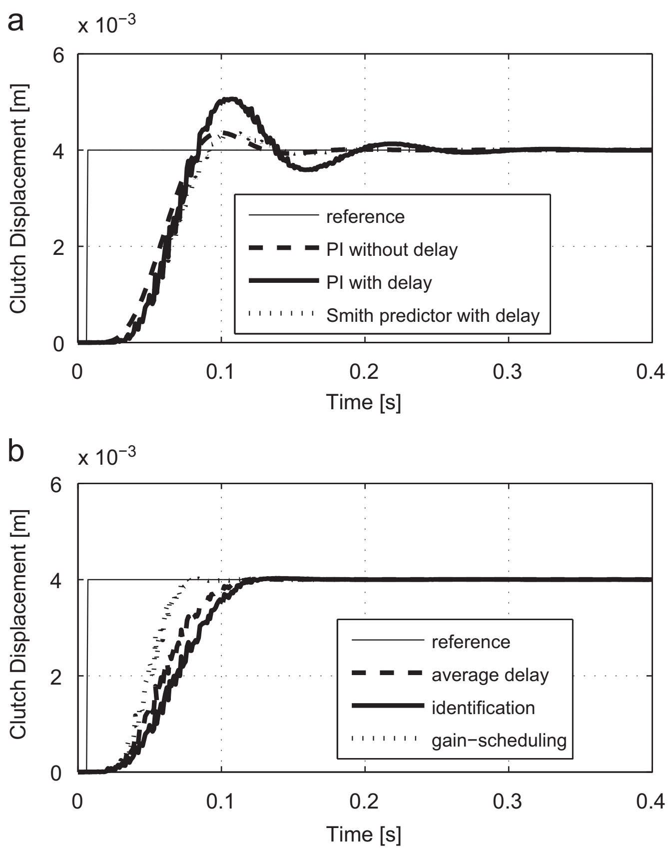

Fig. 7. Clutch displacements: (a) PI and Smith controllers and (b) predictive controllers.

Being components of the same powertrain subsystem it can be considered that the communication delays from sensor-to-controller and from controller-to-actuator have the same values and they are uniform distributed. In Fig. 6a, a time distribution of the communication delay (sensor-to-controller) under interval $[0, 6T]$, generated using a Matlab/Simulink model, is shown. Choosing N=10, in Fig. 6b the average value of N previous delays (13), used for the gain-scheduling method, is represented for the delays illustrated in Fig. 6a.

It was considered that d_m=0 and d_M=12 from (12) and the predictive control strategy with the three methods was applied using the following tuning parameters: $hc = na+1 = 3$, $hi = d+1$ and $hp = hc+d$. The control action obtained using the identified ARX model (11) was then applied to the model of the valve–clutch system.

The results obtained are compared with two different controllers: a PI controller and a Smith-like predictive controller with adaptation to communication delay developed in [27].

A step signal was applied as reference for the clutch piston displacement and it was desired that the system tracks the reference signal as fast as possible, the following figures showing the controlled outputs and the reference signal. In Fig. 7a, four signals were represented: the reference clutch displacement value, the response of the system using the PI controller without communication delay and with communication delay, the response of the system with delay when the Smith-like predictive controller is applied. Fig. 7b illustrates the reference clutch displacement value and the responses of the system with communication delay when the predictive average value method, the predictive identification method and the predictive gain-scheduling method are applied. The responses are clearly different from those obtained with the PI controller and the Smith predictor. The set-point response for the PI and the Smith-like predictive controllers have an obvious overshoot, while the set-point curves for the predictive methods are similar except that the rise time for the gain-scheduling method is much smaller than those for the other methods and all the responses have almost no overshoot.

In Fig. 8a, the control signals for the PI controllers and for the Smith predictor were represented and in Fig. 8b, the control signals for the predictive methods were represented. The results illustrate that the variations of the control signal are much smaller for the proposed identification method, while the variations are much bigger for the gain-scheduling method.

It can be concluded that the performances of the predictive control methods are better than the performances of the Smith predictor developed in [27].

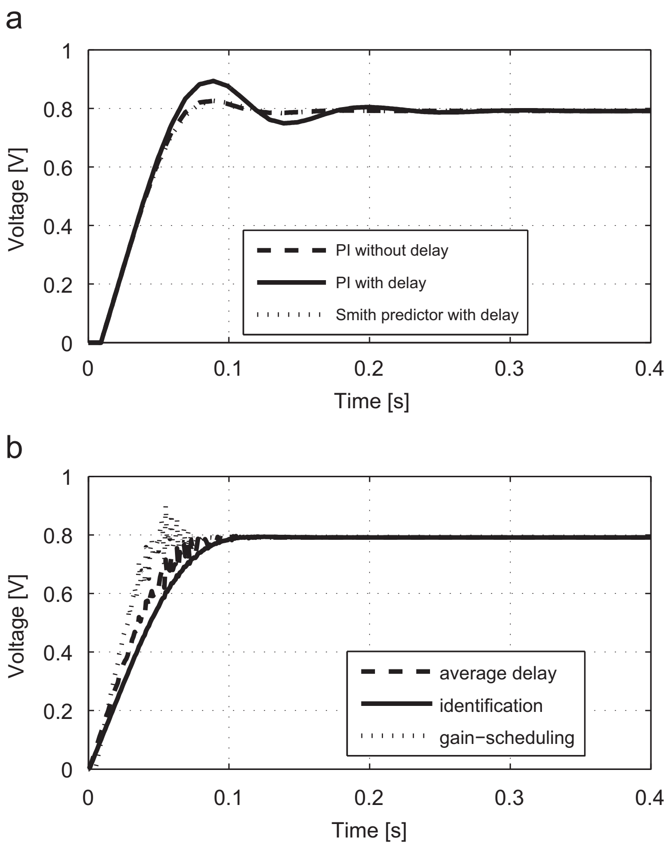

Fig. 8. Control signals: (a) PI and Smith controllers and (b) predictive controllers.

5. Conclusions

In this paper a linearised input–output model for an electro-hydraulic actuated clutch is developed. Simplifications were made in order to obtain a suitable transfer function to be implemented in Matlab/Simulink and to achieve an appropriate behaviour for the outputs. The model was validated by comparing the results with data obtained on a real test-bench provided by Continental Automotive Romania. Based on the validated input–output model, a networked predictive controller has been designed for the wet clutch actuated by an electro-hydraulic valve with the aim of controlling the clutch piston displacement, while decreasing the influence of the variable-time delays on the closed-loop control performances over the communication network. Simulations as well as the experimental results are presented to show the applicability of the modelling and control approaches.

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