Al-Khwarizmi Engineering Journal,Vol. 11, No. 3, P.P. 72- 84 (2015)

Al-Khwarizmi Engineering Journal

Integral Sliding Mode Control Design for Electronic Throttle Valve System

Shibly Ahmed AL-Samarraie* **Alaq Sabah Badri****
Mustafa H. Mishary***

,,***Department of Control and Systems Engineering/ University of Technology*
*Email: dr.shiblyahmed@yahoo.com
**Email: alaqsabah@yahoo.com
***Email: albmustafa@yahoo.com

(Received 13 August 2014 ; accepted 18 May 2015)

Abstract

One of the major components in an automobile engine is the throttle valve part. It is used to keep up with emissions and fuel efficiency low. Design a control system to the throttle valve is newly common requirement trend in automotive technology. The non-smoothness nonlinearity in throttle valve model are due to the friction model and the nonlinear spring, the uncertainty in system parameters and non-satisfying the matching condition are the main obstacles when designing a throttle plate controller.

In this work, the theory of the Integral Sliding Mode Control (ISMC) is utilized to design a robust controller for the Electronic Throttle Valve (ETV) system. From the first instant, the electronic throttle valve dynamics is represented by the nominal system model, this model is not affected by system parameters uncertainty and the non-smooth nonlinearities. This is a consequence of applying the integral sliding mode control. The ISMC consists of two part; the first is the nominal control which is used to control the nominal system, while the second is a discontinuous part which is used to eliminate the effects of the parameters uncertainty and the non-smooth nonlinearities from system model. These features for the ISMC are proved mathematically and demonstrated numerically via seven numerical simulations and for different desired trajectories. The simulation results clarify that for different system parameters, the ETV behaves as a nominal system. This enables to freely and precisely select the system response characteristics and the time required for the throttle angle to reach the desired value. Moreover the ability to deal with the chattering problem is demonstrated through the worked simulation tests, where the chattering is eliminated via approximating the signum function by arc tan function.

***Keywords**: Electronic throttle valve, Nonlinear spring model, Integral sliding mode control, Non-smooth model.*

1. Introduction

One of the key components in an automobile engine is the throttle valve part. Throttle valve consists of a throttle plate, a motor causes the plate to rotate in angular position and return spring. The joint between the motor and the throttle plate is considered rigid. The opening angle of a throttle plate allows the air to flow to the engine. Its function is to adjust the air–fuel ratio during combustion by changing the opening angle of the valve plate, through which there exists airflow. Engine efficiency and emissions are affected by the control of the throttle plate, in particular its angular position. Depending on the current engine load, the angular position must track a trajectory as determined by the accelerator [1]. The electronic throttling valve is shown in Figure (1).

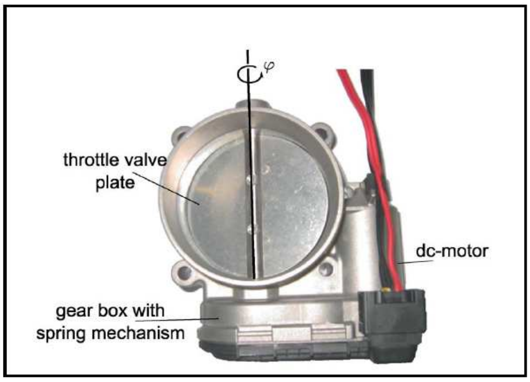

Fig. 1. Electronic throttle valve [2].

The electronic throttle valve is an electromechanical system with a DC motor as the actuating torque provider. Due to the presence of the non-smooth nonlinearities in the electronic throttle valve system model and the uncertainty in system model parameters the objective of designing a successful controller is a challenging problem. Moreover these nonlinearities and the uncertainty in system model do not lie in the control input channel which adds another problem named as the matching condition.

Many authors used the sliding mode control theory for controlling the throttle control system. All of them overcome the main problem (model nonlinearity) using sliding mode control theory as a robust tool with respect to uncertainty and nonlinearities in system model. Due to the existence of nonlinearities including stick–slip friction, backlash, and a discontinuous nonlinear spring, the sliding mode control theory is utilized to design a robust controller that will force the throttle angle to follow the desired trajectory [3].

Mercorelli [1] show the robustness of the tracking the desired angle, is addressed using a minimum variance control approach. This paper presents feasible real-time self-tuning of an approximated proportional derivative PD regulator. The saturation function for the proportional and the integral terms in the control law was used by AL-Samarraie [3] to force the state to slide along the switching manifold. The benefit of nonlinear integral term is to remove the chattering and to minimize the steady state error that happened due to the existence of the external disturbances. AL-Samarraie and Abbas [4] show the effectiveness of using a nonlinear PID controller to force the throttle angle to track the desired reference. The result showed the ability of the nonlinear PID controller to work in the complete system model without simplification and showed that the controller is robust to the variations of system parameters.

An observer-based sliding-mode controller with specific transient response for the ETV system was designed by Nakano et. al., [5]. By using a function-augmented sliding hyper-plane, it is ensured that the output tracking error converges to zero in a finite time. Based on a dynamic LuGre model to represent friction effects, Witty et. al., [6] developed a dynamic model for an electronic throttle valve. An adaptive pulse controller is applied to achieve precise throttle positioning. Horn et. al., [7] use two control strategies, the integral sliding-mode controller as well as the super-twisting algorithm, that, once tuned, they lead to accurate tracking interpretation over the whole range of process. A harmony search algorithm-based fuzzy-PID controller for electronic throttle valve was used by Wang et. al., [8] to improve the responsiveness of ETC. The controller gains are identified by using fuzzy rules.

The model of the ETV is improved and the discrete-time sliding mode controller together with the sliding mode observer is designed Ozguner et. al., [9], to control the ETV which show the successful rejection of the parameter uncertainty and external disturbance. Dagci et. al., [10] recover non-smooth nonlinearities, stick-slip friction, a nonlinear spring and gear backlash, to design robust controller to control the ETV. Pan et. al., [11] utilize a Variable Structure (VS) controller based on Backstepping approach and a sliding mode observer with equivalent control to design a controller for the ETV. It has been proved that in spite of existing non-smooth nonlinearities and unmatched parameter uncertainty the (VS) controller with the observer guaranteed accurate tracking to reference angle [11].

The main challenging problems in designing a controller for the electronic throttle valve are; a) the non-smooth nonlinearity in its model due to the friction model and the nonlinear spring, b) the uncertainty in system model parameters and c) the electronic throttle valve model does not satisfy the matching condition. The first and second problems can be solved via the sliding mode control method, since it can deal with these types of non-smooth nonlinearity in affecting system behavior. The third problem which is frequently arises in the electromechanical system where the uncertainty in system model and the disturbances does not appear in the control channel.

The performance of the ETV control system is the main goal behind the works presented above. By performance we mean the dynamic characteristics of the throttle angle response where it is required to reach the desired value without overshot and with minimum time. To do that the integral sliding mode controller is utilized in this work to precisely force the throttle angle to follow the desired value with desired dynamic characteristics from the first instant and desired time period required to reach the reference value.

2. Mathematical Model

The mathematical model for the electronic throttle valve which considered in the present work is taken from reference [11]. In terms of the valve plate position θ, the rotor angular velocity ω, and the current i induced through the dc motor windings, the electronic throttle valve mathematical model is given by [11];

$$\left.\begin{array}{l} \dot{\theta} = (K_{g1}K_{g2})\omega \\ \dot{\omega} = -\frac{B_{tot}}{J_{tot}}\omega + \frac{K_t}{J_{tot}}i - \frac{1}{J_{tot}}T_f(\omega) - \frac{1}{J_{tot}}T_{sp}(\theta) \\ \dot{i} = -\frac{K_V}{L}\omega - \frac{R}{L}i + \frac{1}{L}u \end{array}\right\} \quad \ldots(1)$$

Where u is the input voltage to the dc motor, $T_f(\omega)$ and $T_{sp}(\theta)$ are the stick–slip friction torque and the nonlinear spring torque respectively. The frictional torque and the nonlinear spring torque are plotted in Figs, (2) and (3) respectively and defined mathematically as follows [11];

$$T_f(\omega) = \begin{cases} F_s, & \omega > 0 \\ 0, & \omega = 0 \\ -F_s, & \omega < 0 \end{cases} \qquad = F_s \; sgn(\omega) \quad \ldots(2)$$

And

$$T_{sp}(\theta) = \begin{cases} D + m_1(\theta - \theta_o) \; if \quad \theta_o < \theta < \theta_{max} \\ -D - m_1(\theta_o - \theta) \; if \quad \theta_{min} < \theta < \theta_o \end{cases}$$
$$= m_1(\theta - \theta_o) + D \; sgn(\theta - \theta_o) \quad \ldots(3)$$

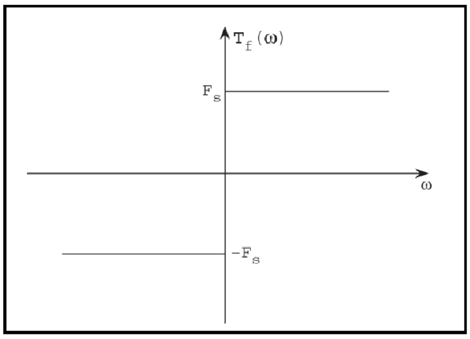

Fig. 2. Coulomb friction [11].

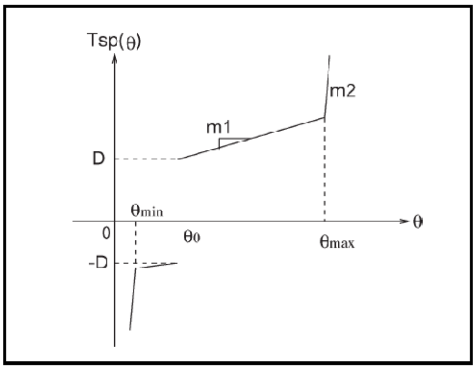

Fig. 3. Nonlinear spring [11].

The physical parameters that appear in the mathematical model are defined in Table (1).

Table 1,
ETV parameters [11].

Parameters	Definition
K_{g1}	Gear ratio of the intermediate gear to the pinion gear
K_{g2}	Sector gear ratio to the intermediate gear
B_{tot}	The total ETV system damping
J_{tot}	The total ETV system inertia
K_t	Motor torque constant
K_V	Motor back emf constant
L	Motor inductance
R	Motor resistance
F_s	Positive constant in Coulomb friction
D	Spring offset
m_1	Spring gain
m_2	Spring limit stop gain
θ_{max}	Spring maximum position
θ_{min}	Spring minimum position
θ_o	Spring default position

To write the ETV model in state space form, define $x_1 = \theta$, $x_2 = \dot{\theta} = (K_{g1}K_{g2})\omega$, and $x_3 = i$. Accordingly the ETV model, as given in equation (1), is written as follows;

$$\left.\begin{array}{l} \dot{x}_1 = x_2 \\ \dot{x}_2 = a_{21}(x_1 - x_{1o}) + a_{22}x_2 + a_{23}x_3 \\ \quad -\mu sgn(x_2) - \beta sgn(x_1 - x_{1o}) \\ \dot{x}_3 = a_{32}x_2 + a_{33}x_3 + b_3 u \end{array}\right\} \quad \ldots(4)$$

Where the state space model parameters are

$$a_{21} = \frac{K_{g1}K_{g2}}{J_{tot}}m_1 , \quad a_{22} = -\frac{B_{tot}}{J_{tot}} ,$$
$$a_{23} = \frac{K_{g1}K_{g2}K_t}{J_{tot}} , \quad \mu = \frac{K_{g1}K_{g2}}{J_{tot}}F_s ,$$
$$\beta = \frac{K_{g1}K_{g2}}{J_{tot}}D, \quad a_{32} = -\frac{K_V}{LK_{g1}K_{g2}} ,$$

$a_{33} = -\frac{R}{L}$, and $b_3 = \frac{1}{L}$.

For these parameters, the maximum and minimum values are given in Table (2) [2].

Table 2,
The maximum and minimum ETV parameters values [2].

parameters	minimal value	maximal value	unite
a_{21}	-69	-95	$1/s^2$
a_{22}	-32	-54	$1/s$
a_{23}	242.424	250	$v.s$ $/rad.kgm^2$
a_{32}	-20	-25	$v.s\ /rad.H$
a_{33}	-1300	-2125	Ω/H
b_3	1000	1250	$1/H$
β	143.5	157	rad/s^2
μ	57	76	rad/s^2
θ_o	0.095	0.095	rad

To simplify the ETV model, the motor inductance L is ignored (i.e., by considering L ≈ 0). Accordingly the throttle valve system model in equation (10) (4) is reduced to a system with lower dimension as can be noted in the following steps:

$$\left(\frac{1}{b_3}\right)\dot{x}_3 = \left(\frac{a_{32}}{b_3}\right)x_2 + \left(\frac{a_{33}}{b_3}\right)x_3 + u,\ \left(\frac{1}{b_3}\right) = L$$
$$\therefore 0 \approx \left(\frac{a_{32}}{b_3}\right)x_2 + \left(\frac{a_{33}}{b_3}\right)x_3 + u \ \text{ for } \ L \approx 0$$

Where $\left(\frac{a_{32}}{b_3}\right)$ and $\left(\frac{a_{33}}{b_3}\right)$ are finite values since b_3 occurs in both a_{32} and a_{33}. Now solving for x_3 yields:

$$x_3 = -\left(\frac{a_{32}}{a_{33}}\right)x_2 - \left(\frac{b_3}{a_{33}}\right)u \qquad \ldots(5)$$

Substituting the value of x_3 from equation (5) into equation (4), yields:

$$\left.\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -a_1(x_1 - x_{1o}) - a_2x_2 + bu \\ &\quad -\mu sgn(x_2) - \beta sgn(x_1 - x_{1o}) \end{aligned}\right\} \ldots(6)$$

where $a_1 = -a_{21}$, $a_2 = -\left(a_{22} - a_{23}\left(\frac{a_{32}}{a_{33}}\right)\right)$ and $b = -a_{23}\left(\frac{b_3}{a_{33}}\right)$. By writing equation (6) as a nominal and perturbation term, we get;

$$\left.\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -a_{1n}(x_1 - x_{1o}) - a_{2n}x_2 + b_nu + d \end{aligned}\right\} \quad \ldots(7)$$

where the subscript n refers to the nominal parameters values and:

$$\begin{aligned} d = -\Delta a_1(x_1 - x_{1o}) - \Delta a_2x_2 + \Delta bu \\ -\mu sgn(x_2) - \beta sgn(x_1 - x_{1o}) \end{aligned} \quad \ldots(8)$$

is the collection of the non-smooth nonlinearity and the uncertainty in system model. Namely it consists of $\mu sgn(\omega) + \beta sgn(x_1 - x_{1o})$ as non-smooth discontinuities due to Coulomb friction and nonlinear spring and $\Delta a_1(x_1 - x_{1o}) + \Delta a_2x_2 + \Delta bu$ is due to uncertainties in system parameters. In addition the model parameters related to their nominal values by the following inequalities;

$$|a_1 - a_{1n}| = |\Delta a_1| < \delta_{a_1},$$
$$|a_2 - a_{2n}| = |\Delta a_2| < \delta_{a_2},$$
$$|b - b_n| = |\Delta b| < \delta_b, \quad |\beta - \beta_n| = |\Delta\beta| < \delta_\beta,$$
$$|\mu - \mu_n| = |\Delta\mu| < \delta_\mu$$

In these inequalities $\delta_{(\ )}$ represent the bound for maximum uncertainty for the parameters. The parameters of equation (6) are given in Table (3) below with maximum uncertainty δ for each of them.

Table 3,
ETV parameters values in equation (6).

parameters names	*Nominal value*	δ
a_1	82	13
a_2	46.54	12.26
b	177.2	63.2
β	*150.25*	*6.75*
μ	*66.5*	9.5

The throttle valve system model given in equation (6) is a second order system and will be used later in designing an ISMC that will force the state to track a certain desired throttle angle (system output).

3. Integral Sliding Mode Control

The major advantage of sliding mode is low sensitivity to plant parameter variations and disturbances which eliminates the necessity of exact modeling. Sliding mode control enables the decoupling of the overall system motion into independent partial components of lower dimension and, as a result, reduces the complexity of feedback design [13]. Sliding mode control implies that control actions are discontinuous state functions which may easily be implemented by conventional power converters with “on-off” as the only admissible operation mode. Due to these properties the intensity of the research at many scientific centers of industry and universities is maintained at high level, and sliding mode control has been proved to be applicable to a wide range of problems in robotics, electric drives and generators, process control, vehicle and motion control.

In the present, the nonlinear system is described by differential equations in an arbitrary n-dimensional state space with a scalar control action:

$$\dot{x} = f(x,u)$$

with $x \in \mathcal{R}^n$, $f \in \mathcal{R}^n$, $u \in \mathcal{R}$, and t denoting the time. The conditions that will ensure sliding motion may be obtained from geometrical considerations: the deviation from the switching surface s and its time derivative should have opposite signs in the vicinity of a sliding surface $s = 0$ (or sliding manifold) [12].

$$\lim_{s\to 0^+} \dot{s} < 0 \text{ and } \lim_{s\to 0^-} \dot{s} > 0$$

Or

$$s * \dot{s} < 0 \qquad \ldots(9)$$

Inequality in equation (9) is referred to as reaching or sliding condition. The control signal that will ensure sliding motion (satisfying sliding condition) is selected as a discontinuous function of the state as follows:

$$u(x) = \begin{cases} u^+(x) \ if \ s > 0 \\ u^-(x) \ if \ s < 0 \end{cases} \qquad \ldots(10)$$

where the scalar functions $u^+(x)$,$u^-(x)$ and $s(x)$ are continuous and smooth, and $u^+(x) \neq u^-(x)$. Accordingly the sliding mode assumes to occur on the surface $s(x) = 0$.

A serious obstacle for utilization of sliding modes in control systems is a phenomenon referred to as “chattering”. The term chattering describes the phenomenon of finite-frequency, finite amplitude oscillations appearing in many sliding mode implementations [12]. The chattering in sliding mode control system can be eliminated by using a smooth function in the control. The price for this elimination is a steady state error as proved in reference [14].

As a new sliding mode controller design the Integral Sliding Mode Control (ISMC) seeks to eliminate the reaching phase by enforcing sliding mode throughout the entire system response [12]. In integral sliding mode the order of the motion equation is equal to the order of the original system rather than reduced by the dimension of the control input as in the conventional sliding mode design. For this reason, it is also named as the full order sliding mode. As a result, robustness of the system can be guaranteed starting from the initial time instant [12].

To derive the control law, consider the following dynamic system;

$$\dot{x} = f(x) + g(x)u + d(x,t) \qquad \ldots(11)$$

where $x \in \mathcal{R}^n$, $f \in \mathcal{R}^n$, $g \in \mathcal{R}^{n\times 1}$,$u \in \mathcal{R}$ and $d \in \mathcal{R}$collected the unmodeled dynamics, the non-smooth nonlinearities and external disturbances.

Rewrite Equation (11) in terms of certain and uncertain dynamics as follows:

$$\dot{x} = f_n(x) + g_n(x)u + \Delta f(x) + \Delta g(x)u + d(x,t) \qquad \ldots(12)$$

where $f_n(x)$, $g_n(x)$ and $\Delta f(x)$, $\Delta g(x)$ are the nominal and the uncertain system dynamics in equation (11). In addition let the control law is redesigned to be

$$u = u_n + u_s \qquad \ldots(13)$$

Where u_n is the Nominal Control used to stabilize the nominal system dynamics with the desired characteristics. The nominal system dynamics is :

$$\dot{x} = f_n(x) + g_n(x)u_n \qquad \ldots(14)$$

The discontinuous control u_s designed to reject the perturbation term in equation (12). The perturbation term and the design of the ISMC are clarified in the following analysis; equation (12) can be written as :

$$\dot{x} = f_n(x) + g_n(x)u_n + g_n(x)u_S + \delta(x,u) \qquad \ldots(15)$$

where $\delta(x,u) = \Delta f(x) + \Delta g(x)u + d(x,t)$ is the perturbation term. The perturbation attributable to parameter variations, unmodeled dynamics, non-smooth nonlinearities and external disturbances and is assumed to fulfill the matching condition [12] i.e.,

$$\delta(x,u) = g_n(x)\hat{\delta}(x,u) \qquad \ldots(16)$$

As a first step in ISMC design procedure, the sliding variable $s(x)$ is defined as:

$$s(x) = s_o(x) + z, \quad s(x),\ s_o(x)\ \&\ z \in \mathcal{R}^1, \ldots(17)$$

which consists of two parts: the first part $s_0(x)$ may be designed as a linear combination of the system states, similar to the conventional sliding mode design; the second part z introduces the integral term and will be determined below. To apply the sliding condition in equation (9) $\dot{s}$ is differentiated as follows;

$$\begin{aligned}\dot{s} &= \frac{\partial s_o}{\partial x}\dot{x} + \dot{z} = \frac{\partial s_o}{\partial x}\big(f_n(x) + g_n(x)u_n \\ &\qquad + g_n(x)u_S + \delta(x,u)\big) + \dot{z} \\ &= \frac{\partial s_o}{\partial x}\big(g_n(x)u_S + \delta(x,u)\big) + \frac{\partial s_o}{\partial x}(f_n(x) + g_n(x)u_n) + \dot{z}\end{aligned}$$

By choosing (similar to the derivation of the integral sliding mode in [12])

$$\dot{z} = -\frac{\partial s_o}{\partial x}(f_n(x) + g_n(x)u_n) \qquad \ldots(18)$$

$\dot{s}$ becomes;

$$\dot{s} = \frac{\partial s_o}{\partial x}\left(g_n(x)u_S + \delta(x,u)\right) \quad \ldots(19)$$

And accordingly the sliding condition becomes;

$$s\,\dot{s} = s\,\frac{\partial s_o}{\partial x}\left(g_n(x)u_S + \delta(x,u)\right) \quad \ldots(20)$$

By selecting u_S as in the conventional sliding mode

$$u_S = -k(x)\,sign(s) \quad \ldots(21)$$

Then equation (20) becomes;

$s * \dot{s} = s\,\frac{\partial s_o}{\partial x} g_n(x)u_S + s\,\frac{\partial s_o}{\partial x}\delta(x,u)$,
Since $s * sign(s) = |s|$, then

$$\begin{aligned} s*\ \dot{s} &= -|s|\,\frac{\partial s_o}{\partial x} g_n(x)\,k(x) + s\,\frac{\partial s_o}{\partial x}\delta(x,u) \\ &\le -|s|\,\frac{\partial s_o}{\partial x} g_n(x)\ k(x) + |s|\,\left|\frac{\partial s_o}{\partial x}\delta(x,u)\right| \\ &\le -|s|\left\{\frac{\partial s_o}{\partial x} g_n(x)k(x) - \left|\frac{\partial s_o}{\partial x}\delta(x,u)\right|\right\} \\ &\le -|s|\left(\frac{\partial s_o}{\partial x} g_n(x)\right)\left\{k(x) - \frac{\left|\frac{\partial s_o}{\partial x}\delta(x,u)\right|}{\frac{\partial s_o}{\partial x}g_n(x)}\right\} \end{aligned} \quad \ldots(22)$$

where it is assumed that $\left(\frac{\partial s_o}{\partial x} g_n(x)\right) > 0$. The discontinuous gain $k(x)$ that will make the inequality in equation (22) less than zero (sliding motion) is selected as follows;

$$k(x) > \frac{\left|\frac{\partial s_o}{\partial x}\delta(x,u)\right|}{\frac{\partial s_o}{\partial x}g_n(x)}$$

or

$$k(x) = k_o + \frac{\left|\frac{\partial s_o}{\partial x}\delta(x,u)\right|}{\frac{\partial s_o}{\partial x}g_n(x)}, \quad k_o > 0 \quad \ldots(23)$$

Where k_o is positive constant. Since the right hand side of equation (19) is discontinuous due to u_S, the sliding variable s will reached the origin in finite time T [15] and the system dynamics then is in sliding motion. Accordingly the system dynamics can be determined using the equivalent control method as follows; when $s(t) = 0,\ \forall\, t \ge T$, $\dot{s}(t) = 0$ too and with $\delta(x,u)$ satisfies matching condition in equation (16), the equivalent control can be determined from equation (19)

$$0 = \frac{\partial s_o}{\partial x}\left(g_n(x)u_S + \delta(x,u)\right)$$

$$\Rightarrow [g_n(x)u_S]_{eq} = -\delta(x,u) \quad \ldots(24)$$

Equation (15) with the equivalent control in equation (24) is reduced to the nominal system dynamics as given in equation (14) with a dimension equal to n. For this reason, the ISMC is named as the full order sliding mode because the dimension in equation (14) is equal to the dimension of the original system in equation (11).

To eliminate the reaching phase, which is a special property for the ISMC, the initial condition for z is selected such that the initial condition for the sliding variable s is zero. This means that the system dynamics is in sliding mode from the first instant. Namely by selecting $z(0) = -s_o(0)$ we have $s(0) = 0$ and $s(t) = 0,\ \ \forall\, t \ge 0$.

The ideal control can now be selected as a continuous state feedback with the desired dynamic characteristics [12]. Eventually the integral sliding mode control law is;

$$\left.\begin{aligned} \dot{z} &= -\frac{\partial s_o}{\partial x} f_n(x) - \frac{\partial s_o}{\partial x} g_n(x)u_n,\ z(0) = -s_o(0) \\ s &= s_o + z \\ u &= u_n - k(x) * sign(s) \end{aligned}\right\} \quad \ldots(25)$$

4. ISMC Design for Electronic Throttle Valve

In this section, an ISMC is designed to the electronic throttle valve that taken into consideration the uncertainty in system model and the presence of a non-smooth nonlinearity and external disturbances. The nominal system for the reduced mathematical model (equation (6)) is given by;

$$\left.\begin{aligned} \dot{x_1} &= x_2 \\ \dot{x_2} &= -a_{1n}(x_1 - x_{1o}) - a_{2n}x_2 + b_n u_n \end{aligned}\right\} \quad \ldots(26)$$

Where the sliding variable is defined as;

$$s = x_2 + z,\ \ z(0) = -x_2(0) \quad \ldots(27)$$

To determine the control law for the ETV system we need to define and calculate the following according to equation (25);

$$\frac{\partial s_o}{\partial x} = [0\ \ 1],\ f_n(x) = \begin{bmatrix} x_2 \\ -a_{1n}(x_1 - x_{1o}) - a_{2n}x_2 \end{bmatrix}$$

$$\Rightarrow \frac{\partial s_o}{\partial x} f_n(x) = -a_{1n}(x_1 - x_{1o}) - a_{2n}x_2$$

$$g_n(x) = \begin{bmatrix} 0 \\ b_n \end{bmatrix} \Rightarrow \frac{\partial s_o}{\partial x} g_n(x) = b_n$$

From equation (8) the perturbation term is

$$\begin{aligned} \delta(x,u) &= \begin{bmatrix} 0 \\ d \end{bmatrix} \\ &= \begin{bmatrix} 0 \\ -\Delta a_1(x_1 - x_{1o}) - \Delta a_2 x_2 + \Delta b u \\ -\mu sgn(x_2) - \beta sgn(x_1 - x_{1o}) \end{bmatrix} \end{aligned}$$

$$\Rightarrow \frac{\partial s_o}{\partial x}\delta(x,u) = \Delta a_1(x_1 - x_{1o}) + \Delta a_2 x_2 + \Delta bu - \mu sgn(x_2) - \beta sgn(x_1 - x_{1o})$$

$$\Rightarrow \left|\frac{\partial s_o}{\partial x}\delta(x,u)\right| = |\Delta a_1(x_1 - x_{1o}) + \Delta a_2 x_2 + \Delta bu - \mu sgn(x_2) - \beta sgn(x_1 - x_{1o})|$$

$$\leq |\Delta a_1||x_1 - x_{1o}| + |\Delta a_2||x_2| + |\Delta b||u| + \mu_{max} + \beta_{max}$$

$$\leq \delta_{a_1}|x_1 - x_{1o}| + \delta_{a_2}|x_2| + \delta_b|u_n - k(x) * sign(s)| + \mu_{max} + \beta_{max}$$

$$\leq \delta_{a_1}|x_1 - x_{1o}| + \delta_{a_2}|x_2| + \delta_b|u_n| + \delta_b(x) + \mu_{max} + \beta_{max}$$

$$\therefore k(x) = k_o + \left(\frac{1}{b_n}\right)\left(\delta_{a_1}|x_1 - x_{1o}| + \delta_{a_2}|x_2| + |u_n| + \delta_b k(x) + \mu_{max} + \beta_{max}\right)$$

Then by solving for $k(x)$ we get

$$k(x) = \left(\frac{b_n}{b_n - \delta_b}\right)\left\{k_o + \left(\frac{1}{b_n}\right)\left(\delta_{a_1}|x_1 - x_{1o}| + \delta_{a_2}|x_2| + \delta_b|u_n| + \mu_{max} + \beta_{max}\right)\right\} \quad \ldots(28)$$

In the next step the ideal control is designed as follows; first let $e = x_1 - x_{1r}$ and $\dot{e} = x_2 - x_{2r}$ then the nominal model given in equation (26) becomes

$$\ddot{e} = -a_{1n}(x_1 - x_{1o}) - a_{2n}x_2 - \dot{x}_{2r} + b_n u_n \quad \ldots(29)$$

Where x_{1r} and x_{2r} are the reference angular position and reference angular velocity in state space, respectively. Accordingly the nominal control is designed as:

$$u_n = \left(\frac{1}{b_n}\right)\begin{Bmatrix} a_{1n}(x_1 - x_{1o}) + a_{2n}x_2 + \dot{x}_{2r} \\ -c_1 e - c_2\dot{e} \end{Bmatrix} \quad \ldots(30)$$

As a result, the nominal system error dynamics becomes;

$$\ddot{e} = -c_1 e - c_2\dot{e},\; c_1, c_2 > 0 \quad \ldots(31)$$

The values of c_1 and c_2 are assigned according to the required system dynamics characteristics.

Finally the ISMC for the ETV system is;

$$\left.\begin{aligned} &\dot{z} = c_1 e + c_2\dot{e} - \dot{x}_{2r},\; z(0) = -x_2(0) \\ &s = x_2 + z \\ &u = u_n - k(x)sign(s) \end{aligned}\right\} \quad \ldots(32)$$

with $k(x)$ and u_n are as given in equations (28) and (30) respectively.

The integral sliding mode control, as can be notedfrom equation (32), leaves the electronic throttle valve with the required system characteristics equation (30) from the first instant. Note also that the sliding variable s_o can be chosen as a linear combinations of states but, however, it can be chosen in terms of the states that the control input acts through their channels. In the ETV, the input u exists in x_2 channel, accordingly we select $s_o = x_2$.

5. Simulation Results

The simulations which are presented below are classified into three category. Each of them either explores the ISMC abilities or explains and analyzes the integral sliding mode controller features. The simulations are based on the full order model as given in equation (4) and the ISMC is as given in equation (32). Also $k(x)$ and u_n are given in equations (28) and (30) respectively.

Let $k_o = 0.2$, while the system parameters, their nominal values and maximum uncertainties are all given in Tables (2) and (3). Note that the value of k_o is slected small in order to reduce the amplitude of the chattering induced in system response. For the nominal controller u_n, c_1 and c_2 are determined based on pole placement method. The poles are selected such that the throttle angle reaches the target without overshot as one of the main requirement in automotive industry. To this end, let the characteristic roots for the nominal system equation (31) are $-35, -35$, then $c_1 = 35^2$ and $c_2 = 70$.

Throughout the simulations the ability of the ISMC are investigated in forcing the throttle valve plate angle to follow three different desired trajectories which are described as follows:

Trajectory A (**Traj.A**): constant reference throttle angle

$\theta_r = 70\ deg.$

Trajectory B (**Traj.B**) [3]: the reference angle is piecewise constant; namely

$$\begin{aligned}\theta_r &= 5.44\,\text{deg.}\ for \quad 0 \le t \le 1\\ &= 70\,\text{deg.}\ for \quad 1 < t \le 2.5\\ &= 25\,\text{deg.}\ for \quad 2.5 < t \le 4\\ &= 45\,\text{deg.}\ for \quad 4 < t \le 5\end{aligned}$$

Trajectory C (**Traj.C**): the reference angle is a periodic function given by

$$\theta_r = (\pi/180) * \left(30 + 3 * \sin\left(\left(\frac{2\pi}{3}\right) * \mathrm{t}\right)\right)$$

In addition the initial conditions employed for all simulations are $(\theta, \dot{\theta}, i) = (5.44^{\circ}, 0, 0)$.

5.1. Numerical simulation test 1: Control system performance with different system parameters

The first set of simulations in this simulation test is carried out with the following system parameters;

$$a_{21} = -84, \qquad a_{22} = -40, \qquad a_{23} = 244,$$

$$\mu = 65, \qquad \beta = 153$$

$$a_{32} = -21, \qquad a_{33} = -1550, \qquad b_3 = 1100$$

Note that the parameters values lie within the range of parameters as illustrated in Table (2).

Figures. (4), (5) and (6) are the ETV system time responses for each type of trajectories presented above. They clarify the ability of the ISMC in forcing the throttle plate angle to follow the desired trajectory within an interval of time not exceeds 0.25 sec. This time interval is related directly to the characteristic roots selected to the nominal system dynamics equation (31).

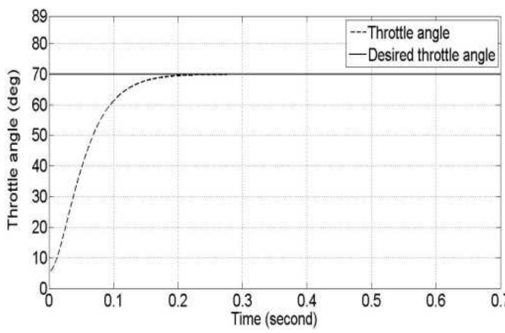

Fig. 4. Throttle angle versus time (Traj.A & 1st parameter set).

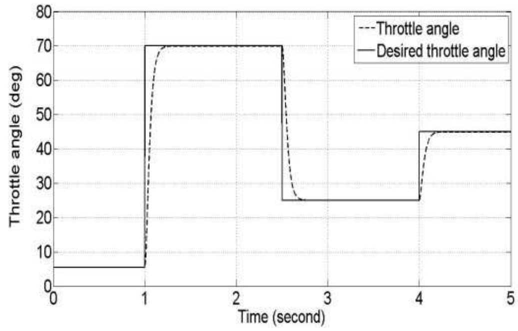

Fig. 5. Throttle angle & its desired versus time (Traj.B & 1st parameter set).

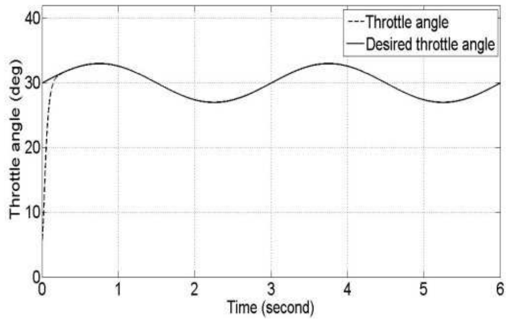

Fig. 6. Throttle angle versus time (Traj.C & 1st parameter set).

The main objective for the designed controller is to regulate the sliding variable s to zero level in finite time. As a main feature of the ISMC the sliding variable is equal to zero from the first instant i.e., $s = 0,\ \forall t \ge 0$. In fact, this is the ideal case where an infinite switching process is assumed. With a finite switching process around $s = 0$, the sliding variable oscillates around $s = 0$ with a certain bound as can be shown in Figures (7), (8) and (9). This bound is function of the switching gain $k(x)$ and the interval of time used for the simulation process.

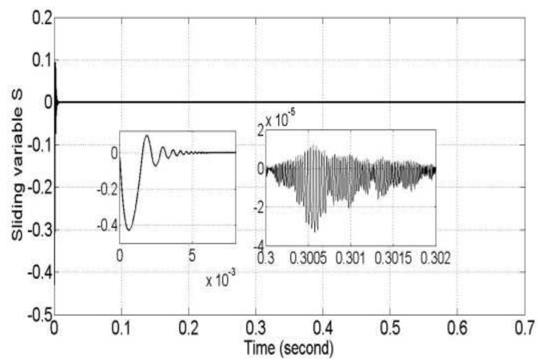

Fig. 7. Switching surface versus time (Traj.A & 1st parameter set).

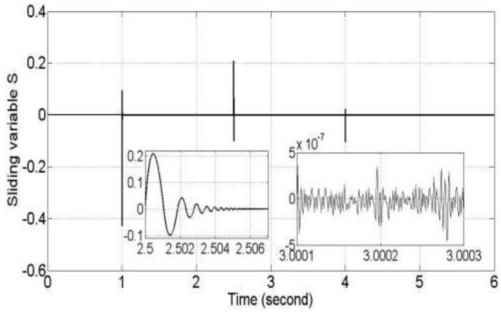

Fig. 8. Switching surface versus time (Traj.B & 1st parameter set).

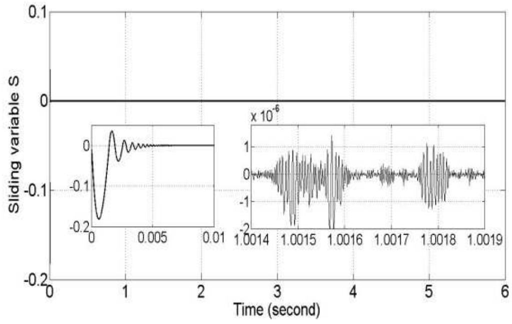

Fig. 9. Switching surface versus time (Traj.C & 1st parameter set).

It can be also noted that the sliding variable requires less than $0.005\, sec.$ to reach and stay very close to zero value. This is a direct consequence of ignoring the electrical system dynamics when designing the ISMC.

The control signal (in voltage) for the three trajectory types are shown in Figures (10), (11) and (12). The figures show, for the sever case, where the throttle angle is opened to 70° and when considering maximum uncertainty in system model, its value is less than 15 voltage. Note that the dark area in Figure (10) represent the high switching process of the controller due to the discontinuity nature of the sliding mode control law (Eq. (32)).

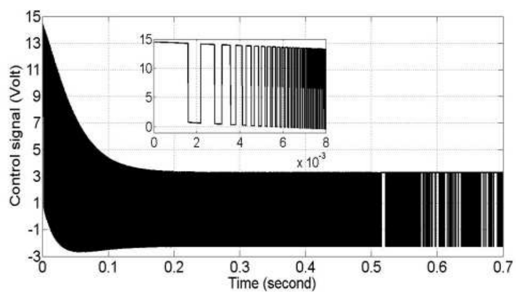

Fig. 10. Control action (u) versus time (Traj.A & 1st parameter set).

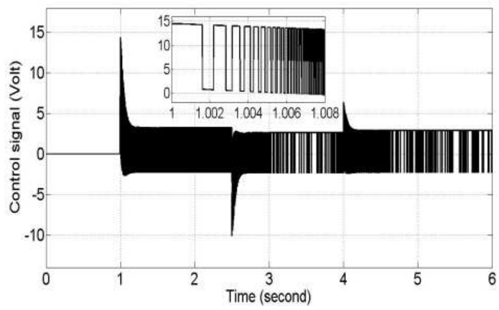

Fig. 11. Control action (u) versus time (Traj.B & 1st parameter set).

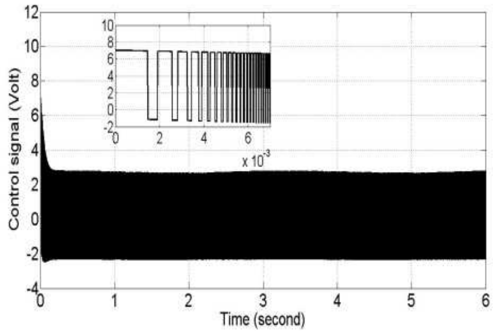

Fig. 12. Control action (u) versus time (Traj.C & 1st parameter set).

The second set of simulations in this simulation use the following system parameters;

$$a_{21} = -75, \quad a_{22} = -47, \quad a_{23} = 248,$$
$$\mu = 72, \quad \beta = 145$$
$$a_{32} = -23, \quad a_{33} = -2000, \quad b_3 = 1200$$

The aim of this simulation is to show that the system response is identical to system response for the first set of simulation demonstrate which represents the distinguished property for the integral sliding mode control theory. This property is clarified later in the numerical simulation 2. The ETV system response for the trajectory types A, B and C are shown in Figures (13), (14) and (15). These figures are very similar to those in Figures (4), (5) and (6) since the nominal system dynamics are the same.

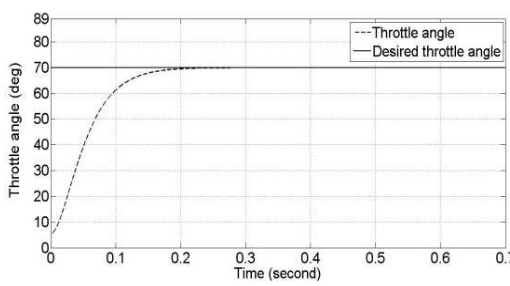

Fig. 13. Throttle angle versus time (Traj.A & 2nd parameter set).

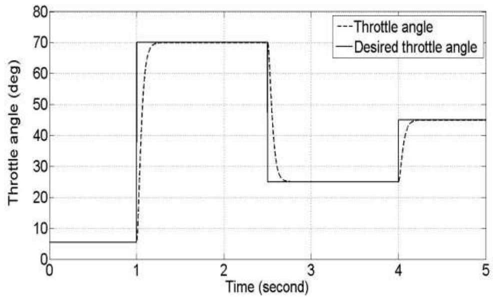

Fig. 14. Throttle angle & its desired versus time (Traj.B & 2nd parameter set).

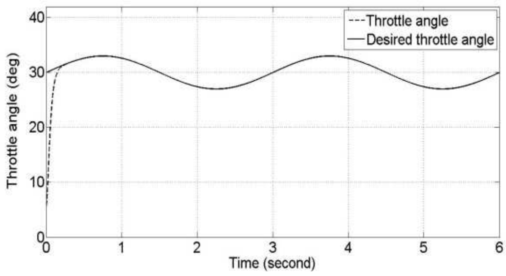

Fig. 15. Throttle angle versus time (Traj.C & 2nd parameter set).

5.2. Numerical simulation test 2: Integral sliding mode controller characteristics

In this simulation, the characteristics of the ISMC are analyzed where the trajectory type A is considered in this simulation and the parameters used for the simulation are the first set of parameters in the preceding simulation test.

In the first test, the aim is to show that the ETV system responses for trajectory A are identical for different system parameters (first and the second test in the first simulation test) as shown in Figure (16). The ETV system is accordingly invariant to the uncertainties in system parameters and to the non-smooth nonlinearities.

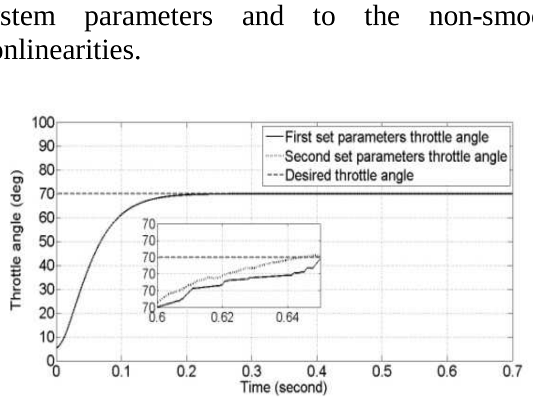

Fig. 16. Throttle angle versus time (Traj.A 1st & 2nd parameter set).

Representing the ETV system by the nominal system dynamics from the first instant is main feature for system dynamics when using ISMC. Figure (17) shows the nominal system of equation (31) and the ETV response for trajectory A where the curves appear very close to each other.

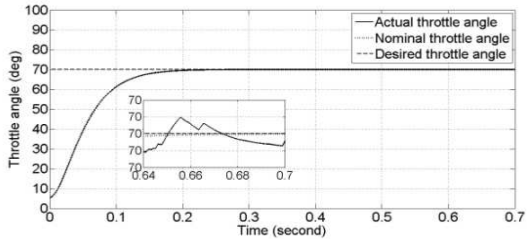

Fig. 17. Throttle angle versus time (Traj.A actual & nominal throttle angle).

The final test, it is devoted to plot the time history for the two terms of the sliding variable x_2 and z. Figure (18) shows that the term z is updated to maintain the sliding variable equal to zero. It can be deduced that the ISMC use a dynamic sliding variable to eliminate the uncertainty and the disturbances affecting the system dynamic and leaves it nominal.

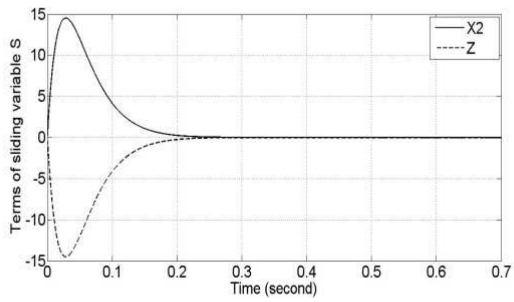

Fig. 18. Terms of sliding variables (x_2,z) versus time.

5.3. Numerical simulation test 3:Chattering problem

When it is required to eliminate the chattering, the signum function which appears in the sliding mode control law is replaced by an approximate function. The arc tan function is used instead of the signum function as follows:

$$sign(s) \approx \frac{2}{\pi}\tan^{-1}(h*s) \qquad \ldots(33)$$

Where $h > 1$ is a design parameter adjusted in such a way that the response resembles the sliding motion but in a continuous manner. The ETV system is simulated with approximate signum function as in equation (33) for trajectory type A and the first set of parameters as in simulation test 1.

Figure (19) shows that the throttle valve response is very close to system response without approximation. The chattering in Figures (20) and (21) is greatly reduced compared with these in Figures (7) and (10).

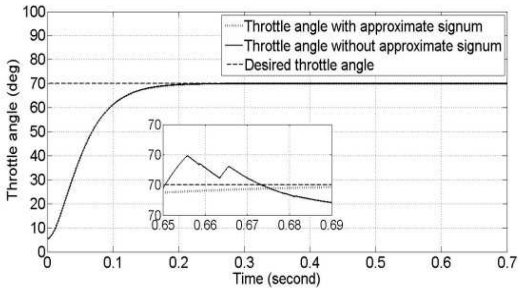

Fig. 19. Throttle angle versus time (Traj.A signum & arc tan function).

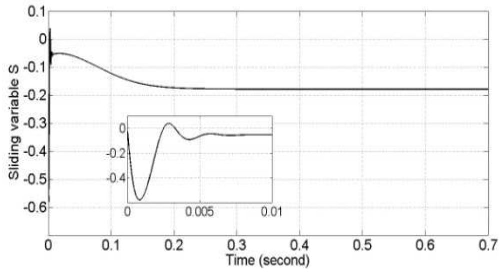

Fig. 20. Switching surface versus time (Traj.A signum & arc tan function).

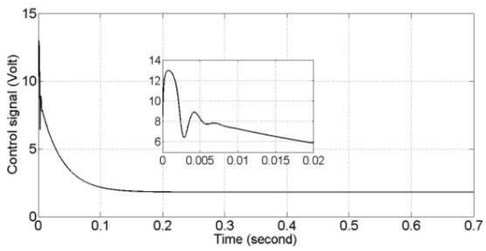

Fig. 21. Control action (u) versus time (Traj.A signum & arc tan function).

6. Conclusions

In this work, the integral sliding mode control theory is utilized to design a robust controller for the Electronic Throttle Valve (ETV) system. ISMC has the ability to cancel the effect of disturbance and uncertainty from any dynamic system by partite the control into two part first the ideal (or nominal) control u_n which is used to stabilize the nominal system dynamics with the desired characteristics and the second part is discontinuous control u_s which is designed to reject the perturbation term.

The main results in this work, due to applying the ISMC for the ETV system, can be summarized as follows:

1. The robustness of the proposed controller has been proved for a bounded uncertainty in system parameters with the presence of a non-smooth nonlinearity and external disturbances and validated via numerical simulations for different parameters values in numerical simulation 1.
2. The system response for different parameters (within the uncertainty bound) is identical. Namely, the ISMC eliminates the perturbation term and leaves the ETV and behaves as a nominal system; irrespective to the presence of the uncertainty in system model and the effects of the non-smooth nonlinearities. Although that this property is the main feature of the ISMC via the equivalent control method it is also proved in simulation tests 1 and 2.
3. The chattering problem is solved by replacing the signum function with arc tan function as approximation. In spite of this approximation, the time required to reach the target is still nearly equal (with suitable value for h) as proved in simulations test 3. The simulation results show that the chattering is eliminated.

7. References

[1] Paolo Mercorelli, "Robust Feedback Linearization using an Adaptive PD Regulator for a Sensorless Control of a Throttle Valve", Mechatronics Vol. 19, No. 8, pp. 1211-1354, 2009.

[2] Reichhartinger M. and Horn M., "Application of Higher Order Sliding-Mode Concepts to a Throttle Actuator for Gasoline Engines", IEEE Transactions on Industrial Electronics, Vol. 56, No. 9, pp. 3322-3329, 2009.

[3] Shibly Ahmed AL-Samarraie, "Design of a Continuous Sliding Mode Controller for the Electronic Throttle Valve System", Journal of Engineering, No. 4, Vol. 17, August, 2011.

[4] Shibly Ahmed AL-Samarraie and Yasir Khudhair Abbas, "Design of Electronic Throttle Valve Position Control System using Nonlinear PID Controller", International

Journal of Computer Applications, Vol. 59, No.15, December, 2012.

[5] Kazushi Nakano, Umerujan Sawut, Kohji Higuchi, and Yasusuke Okajima, "Modelling and Observer-based Sliding-Mode Control of Electronic Throttle Systems", ECTI Transaction on Electrical Eng., Electronics, and Communications, Vol. 4, No. 1, February, 2006.

[6] Carlos Canudas De Witty, Ilya Kolmanovsky, and Jing Sun, "Adaptive Pulse Control of Electronic Throttle", Proceedings of the American Control Conference Arlington, June, 2001

[7] Martin Horn, Anton Hofer and Markus Reichhartinger, "Control of an Electronic Throttle Valve based on Concepts of Sliding-Mode Control", 17th IEEE International Conference on Control Applications San Antonio, Texas, USA, September, 2008.

[8] Hui Wang, Xiaofang Yuan, Yaonan Wang and Yimin Yang, "Harmony Search Algorithm-based Fuzzy-PID Controller for Electronic Throttle Valve", Neural Comput & Applic, 2013.

[9] Urnit Ozguner, Sulgi Hong, and Yaodong Pan, "Discrete-time Sliding Mode Control of Electronic Throttle Valve", Proceding of the 4Oih IEEE Conference on Decision and Control Orlando, Florida USA, December, 2001.

[10] Oguz Hasan Dagci, Yaodong Pan, and Umit Ozguner, "Sliding Mode Control of Electronic Throttle Valve", Proceedings of the American Control Conference Anchorage, AK May, 2002.

[11] Yaodong Pan, Ümit Özgüner and O˘guzHasan Da˘gci, "Variable Structure Control of Electronic Throttle Valve", IEEE Transaction on Industrial Electronics, Vol. 55, No. 11, November, 2008.

[12] V. I. Utkin, J. Guldner, and J. Shi, "Sliding Mode Control in Electro-mechanical Systems", CRC Press. Taylor & Francis Group, 2009.

[13] Andrei A. Agrachev A. Stephen Morse Eduardo D. Sontag H´ector J. Sussmann Vadim I. Utkin, "Nonlinear and Optimal Control Theory", Springer-Verlag Berlin Heidelberg, 2008.

[14] Shibly Ahmed AL-Samarraie, "Invariant Sets in Sliding Mode Control Theory with Application to Servo Actuator System with Friction", WSEAS Transactions on Systems and Control Issue 2, Vol. 8, April, 2013

[15] Giorgio Bartolini, Leonid Fridman, Alessandro Pisano and Elio Usai (Eds.), "Modern Sliding Mode Control Theory", Springer-Verlag Berlin Heidelberg, 2008

تصميم مسيطر تكاملي منزلق النمط للسيطرة على صمام الخنق

شبلي احمد السامرائي* الق صباح بدري** مصطفى حسين مشاري***
*،**،***قسم هندسة السيطرة والنظم /الجامعة التكنولوجية
*البريد الالكتروني: dr.shiblyahmed@yahoo.com
**البريد الالكتروني: alaqsabah@yahoo.com
***البريد الالكتروني: albmustafa@yahoo.com

الخـلاصــة

يعتبر صمام الخنق احد المكونات الرئيسة في محرك السيارة حيث يتم استخدامه لخفض مستوى الانبعاثات و الاقتصاد في استهلاك الوقود. كما يعتبر تصميم نظام التحكم لصمام الخنق من المتطلبات الشائعة حديثا في تكنولوجيا السيارات. تمثل الخصائص اللاخطية للمنظومة نتيجة وجود الاحتكاك والنابض اللاخطي في الموديل الرياضي، والتغاير (Uncertainty) في قيم النظام وعدم تحقق شرط المطابقة (Matching condition) عقبات رئيسة عند تصميم المسيطر للتحكم بصمام الخنق.

تم في هذا العمل استخدام نظرية المسيطر التكاملي منزلق النمط (ISMC) لتصميم مسيطر متين (Robust) للتحكم بصمام الخنق الإلكتروني ETV). حيث من اللحظة الأولى، تم تمثيل صمام الخنق بنظام رياضي مثالي لا يتأثر بالتغاير في قيم المنظومة والخصائص اللاخطية اعتمادا على تطبيق نظرية المسيطر التكاملي المنزلق. تتكون نظرية المسيطر التكاملي المنزلق من جزئين، الاول هو المسيطر المعتاد (Nominal Controller) والذي يتحكم ويسيطر على المنظومة المعتادة (Nominal System) اي بدون اضطراب وتغاير بقيم المنظومة، اما الجزء الثاني غير المستمر فيستخدم لازالة تاثير التغاير بقيم المنظومة وازالة تاثير الخصائص اللاخطية فيها. وأثبتت الميزات للمسيطر التكاملي المنزلق رياضيا وأثبتت عدديا من خلال اجراء سبع تجارب للمحاكاة وبحالات مختلفة. بينت نتائج المحاكاة أن منظومة صمام الخنق تتصرف كنظام المواصفات المعتادة بمختلف القيم. حيث يمكن تحديد خصائص استجابة النظام والوقت اللازم لزاوية الخنق للوصول إلى القيمة المطلوبة بحرية وبدقة. كما بينت قدرة المسيطر للتعامل مع مشكلة الاشباع والارتجاج (Chattering)، فمن خلال المحاكاة تم الغاء الارتجاج باستخدام دالة (Arc tan) بدلا من دالة الإشارة.