Control of an Electronic Throttle Valve for Drive-by-Wire Applications

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Abstract Electrically actuated control devices for regulating the amount of air entering gasoline engines play an essential role in drive-by-wire applications. In this paper an approach to the control of so-called electronic throttle valves is outlined. First a standard sliding-mode controller is presented. It is shown that the performance of the feedback loop can be improved significantly by incorporating time-variable boundary layers.

1 Introduction

The continuously stringent emission and fuel economy regulation implemented by governments motivates the development of drive-by-wire systems in automobiles with combustion engines. Besides modern cars often are equipped with driver-assistance systems such as cruise control, collision warning systems or driver impairment monitoring. Many of them are based upon drive-by-wire concepts as well. A major component of these systems is the so-called electronic throttle valve which eliminates the mechanical link between the accelerator pedal and the throttle valve (see Fig. 1). In electronic

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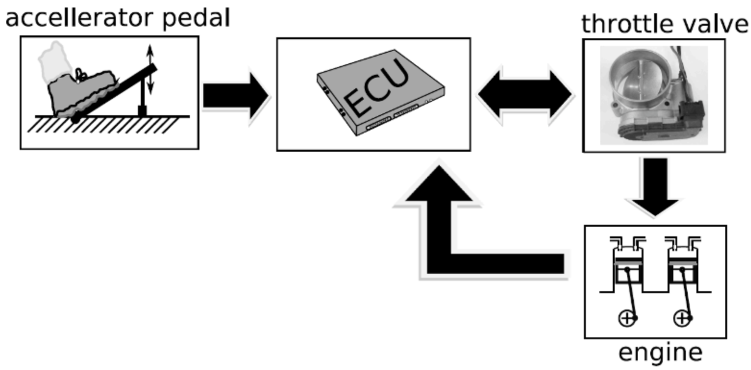


Fig. 1 Principle of a drive-by-wire system.

throttle valves the valve-plate is actuated by a dc-drive via a gear unit. A resistive position sensor measuring the opening angle φ of the valve is integrated into the gear unit. The computation of a suitable reference angle φ^* is managed by the engine control unit (ECU) which is provided with relevant engine data. In the case of an electronic failure two counteracting springs reposition the valve-plate into the so-called limp-home angle φ_0 so that an emergency operation is gurarantееd.

In the recent years a number of publications has been dedicated to the modeling, simulation and control of electronic throttle valves. In [1] and [2] a detailed mathematical model of a throttle valve is presented. Experimental procedures to identify unknown model parameters are described as well. In [3] a dynamic friction model and approaches to compensate for fricition phenomena are proposed and practically demonstrated. In [4] an output feedback LQR for a throttle valve and a wastegate system is designed. Due to its undesirable high order the controller is reduced to a classical PI-structure. A control scheme consisting of two LQG based approaches is presented in [5]. A drive-by-wire throttle control in combination with a sliding-mode concept is studied in [6]. The proposed anticipatory band method is compared to conventional bang-bang and PI strategies. A classical sliding-mode technique including a full-state observer is outlined in [7], in [8] a discrete-time version of the control law can be found. In [9] the sliding-mode controller is based on a feedback linearized model of the throttle valve. The application of second order sliding-mode strategies to a throttle valve is presented in [11].

The paper is organized as follows: In Section 2 a mathematical model for the throttle valve system is presented. Section 3 outlines the principles of the proposed control strategy. Section 4 shows simulation results whereas Section 5 is dedicated to experimental findings. Section 6 concludes the work.

2 Mathematical Model

The model used in this work is sufficiently precise to describe the motion of the valve-plate [11] and serves as a basis for the controller design task. The dynamic behaviour of the opening angle φ can be modeled by the differential equation

$$\frac{d^2\varphi}{dt^2} = -f_{sp} - f_{fr} + f_{mot} \tag{1}$$

where the right-hand side functions

$$f_{sp} = c_0 + c_1\left(\varphi - \varphi_0\right), \tag{2}$$

$$f_{fr} = k_v \frac{d\varphi}{dt} + k_c\, sign\left(\frac{d\varphi}{dt}\right) \quad \text{and} \tag{3}$$

$$f_{mot} = k_m\, u - k_t \frac{d\varphi}{dt} \tag{4}$$

represent the nonlinear spring characteristic, the friction phenomena and the impact of the dc-motor respectively. The parameters c_0, c_1, k_v, k_c, k_m and k_t are positive constants. The control signal u is a pulse-width modulated signal. Note that in the model made up by equations (1), (2), (3) and (4) phenomena like static friction, gear-backlash as well as the dc-motor dynamics are neglected.

The task of the controller design is to make the opening angle φ track a reference angle φ^* which is generated by the ECU. This motivates the introduction of the tracking error ϵ_1 and its time derivative ϵ_2 as

$$\epsilon_1 := \varphi - \varphi^* \quad \text{and} \quad \epsilon_2 := \frac{d\epsilon_1}{dt}. \tag{5}$$

The representation of system (1) using the variables defined in (5) is given by the two first order differential equations

$$\begin{aligned} \frac{d\epsilon_1}{dt} &= \epsilon_2 \\ \frac{d\epsilon_2}{dt} &= -c_0 - c_1\left(\epsilon_1 + \varphi^* - \varphi_0\right) - k\left(\epsilon_2 + \frac{d\varphi^*}{dt}\right) - \\ &\quad - k_c\, sign\left(\epsilon_2 + \frac{d\varphi^*}{dt}\right) - \frac{d^2\varphi^*}{dt^2} + k_m\, u, \end{aligned} \tag{6}$$

where $k = k_v + k_t$. The parameters of model (6) can easily be identified by experiments described in [2]. The nominal values as well as the bounds of the parameters are summarized in Table 1. Due to its discontinuous right-hand side, uncertain model parameters and imperfections in modelling the control of the throttle valve system is a challenging task. A promising approach to

Table 1 Estimated plant parameters for system (6) with bounds and nominal value.

parameter	value		
	minimal	nominal	maximal
c_0	110	150	195
c_1	70	80	90
k	90	100	110
k_c	33	66	104
k_m	110	120	130
φ_0	0.095	0.095	0.095

address the control problem is the application of a variable structured control law, often referred to as sliding-mode control.

3 Control

The desired dynamic of the tracking error [12, 13] is specified by the differential equation

$$\frac{d\,\epsilon_1}{d\,t} = -\lambda\epsilon_1 \quad \text{with} \quad \lambda > 0. \tag{7}$$

The structure of model (6) motivates the choice of the so-called switching function

$$\sigma(t) := \epsilon_2 + \lambda\epsilon_1. \tag{8}$$

It is the task of the controller to steer an arbitrary initial value $\sigma_0 := \sigma(0)$ to $\sigma = 0$ within *finite* time. The stabilization of $\sigma = 0$ is solved with the help of the so-called equivalent control method [14]. Thereby it is assumed that the surface $\sigma(t) = 0$ has already been reached. To remain there the required control signal u_{eq} for system (6), parametrized with nominal plant parameters, can be calculated from

$$\frac{d\,\sigma}{d\,t} \stackrel{!}{=} 0. \tag{9}$$

The resulting equivalent control signal is expressed as

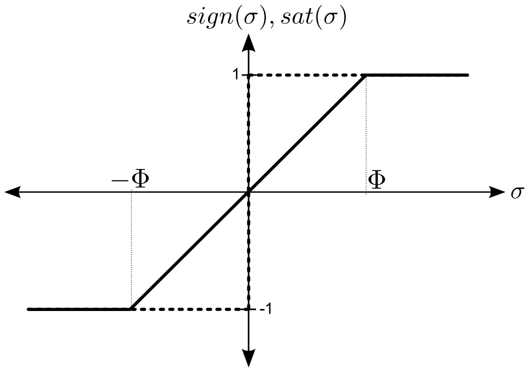
$$u_{eq} = \frac{1}{k_m}\left[c_0 + c_1\,(\varphi - \varphi_0) + k\frac{d\,\varphi}{d\,t} + k_c\,sign\left(\frac{d\,\varphi}{d\,t}\right) + \frac{d^2\,\varphi^*}{d\,t^2} - \lambda\epsilon_2\right]. \tag{10}$$

A discontinuous part u_{disc} is added to the above control law, i.e.

$$u = u_{eq} + u_{disc} = u_{eq} - \frac{\kappa}{k_m} sign(\sigma) \qquad \text{with } \kappa > 0 \tag{11}$$

to guarantee finite time convergence of σ. The parameter κ is chosen sufficiently large to reach $\sigma = 0$ even in the case of uncertain parameters. An implementation of control law (11) requires high switching frequency in the control signal u. This evokes the so-called chattering phenomenon [12]. A well known countermeasure is to approximate the discontinuity by a saturation function [15], see Fig. 2. Thereby a so-called boundary layer of width Φ is introduced around $\sigma = 0$.

Fig. 2 The discontinuous *sign*-function is approximated by a *sat*-function.



4 Numerical Simulation

The control law (11) is realized in Matlab/Simulink. The simulations are based on a detailed mathematical model, which includes gear-backlash, static friction, mechanical stops and as well the dc-motor dynamics. The simulations are carried out with the reference signal φ^* shown in Fig. 3. The reference signal covers almost the whole range of operation and comprises angles above and below the limp home angle φ_0. Also high angular velocities and discontinuities are covered. The initial controller parameters satisfying sufficient conditions for robust stability [13] were tuned online. A simulation result for the parameters $\kappa = 600$, $\lambda = 55$ and $\Phi = 0.3$ is depicted in Fig. 4. The feedback loop shows excellent steady state behaviour. The absolute value of the tracking error remains below $1°$ during the entire simulation which is very satisfactory. At time instants of high angular velocities in the reference signal the control signal contains undesired high frequency components which induce mechanical vibrations. The challenge of the experimental parameter tuning is to obtain a closed loop behaviour with satisfactory tracking performance and a chatter-free control signal u.

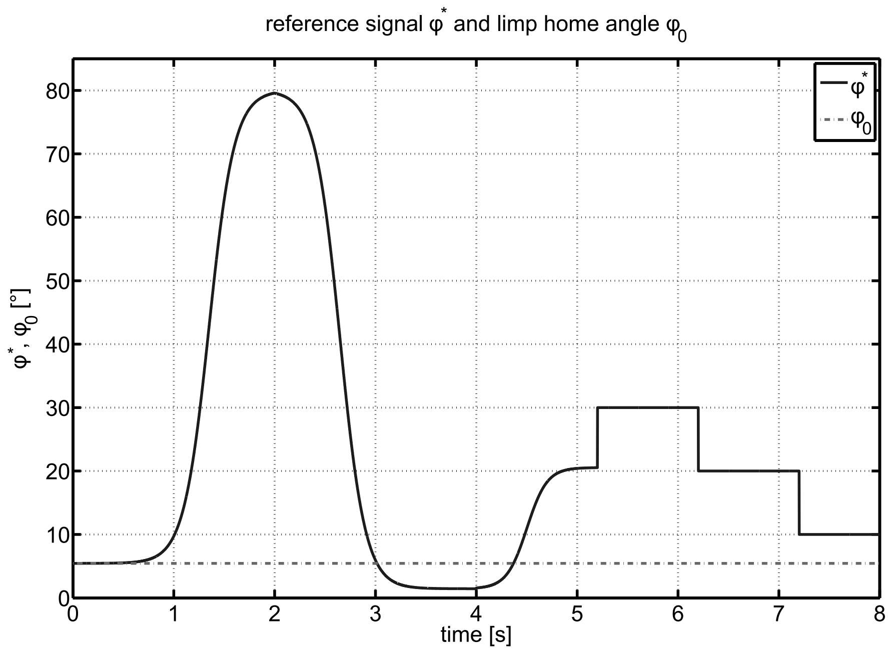


Fig. 3 The reference signal φ^* consists of a smooth and a non-smooth part.

5 Experiment

The experimental hardware consists of a dSpace Microautobox, a H-Bridge power amplification circuit and the electronic throttle valve system. The present control law is executed with a sampling time of $T_s = 1ms$. The implementation of control law (11) is straight forward. The second time derivative of the reference signal φ^* is computed with the help of a DT_2-element. Fig. 5 shows experimental results achieved with controller parameters obtained from simulation. Solely the width Φ of the boundary layer was modified to $\Phi = 0.5$. The feedback loop shows satisfactory tracking performance. The control signal u reveals the undesired chattering-effect. A way to minimize the chattering-effect is to increase the width of the boundary layer. In Fig. 6 results for $\Phi = 1.5$ are depicted. The control signal u shows a negligible chattering-effect at the cost of an unacceptable steady state error. These results motivate the choice of a time varying boundary layer [15]. As observed, the chattering-effect is proportional to the angular velocity of the valve-plate. This fact suggests the choice [10]

$$\Phi = \Phi_0 + \left|\frac{d\varphi}{dt}\right| \quad \text{with} \quad \Phi_0 > 0 \tag{12}$$

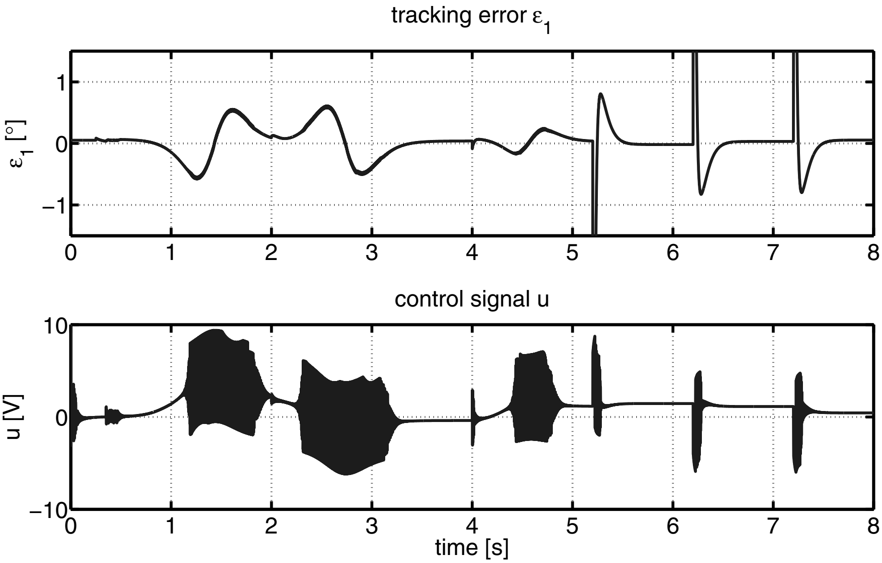


Fig. 4 Simulation results for $\kappa = 600$, $\lambda = 55$ and $\Phi = 0.3$.

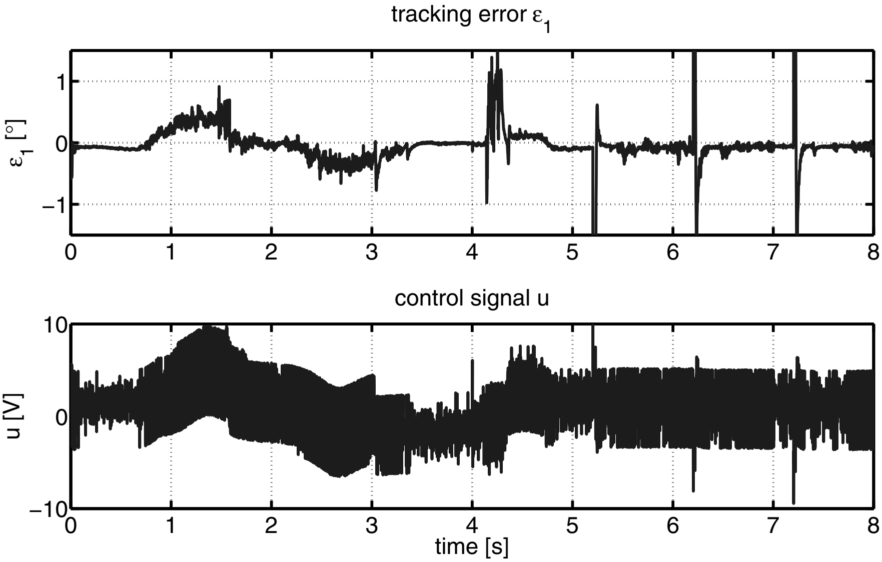


Fig. 5 Experiment: Constant boundary layer width of $\Phi = 0.5$.

for the boundary layer. As a consequence of the above definition the conditions for robust stability have to be adapted appropriately [15]. Results of an experiment with $\Phi_0 = 0.3$ are plottet in Fig. 7. The steady state behaviour is excellent and the chattering-effect in the control signal u is sufficently

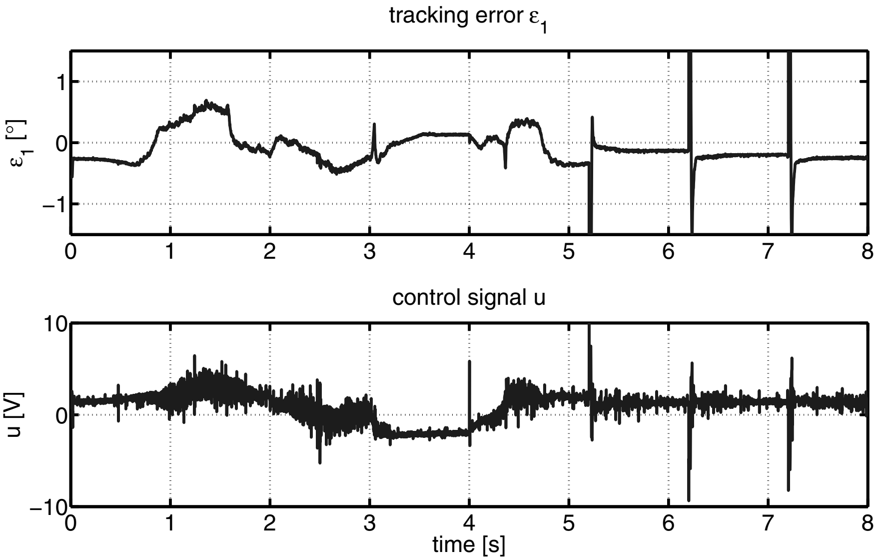


Fig. 6 Experiment: Constant boundary layer width of $\Phi = 1.5$.

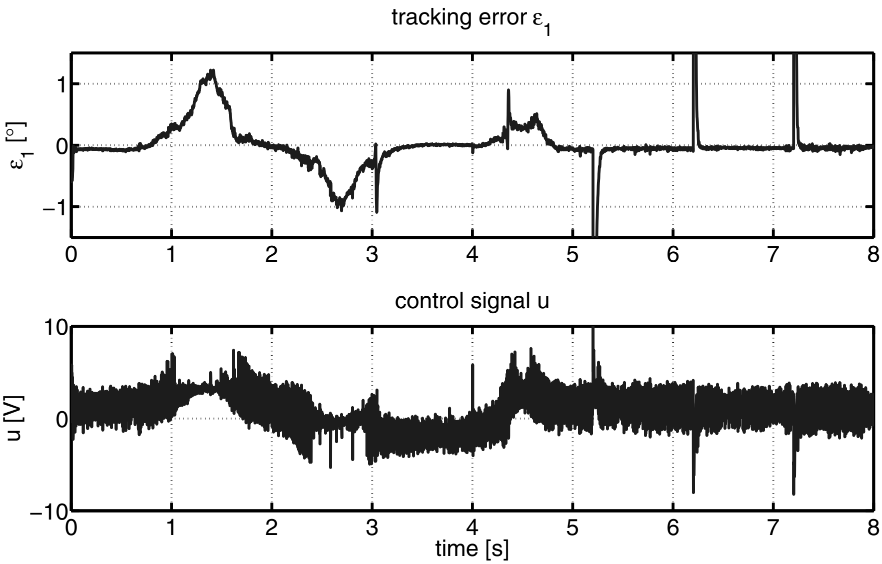


Fig. 7 Experiment: time varying boundary-layer

reduced. The corresponding width Φ of the boundary layer during the experiment is outlined in Fig. 8.

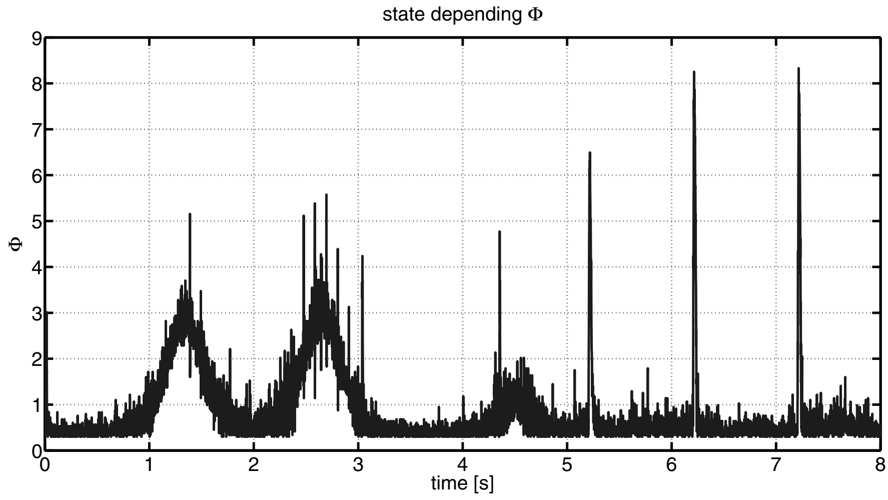


Fig. 8 The thickness of the boundary-layer is varying during the experiment.

6 Conclusion

The scope of the paper is the evolution of a control strategy for electronic throttle valves based on the concepts of variable structure systems. Based upon a standard sliding-mode controller an improved strategy is derived by employing time-variable boundary layers. It is demonstrated that the resulting feedback loop shows excellent tracking performance in simulation and experiment. The implementation of the control law is straight forward and hence suitable for a realization in the engine control unit.

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