MODELLING, IDENTIFICATION AND CONTROL OF ELECTRONIC THROTTLE USING DSPACE TOOLS

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Abstract

Electronic throttle consist of DC motor, spur gears, return spring, position sensor, power electronics and ECU. The position control of this electromechanical system is relatively difficult due to very high friction and strong nonlinearity of the spring. Simple application of linear control, such as PID, fails. This paper deals with the nonlinear control using principles of feedback linearization. The key component of controller structure is the friction compensator. In the paper, there is described modelling and parameter identification as well as the implementation details leading to satisfactory behavior. The controller designed in Simulink environment has been verified experimentally using dSpace RCP software/hardware.

1 Introduction

The current research and development in automotive industry is focused on X-by wire concept. The main idea is adopted from aircrafts, where “fly by wire” is nowadays the standard approach used in military as well as civil aircrafts. The conventional mechanical controls have been substituted by the computer control and the actuator is connected to the pilot only by means of a “wire”. As a result, the maneuverability of the aircraft increases significantly. Fly by wire is very often mentioned as a typical mechatronic solution.

Figure 1. Photo of throttle

In automotive design there are many variants of this philosophy in use already or in development, e.g.: steering by wire, where the conventional mechanical steering possibly with hydraulic booster is supposed to be replaced by sensor and electrical servomotor (currently used in some building machines); the brake by wire, where conventional hydraulic or pneumatic actuators are replaced by electrical ones; and also throttle by wire, which is the common part of many nowadays vehicles.

The throttle by wire consists of pedal sensor, throttle body (with DC motor, spur gears and potentiometer as position sensor) and electronic control unit (ECU). Thus, the conventional mechanical linkage of pedal and throttle via bowden cable is replaced by mechatronic design. As a result, the ECU can fully control the behaviour of throttle plate and therefore the better fuel economy and emissions can be achieved as well as the possibility of advanced tracking control algorithms and other overall system improvements.

Due to mass production of the automotive parts and corresponding compromises between technical quality and costs, there is significantly high friction in throttle mechanism. Moreover, the safety regulation requires the return of the valve to slightly open position (so called limp home, LH) in case of system failure. This feature is ensured by the relatively very strong spring near LH position. However, such spring stiffness in full valve range would also mean the enormous motor load and consequent energy consumption and heating. Therefore, the nonlinear spring is used in some throttles (Pavković et al. 2006, Deur et al. 2003).

As can be seen from literature survey and as we also experimentally proved, the linear control should fail in the task of electronic throttle control. The contribution of this paper is in the implementation of relatively simple but satisfactory functional control algorithm and its verification using rapid control prototyping (RCP) techniques.

The rest of paper is organized as follows: in Section 2 the complete electromechanical model with friction and nonlinear spring is described. Next in Section 3 the estimation of proposed model parameters is mentioned. Finally, the controller structure is discussed in Section 4 and after that practical results are presented.

2 Modelling of the electromechanical throttle

2.1 Description of the system

The Fig. 2 shows schematically the electromechanical throttle system. The DC motor with permanent magnets is linked to the throttle plate shaft by means of spur gearbox. The return spring with nonlinear static characteristic shown in Fig. 3 guarantees the minimal air flow through the valve in case of no supply voltage (LH position). The electronics used in our experimental setup includes H-bridge and the output voltage is set via analog value (0-5V). The position of throttle shaft is measured using potentiometer.

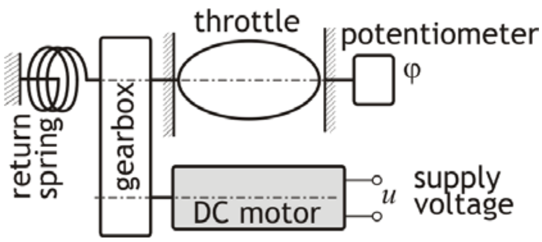

Figure 2. Schema of throttle

2.2 Model of DC motor

The standard well known model can be used for DC motor with permanent magnets:

$$u_{\mathrm{V}} = Ri + L\frac{\mathrm{d}i}{\mathrm{d}t} + k_{\mathrm{emf}}\dot{\varphi} \tag{1}$$

The dynamics of electrical subsystem can be neglected in our situation, so in steady state, the current is:

$$i = \frac{u_{\mathrm{V}}}{R} - \frac{k_{\mathrm{emf}}\dot{\varphi}}{R} \tag{2}$$

The electrical torque generated by motor recalculated to throttle shaft is of form:

$$m_e = i_{12}\eta_{12}k_{\mathrm{emf}}i = i_{12}\eta_{12}k_{\mathrm{emf}}\left(\frac{u_{\mathrm{V}}}{R} - \frac{k_{\mathrm{emf}}\dot{\varphi}}{R}\right) = i_{12}\eta_{12}k_{\mathrm{emf}}\frac{1}{R}u_{\mathrm{V}} - i_{12}\eta_{12}k_{\mathrm{emf}}{}^2\frac{1}{R}\dot{\varphi} = k_{\mathrm{M1}}u_{\mathrm{V}} - k_{\mathrm{M2}}\dot{\varphi} \tag{3}$$

2.3 Nonlinear spring

The spring torque applied to shaft can be generally expressed in form:

$$M_{\mathrm{K}} = k_{\mathrm{S}}(\varphi)\varphi \tag{4}$$

The stiffness k_S varies according to throttle position as shown in Fig. 3. From the implementation point of view, it is important to guarantee the smooth switching between particular spring domains. In Simulink, one is supposed to use any block with zero crossing detection feature (e.g. Lookup table block cannot be used). In our implementation, we used several blocks Switch.

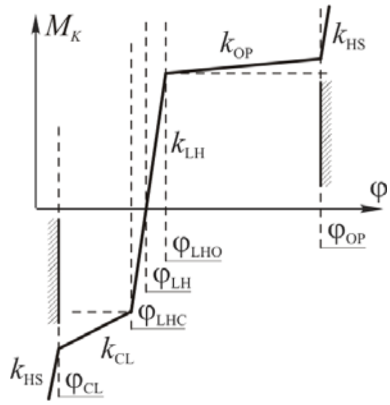

Figure 3. Schema of nonlinear spring (LH – limp home, CL – throttle closed, OP – throttle opened)

2.4 Friction model

In the model, the combination of viscous damping and the dry (Coulomb) friction is considered. Parameters of both friction types are assumed to be constant.

$$M_{\mathrm{b}}(\dot{\varphi}) = -b\dot{\varphi} \tag{6}$$

$$M_{\mathrm{C}}(\dot{\varphi}) = -\mathrm{sgn}(\dot{\varphi})\mu N \tag{7}$$

2.5 Equation of motion and simulation models

When considering previous equations (3)-(6), the equation of motion of the electromechanical system can be written in form:

$$J\ddot{\varphi} = k_{\mathrm{M1}}u_{\mathrm{V}} - k_{\mathrm{M2}}\dot{\varphi} - b\dot{\varphi} - M_{\mathrm{K}}\left(\varphi\right) - \mathrm{sgn}(\dot{\varphi})\mu N \tag{8}$$

Further we assume that there is no prior knowledge about system parameters and we express all the torques in uniform normalized units of signal u scaled to range (-1;1). Also, the rotation of throttle is expressed in measured volts. Finally, the equation of motion is of the form:

$$J^{*}\ddot{\varphi} = u - B\dot{\varphi} - M_{\mathrm{K}}^{*}\left(\varphi\right) - \mathrm{sgn}(\dot{\varphi})T \tag{9}$$

As soon as all acting torques are expressed, the simulation model can be built. Two different models have been prepared and tested.

First, the model in Matlab/SimMechanics environment has allowed the relatively easy implementation of the static friction model with very good numerical stability (using Joint Stiction Actuator). We have used this model for parameter identification (described below). However, there are serious practical reasons which disqualifies the SimMechanics model for the RCP implementation: 1) the used version of SimMechanics (we used Matlab 2006a) does not work with RTI and cannot be compiled for our dSpace target; 2) even if there would be possibility to compile the model, the main features of iteration locking and unlocking algorithm will not be working due to constant step size of ODE solver; and finally 3) the SimMechanics model will be comparatively more demanding and thus the requirements given on target ECU would be much higher.

Therefore, the second type of model has been built in the pure Simulink using standard approach. The previous SimMechanics model has been used as the reference during the critical tuning of friction model parameters. As a result, we have the model which can be used for RCP and behaves identically as the reference SimMechanics one.

3 Identification of system parameters

3.1 RCP hardware and software

The domain of real time simulation with interactions with "real world" can be divided into two main parts: the HIL (Hardware In the Loop), where the real control hardware, such as DSP, is connected to computer simulation of plant; and the RCP (Rapid Control Prototyping), where the real plant is controlled by simulating model. Many important issues are related to this area, like effective simulation models, automatical code generation, fast simulation on PC with I/O communications versus real time targets and others.

For our RCP, we have used the setup based on Matlab/Simulink software and dSpace hardware. Particularly, RealTime Workshop generates the C code for PPC target and RTI (Real-Time Interface) provides link between code and dSpace hardware. Modular dSpace hardware we used consists of AutoBox with processor board for realtime computation DS 1005 PPC, 14 bit D/A converters DS 2103, 16 bit A/D converters DS 2003 and communication cards DS 814 and 815.

The power electronics for throttle control has input analog signal 0-5V (2.5 V central position=zero voltage to DC motor) and output analog signal in range approx. 0.25 – 2.25 V which gives the information about the throttle position (2.25 V full open).

3.2 Static parameters identification

Fig. 4 shows the typical response (angle voltage) of the throttle to sinusoidal input (control signal). If the frequence is low, the response can be considered as static.

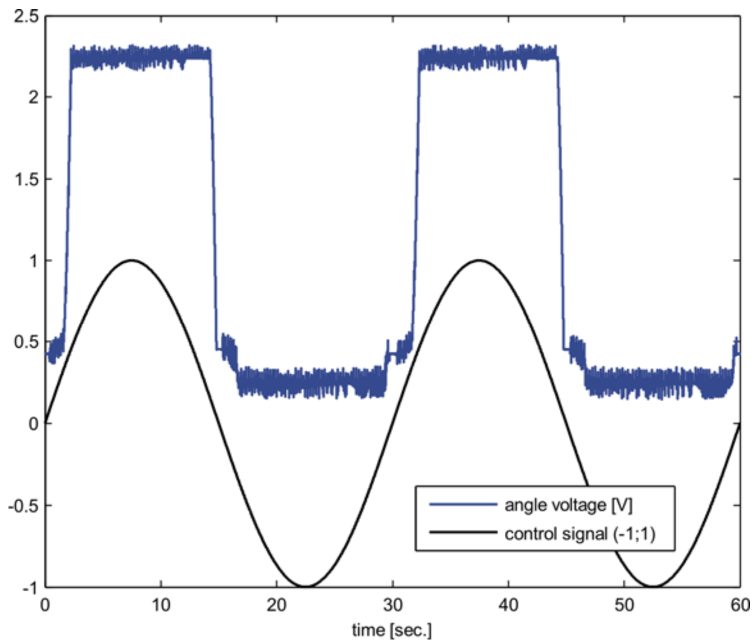

Figure 4. Slow sinus input and response of the system

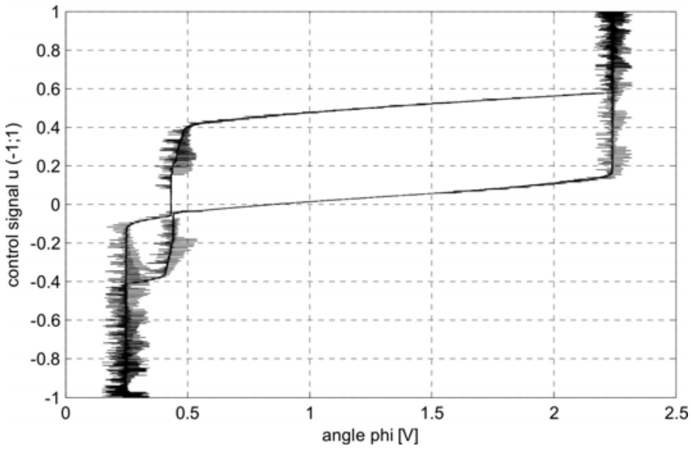

Figure 5. Static $\varphi - u$ characteristic measured

The response plotted in $\varphi - u$ characteristic is shown in Fig. 5. Important facts can be obtained: 1) the LH position, close and open position of the valve can be read; 2) the friction in the system is substantial; 3) the system behaviour is repeatable (many sin cycles produce one static characteristic); 4) the LH spring stiffness is relatively high; 5) the stiffness in opening regime is significant (in (Pavković et al. 2006), it is neglected).

Several experiments with different frequency of sinusoidal input and its amplitude have been performed. After that, static parameters (friction and spring coeficients) of SimMechanics simulation model have been identified. Theoretically, the amplitude and frequency (if low) cannot influence the static characteristic. However, one can easily see in Fig. 6, that responses are very different (although stable). Despite of it, we have chosen one set of static parameters as an estimate for model and use it in controller compensator.

The alternative approach would deal with dynamic friction modelling and probably with other adaptations; or with any of grey-box modelling approaches.

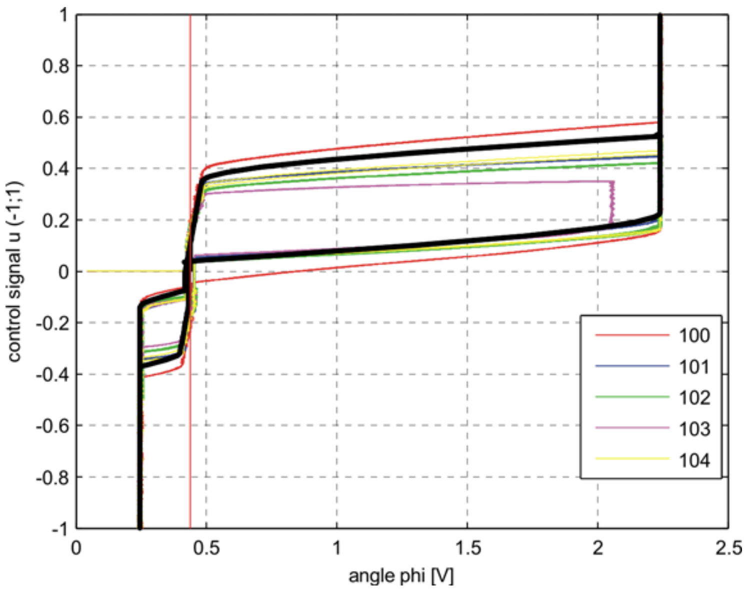

Figure 6. Static $\varphi - u$ characteristic measured and filtered (colored) and identified model (black)

3.3 Dynamic parameters identification

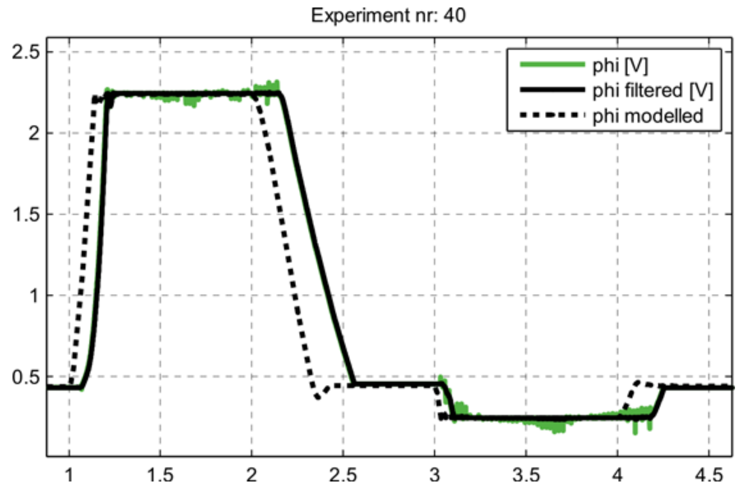

Figure 7. Response on input step

Further, two dynamic parameters J and b must be determined. We have used the step response of the system shown in Fig. 7. One can see that there is still some unmodelled dynamics causing time delay in the system.

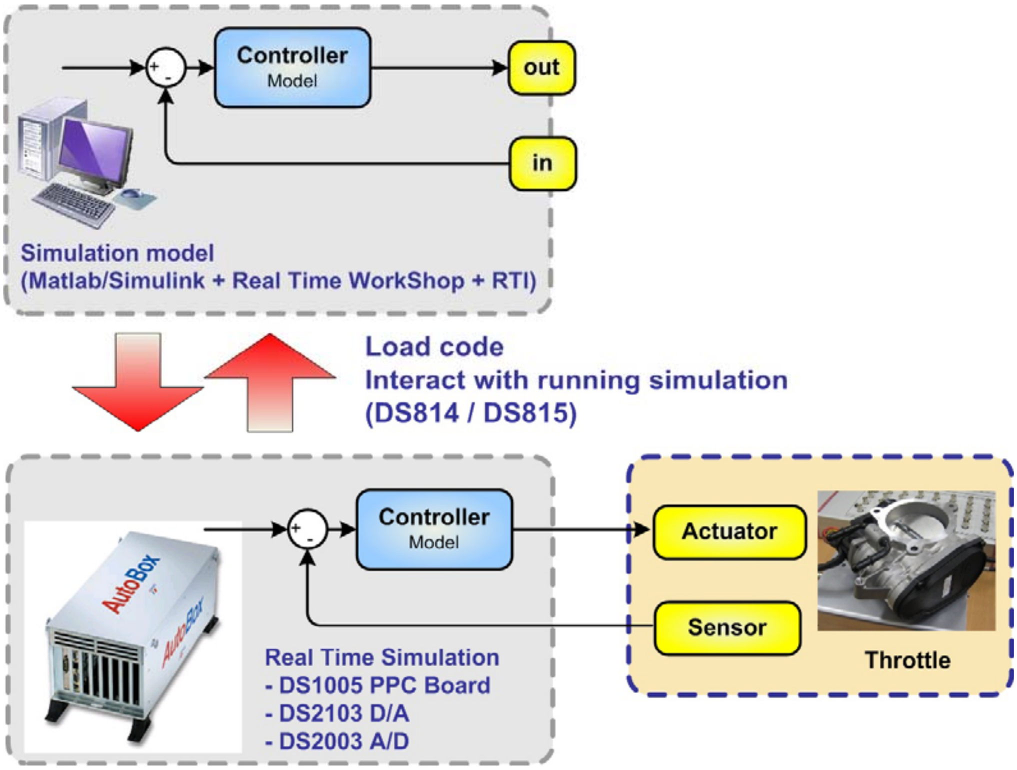

Figure 8: Principal schematic of RCP

4 Controller design

4.1 Feedback linearization

As we find out during identification, the throttle system is highly nonlinear with dominant dry friction and strong spring nonlinearity. Thus, the simple application of standard PID controller cannot be used successfully. The nonlinear control theory offers the well known method called feedback linearization, which first transfer the nonlinear system into the linear one using additional terms based on model of system and next design the control for this linearized system using standard linear tools. Note here, that this feedback linearization is global (compare to linearization around the working point based on differential mathematics) and requires real time computation of inverse dynamics of the system.

For our nonlinear model (9) rewritten in following form (without hard-stops)

$$J^{*}\ddot{\varphi} + B\dot{\varphi} + M_{\mathrm{K}}^{*}\left(\varphi\right) + \mathrm{sgn}(\dot{\varphi})T = u \tag{10}$$

can be the system input redefined as follows:

$$u = z + M_{\mathrm{K}}^{*}\left(\varphi\right) + \mathrm{sgn}(\dot{\varphi})T \tag{11}$$

where z is the new system input. If eq. (11) is substituted into (10), we obtain

$$J^{*}\ddot{\varphi} + B\dot{\varphi} = z \tag{12}$$

which is second order linear system.

Then we can design linear controller (PID) for system (12) and use eq. (11) to compute necessary compensation. General schema of the controller structure is shown in Fig. 9.

In further text of this section there are described friction and spring nonlinear compensators as well as a few small but practically very important implementation details.

4.2 Friction compensator

In the computer model described in Section 2, there is used the velocity $\dot{\varphi}$ for the friction determination. However, this would be rather impractical in the case of compensation. As the only angle is measurable, the angular velocity must be obtained via numerical derivation (problematical) or via some observer (difficult to guarantee stability for nonlinear system). Subsequently, the chattering of friction compensation occurs. Another and much practical approach consists of the use of position error instead. The friction force applied in the compensator is then of form:

$$M_{\mathrm{C}}(\varepsilon) = -\mathrm{sgn}(\varepsilon)T \tag{13}$$

Compare this approach equation to (7). The explanation is simple enough: if the position error is positive, than the velocity must be also positive and serious problems with velocity measurement are solved.

Further, two practical detail must be used to guarantee the satisfactory behaviour of compensator. First, the small dead zone of ε must be used for the sign determination (experimentally found value of 0.01 works fine). Second, the resulting friction force must be filtered. We used the first order filter with experimentally set time constant of value 70ms.

4.3 Spring nonlinearity compensator

The spring compensator is identical with the spring model described in 2.3. Since the simulation model for code generation must used fixed step solver, the Zero crossing detection feature is disabled.

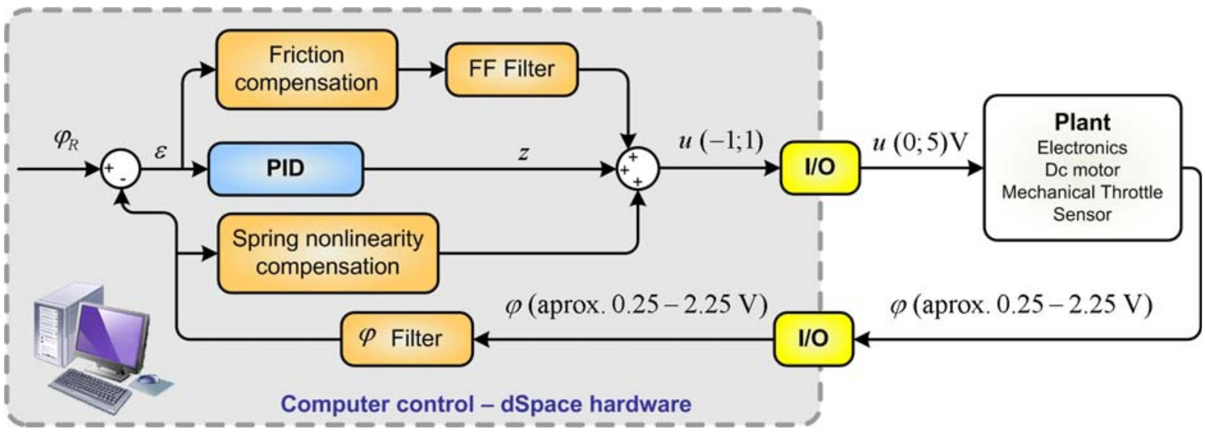

Figure 9. Schema of nonlinear controller

4.4 PID regulator

The parameters of PID regulator have been tuned experimentally. The P value together with D has been increased until the control signal saturation. Integral component helps to correct imprecissions in spring compensator and works fine without the necessity to employ the gain schedulling described in (Pavković et al. 2006). Resulting best values of PID are: Kp = 0.84; Ki = 0.64; Kd = 0.1575 for particular throttle valve. Resulting simulation model is shown in Fig. 13 and 14.

5 Results

The controller described in previous section has been implemented and sucessfully tested. Fig. 10 shows the random step desired position of valve and response of controlled system. For comparison, Fig. 11 shows the results with no friction force filtering in compensator. Fig. 12 shows the return after the manual disturbance.

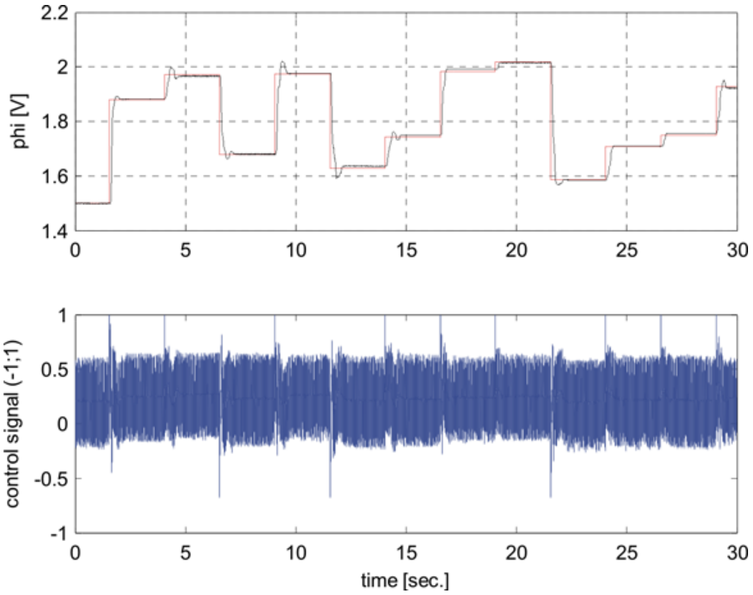

Figure 10. Response of controlled system to random desired step values

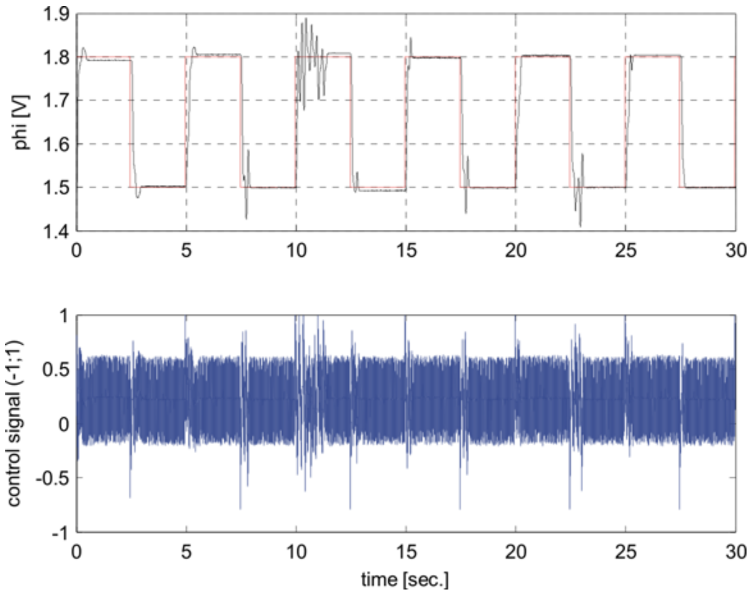

Figure 11. Response of controlled system with no friction compensation

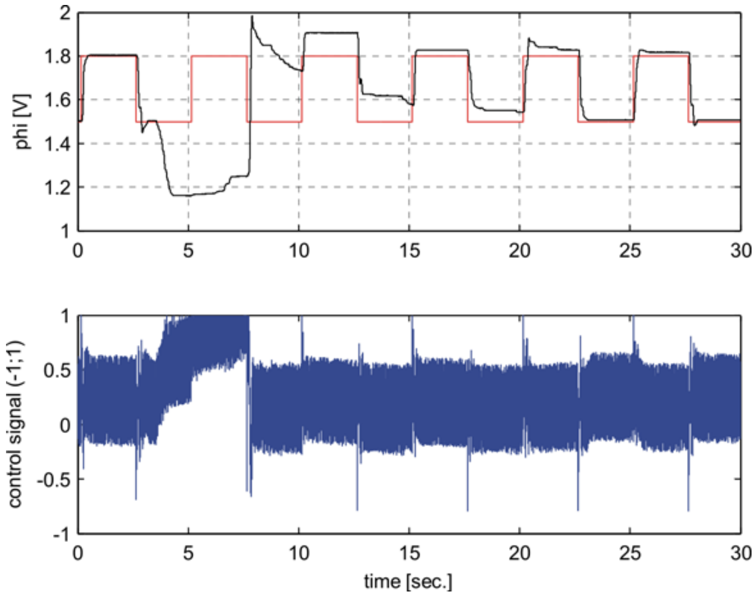

Figure 12. Response of controlled system after the manual disturbance of valve position

6 Conclusion

In this paper we have presented the complete, practicle and comparatively simple method for the design of the satisfactory nonlinear electronic throttle control. The electromechanical model of throttle with numerically stable static friction and nonlinear spring has been built in the SimMechanics environment. The Joint Stiction Actuator allows precise modelling of friction also at zero velocity. Next, the identification of system parameters has been performed based on series of measurements of the real system response using RCP software/hardware and quasistatic and dynamic inputs to the system. After that, the proposed nonlinear controller has been extensively tested on computer non-real time model with appropriate added noise of sensors. Finally, the controller has been implemented in Simulink for RCP hardware. The key component is the friction compensator based on measured position error. The computed compensating friction force must be filtered. Also the small dead zone applied for position error must be used to guarantee the smooth behavior without oscillations. The second important part of the controller is the nonlinear spring compensator. At the end of experiments, the step size for the integrator (ODE solver) has been decreased with satisfactory controller behaviour to reasonable value of 0.001 s which allows implementation on standard DSP.

The contribution of this paper is in the methodological domain (two described simulation models for the fast identification and development of compensators) and also in several small but practically important implementation details (dead zone in friction compensator, filtering). Most of described results are generally applicable for wide class of system with the presence of high Coulomb friction and possibly other strong nonlinearities.

Nomenclature

i_{12} total gear ration from motor to throttle shaft
η_{12} total efficiency of mechanical system
k_{emf} constant of back electromotive force
R armature resistance
m_e electrical torque recalculated to throttle shaft
φ angle of throttle shaft
i electrical current
k_S stiffness of the return spring
M_C friction torque
ε tracking error
T friction torque in normalized units
J^* inertial moment in normalized units
B viscous damping (friction) in normalized units
z new system input
μ dry friction coefficient
N normal force

References

[1] D. E. Newland. *An Introduction to Random Vibrations, Spectral and Wavelet Analysis*. Longman Scientific & Technical, Essex, U.K., third edition, 1994.

[2] Baotic, M., Vasak, M., Morari, M. & Peric, N. (2003) *Hybrid system theory based optimal control of an electronic throttle*, American Control Conference

[3] Contreras, A. F., Quiroz, I. P. & Wit, C. C. (2002). *Further Results on Modelling and Identification of an Electronic Throttle Body*, Proceedings of the 10th Mediterranean Conference on Control and Automation - MED2002, Lisbon, Portugal, July 9-12.

[4] Deur, J., Pavković, D., Jansz, M. & Perić, N. (2003). *Automatic Tuning of Electronic Throttle Control Strategy*, Proceedings of the 11th Mediterranean Conference on Control and Automation MED 2003, Rhodes, Greece, June 18-20.

[5] Hadilebbal, M., Chafouk, H., Hoblos, G. & Lefebvre, D. (2007). *Modeling and identification of non-linear systems by a multimodel approach: application to a throttle valve*, International Journal of Information and Systems Sciences, Volume 3, Number 1, pp. 79-99

[6] Olsson, H. & Astrom, K.J. (1997). *Friction Models and Friction Compensation*, http://www-lag.ensieg.inpg.fr/canudas/

[7] Pavković, D.,Deur, J., Jansz, M. & Perić, N. (2006). *Adaptive Control of Automotive Electronic Throttle*, Control Engineering Practice, 14, pp. 121 – 136.

[8] Slotine, J. & Li, W. (1991). *Applied Nonlinear Control*, Prentice-Hall Engineering/Science/Mathematics

[9] Trebi-Ollennu, A. & Dolan, J. M. (2004). *Adaptive Fuzzy Throttle control for an All Terrain Vehicle*, Institute for Complex Engineered Systems, Carnegie Mellon University, Internal Report 2004

[10] Vašak, M., Baotić, M., Petrović, I. & Perić, N. (2007). *Hybrid Theory-Based Time-Optimal Control of an Electronic Throttle*, IEEE Transactions on Industrial Electronics, 54, pp. 1483-1494.

[11] Yang, C. (2004). *Model-Based Analysis and Tuning of Electronic Throttle Controllers*, 2004 SAE World Congress, Detroit, Michigan, March 8-11

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