DESIGN OF A CONTINUOUS SLIDING MODE CONTROLLER FOR THE ELECTRONIC THROTTLE VALVE SYSTEM

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ABSTRACT

Lowering the emission, fuel economy and torque management are the essential requirements in the recent development in the automobile industry. The main engine control input that satisfies the above requirements is the throttling angle which adjusts the air mass flow rate to the engine port. Due to the uncertainty and the presence of the nonlinear components in its dynamical model, the sliding mode control theory is utilized in this work for the throttle valve angle control system to design a robust controller for this system in the presence of a nonlinear spring and Coulomb friction. A continuous sliding mode control law which consists of a saturation function, instead of a signum function, and the integral of another saturation function is used in this work. This choice for the control structure will prevent the chattering to occurs but with a certain steady state error. On the other hand, the addition of the integral term will effectively reduce the steady state error according to the choice of its parameters. The simulations result for typical references of the opening throttle angle demonstrate the effectiveness of the proposed controller, especially after the addition of a nonlinear integral term.

Key words: Electronic Throttle Valve, Continuous Sliding Mode Control, Nonlinear Integral Control.

الخلاصة

من المتطلبات الحديثة في صناعة السيارات هي خفض مستوى الإنبعاثات، الإقتصاد في صرف الوقود و إدارة ونقل العزوم. إن المدخل الأساسي للمحرك والذي يحقق المتطلبات أعلاه هوعن طريق زاوية الخنق والتي تتحكم بمعدل جريان الهواء الداخل للمحرك. إن إستخدام نظرية المسيطر المنزلق لتصميم المسيطر وبوجود الإحتكاك ونابض لاخطي في هذا البحث هو بسبب المعرفة الغير دقيقة للتمثيل الرياضي للمنظومة و وجود عناصر لاخطية فيه. إن قانون المسيطر المنزلق المستمر المقترح هنا يحوي على دالة الإشباع بدل دالة الإشارة ودالة إشباع أخرى للجزء التكاملي للمسيطر. إن هيكل المسيطر الذي تم إختياره سوف يمنع ظهور الإرتجاج (chattering) ولكن مع بقاء خطأ دائمي في زاوية الخنق عن الزاوية المطلوبة. إن إضافة الجزء التكاملي الاخطي للمسيطر سيعمل على تقليل الخطأ الدائمي في زاوية الخنق وتبعا لعناصر هذا الجزء التي يتم إختيارها. إن نتائج المحاكات الرياضية ولزاوية خنق معتمدة تبعا للمصادر تؤكد قابلية و فعالية المسيطر المقترح و خصوصا بعد إضافة الجزء التكاملي الاخطي.

INTRODUCTION

The electronic throttling angle control system is the newly common requirement trend in automotive technology. Controlling the throttling angle is the control of the plate opining angle which it controls the air amount that enters to the combustion engine. The air flow rate will directly control the output torque engine and consequently the speed will raised or lowered according to the demand. This reveals the importance of controlling the air fuel ratio.

The air mass rate, traditionally, controlled according to the driver demand where the throttling angle is connecting directly to the accelerator by a wire. In this way, many internal and external conditions are ignored in determining the throttle angle such as fuel efficiency, road, or weather conditions which will negatively affects the engine overall efficiency **(Pan et al., 2008)**. To overcome the above deficiency an electronic control module (ECM) is used to accurately determining the required opining angle. Fig.1 shows the details of the electronic control system . Accordingly, the electronic control unit (ECU) determines the precise amount of fuel delivered to the engine. This amount of fuel is just enough to achieve an ideal air fuel ratio (A/F) (stoichiometry, about 14.7:1). The significance of controlling the air fuel ratio (A/F) is well clarified in Fig. 2 where the emissions lowered to a minimum amount (conversion efficiencies of 98% can be reached). The emission gases are like hydrocarbons (HC), carbon monoxide (CO) and nitrogen oxides (NO_x). For a deviations of ± 0.2 air fuel ratio (A/F) the conversion efficiencies of at least one of the emission components is drastically decreased **(Jun-Mo Kang et al., 1999)** (Fig. 2). This make known's the importance of controlling the air fuel ratio as a consequence of throttle angle control.

For the electronic throttle control system, one can mention two problems. The first is the nonlinearities in system model. This is due to the friction in pedal motion and to the nonlinear behavior of the spring in this system. In fact, it inserts discontinuous elements in the mathematical system model. In addition, the voltage, which it is the input to the electronic throttle valve system, does not actuate the throttle angle directly: and this leads to the mismatch property where the disturbance does not actuate the throttle system through the same input channel. The mismatch property is the second problem. For the application of many control theories, these properties added many difficulties.

Many authors had used the sliding mode control theory in designing a controller for the throttle control system. All of them overcome the first problem (model nonlinearity) by using sliding mode control theory as a robust tool with respect to uncertainty and nonlinearities in system model. In **(Kazushi et al., 2006)**, the inductance in the mathematical model of the DC motor is ignored. Thus, the motor voltage becomes the input to the throttle angle dynamics directly. The performance of the control system in this case depends on the actual inductance value. On the other hand, **(Ozugner et al., 2001)** use the complete throttling system model but with nominization of the throttling system model to a nonlinear canonical form. In the same reference the authors construct the switching manifold as a plane in canonical form of the throttle dynamical system state space and then compute the control action required to force and maintain the state close to switching manifold. The state will slide (the sliding motion) until it reach a region near the origin without reaching the origin. This because they use an approximate form to the signum function in order to represent the friction and the nonlinear spring models by a differentiable functions. A higher order sliding mode control theory was used by **(Reichhartinger et al., 2009)** to design a sliding mode controller to the throttling system. The higher order sliding mode concept eliminate the chattering arising in the classical sliding mode control theory and applied for system which have relative degree greater than one with respect to the switching function as in the case of the throttling system model.

In the present work a continuous sliding mode controller is used containing two saturation functions. The first is used instead of the signum function that is used traditionally in classical sliding mode controller design. The saturation function will remove the chattering that will occur due to the signum function; while the second saturation function is used in the integral term which will help greatly in reducing the steady state error caused by the nonsmooth disturbances affect the throttle valve system model.

In the following sections the mathematical model is presented first in terms of the error function (the deference between the throttle angle and the desired one), after that the proposed continuous sliding mode controller is presented with the corresponding stability prove using the Lyapunov stability criteria. The latter two sections are devoted to the controller parameters calculation and the simulations result which prove the effectiveness of the proposed control law.

MATHEMATICAL MODEL

The electronic throttling valve is shown in Fig. 3. It consists of a DC motor, a motor pinion gear, an intermediate gear, a sector gear, a valve plate, and a nonlinear spring. The mathematical model for the electronic throttle valve consists of the DC. motor mathematical model **(Reichhartinger et al., 2009)**

$$\frac{di}{dt} = -\frac{R}{L}i - \frac{k_m}{L}\omega_m + \frac{1}{L}u \qquad (1)$$

and the mechanical system described by

$$\frac{d\theta}{dt} = \omega$$
$$\frac{d\omega}{dt} = \frac{k_m N}{J}i - \alpha(\theta - \theta_o) - \beta * sign(\theta - \theta_o) - \gamma\omega - \delta * sign(\omega) \qquad (2)$$

where i is the armature current, ω_m is the motor angular velocity, u is the input voltage, k_m is the inductive voltage constant, L and R are the inductance and resistance in the armature circuit, respectively. Also, in eq. (2), θ is the throttle valve angle, ω is the angular velocity of the throttle valve, N is the gear ratio and J is the equivalent moment around the throttle axis, and their values are defined by

$$N = \frac{\omega_m}{\omega} \ \& \ J = N^2 J_m + J_{th} \qquad (3)$$

The model parameters α, β, γ and δ are identified experimentally **(Reichhartinger et al., 2009)** (Table 1) and formally modeled the friction effect, includes both the viscous and Coulomb friction, and the nonlinear spring and which acts on the throttle valve. In addition, the maximum actuator value must not exceed the following constraint **(Reichhartinger et al., 2009)**

$$|u| \leq U_{max} = 10\text{V} \qquad (4)$$

Now by ignoring the inductance L and defining the state variables x_1 and x_2 as follows:

$$\left.\begin{aligned} x_1 &= \theta - \theta_r \\ x_2 &= \omega - \omega_r \end{aligned}\right\} \quad (5)$$

the throttle angle dynamical equation is formulated in the present work as:

$$\dot{x}_1 = x_2$$
$$\dot{x}_2 = -a_1(x_1 + \theta_r - \theta_o) - a_2(x_2 + \omega_r) + bu + d_1(\theta, \omega) - \dot{\omega}_r \quad (6)$$

Where $a_1 = \alpha, \quad a_2 = \gamma + \frac{k_m^2 N^2}{JR},$

$b = \frac{k_m N}{JR}$, and

$$d_1(\theta, \omega) = -\beta * sign(\theta - \theta_o) - \delta * sign(\omega)$$

In sliding mode control theory a certain switching function is selected first in terms of the state variable $(s = s(x), x = (x_1, x_2))$. The basic requirement is that when the state at the switching manifold $(s = 0)$, it goes asymptotically to the origin. Thus the following switching function is selected:

$$s(x) = x_2 + \lambda x_1, \ \lambda > 0 \quad (7)$$

It can be simply checked, with the aid of the equation $\dot{x}_1 = x_2$, that when the state is at the switching manifold $(x_2 = -\lambda x_1)$, it is regulated asymptotically to the origin with time constant equal to $(1/\lambda)$. Therefore, the controller u is designed such that it direct the state to the switching manifold and maintain it there for all future time.

Thus, by regarding the switching manifold as the desired output, we derive its dynamical equation as follows:

$$\dot{s} = -a_1(x_1 + \theta_r - \theta_o) - a_2(x_2 + \omega_r) + bu + d_1(\theta, \omega) - \dot{\omega}_r + \lambda x_2 \quad (8)$$

Our aim now (the next section), is to design a controller u such that, after a certain time period, the switching function $s(x)$ reaches its zero level (the switching manifold).

CONTINUOUS SLIDING MODE CONTROLLER DESIGN

The sliding mode control approach is recognized as one of the efficient tools to design robust controllers for complex high-order nonlinear dynamic plant operating under uncertainty conditions **(Agrachev et al., 2004)**. The chattering phenomenon generally seems as motion which oscillates about the sliding manifold. There are two possible mechanisms which produce such a motion. The first is in the absence of switching nonidealities such as delays, i.e., when the switching device is ideally switching at an infinite frequency. While the second is by replacing the signum function by a continuous function **(Slotine, 1983)**. The signum function is a discontinuous function used in sliding mode controller formula and its cause the chattering behavior in the dynamical system.

In this work, a continuous sliding mode control law is proposed which consists of a proportional and integral terms for a saturation function but with a different linear intervals. The proposed control law is

$$u = v - k * Sat_{\epsilon_1}(s) - \eta \int_{t_0}^{t} Sat_{\epsilon_2}(s(\tau)) d\tau \quad (9)$$

where $k > 0 \ and \ \eta > 0$, and $Sat_{\epsilon_i}(s)$, $i = 1,2$ is the saturation function defined by;

$$Sat_{\epsilon_i}(s) = \begin{cases} 1 & |s| > \epsilon_i \\ \frac{s}{\epsilon_i} & |s| \le \epsilon_i \end{cases}$$

Now, to determine the controller parameters, the following Lyapunov function is candidate

$$V(s) = \frac{1}{2}s^2 \qquad (10)$$

Its time rate of change is

$$\dot{V} = s\dot{s} = s * (-a_1(x_1 + \theta_r - \theta_o) - a_2(x_2 + \omega_r) + bu + d_1(\theta,\omega) - \dot{\omega}_r + \lambda x_2)$$

$$= bs * \left\{-\left(\frac{a_1}{b}\right)(x_1 + \theta_r - \theta_o) + \left(\frac{\lambda - a_2}{b}\right)x_2 + u + \left(\frac{1}{b}\right)d_1(\theta,\omega) - \left(\frac{1}{b}\right)(a_2\omega_r + \dot{\omega}_r)\right\} \qquad (11)$$

Now, by considering the uncertainty in system parameters, we have

$$-\left(\frac{a_1}{b}\right)(x_1 + \theta_r - \theta_o) + \left(\frac{\lambda - a_2}{b}\right)x_2 - \left(\frac{1}{b}\right)(a_2\omega_r + \dot{\omega}_r) = f_n(x_1, x_2, \omega_r, \dot{\omega}_r) + \Delta(\theta,\omega,\omega_r,\dot{\omega}_r) \qquad (12)$$

where

$$f_n(x_1, x_2, \omega_r, \dot{\omega}_r) = -\left(\frac{a_1}{b}\right)_n (x_1 + \theta_r - \theta_o) + \left(\frac{\lambda - a_2}{b}\right)_n x_2 - \left(\frac{1}{b}\right)_n (a_2\omega_r + \dot{\omega}_r) \qquad (13)$$

and the sub-script n refers to the nominal system parameters value, while Δ refers to the terms due to the uncertainty in system parameters. Accordingly, Δ is given by;

$$\Delta(\theta,\omega,\omega_r,\dot{\omega}_r) = -\Delta\left(\frac{a_1}{b}\right)(\theta - \theta_o) + \Delta\left(\frac{\lambda - a_2}{b}\right)\omega - \Delta\left(\frac{1}{b}\right)\dot{\omega}_r \qquad (14)$$

Consequently, $\dot{V}$ becomes

$$\dot{V} = bs * \left\{f_n(x_1, x_2, \omega_r, \dot{\omega}_r) + \Delta(\theta,\omega,\omega_r,\dot{\omega}_r) + u + \left(\frac{1}{b}\right)d_1(\theta,\omega)\right\}$$

$$= bs * \left\{f_n(x_1, x_2, \omega_r, \dot{\omega}_r) + \Delta(\theta,\omega,\omega_r,\dot{\omega}_r) + v - k * Sat_{\epsilon_1}(s) - \eta\int_{t_o}^{t} Sat_{\epsilon_2}(s(\tau))d\tau + \left(\frac{1}{b}\right)d_1(\theta,\omega)\right\},$$

(15)

The first controller term v, is taken equal to

$$v = -f_n(x_1, x_2, \omega_r, \dot{\omega}_r) \qquad (16)$$

and by considering the following

$$Sat_{\epsilon_1}(s) = sgn(s) * Sat_{\epsilon_1}(|s|) \qquad (17)$$

eq. (14) becomes

$$\dot{V} = -bs * \left\{k * sgn(s) * Sat_{\epsilon_1}(|s|) - \Delta(\theta,\omega,\omega_r,\dot{\omega}_r) - \left(\frac{1}{b}\right)d_1(\theta,\omega) + \eta * \int_{t_o}^{t} Sat_{\epsilon_2}(s(\tau))d\tau\right\}$$

$$= -b|s| * \left\{k * Sat_{\epsilon_1}(|s|) - sgn(s) * \left(\Delta(\theta,\omega,\omega_r,\dot{\omega}_r) + \left(\frac{1}{b}\right)d_1(\theta,\omega)\right)\right\} - b * \eta * s\int_{t_o}^{t} Sat_{\epsilon_2}(s(\tau))d\tau \qquad (18)$$

If k is chosen as

$k > \left|\Delta(\theta,\omega,\omega_r,\dot{\omega}_r) + \left(\frac{1}{b}\right) d_1(\theta,\omega)\right|,$
$\forall |s| > \epsilon_1$

(19)

Now for a proper choice of η and ϵ_2 ,
$s * \int_{t_0}^{t} Sat_{\epsilon_2}(s(\tau))d\tau > 0,\ \forall |s| > \epsilon_1$, this leads to

$\dot{V} \le 0\ ,\ \forall |s| > \epsilon_1$ (20)

Also, and according to Khalil **(Khalil, 2002)**, the state will reach and stay in the following interval

$|x_1| \le \frac{\epsilon_1}{\lambda}$ (21)

In fact one can omit the addition of the integral term in the control law and select the values of ϵ_1 (for certain λ) according to the required accuracy. Unfortunately, very small value of ϵ_1 will cause a high oscillation around the equilibrium point and may induce the chattering in system dynamical behavior. To increase the accuracy and to prevent the chattering, the integral term is added with an appropriate selection of its gain η and ϵ_2. Namely for

$\epsilon_2 < \epsilon_1$ (22)

the steady state error will be bounded by

$|x_1| \le \frac{\epsilon_2}{\lambda}$ (23)

The inequality (23) also means that the term $s * \int_{t_0}^{t} Sat_{\epsilon_2}(s(\tau))d\tau > 0,\ \forall |s| > \epsilon_2$, and moreover it is greater than or equal to

$\frac{|s|}{\eta} * \left\{max\left|\Delta(\theta,\omega,\omega_r,\dot{\omega}_r) + \left(\frac{1}{b}\right) d_1(\theta,\omega)\right| - k * \frac{|s|}{\epsilon_1}\right\}$

This point can be easily deduced from eq. (18).

COMPUTATIONS OF THE CONTROLLER PARAMETERS

The numerical value of the electronic throttle valve model is presented in Table (1) **(Reichhartinger et al., 2009)**. We need here to calculate four controller design parameters, namely k, ϵ_1, η and ϵ_2. First, the steady state error according to inequality (21) is selected to be less than $0.5\ deg.$, i.e.,

$\frac{\epsilon_1}{\lambda} = \frac{\pi}{360}$

Thus for $\lambda = 12,\ \epsilon_1 = 0.1047$.

To begin, we estimate the following bound:

$$h = \max_t \left|\Delta(\theta,\omega,\omega_r,\dot{\omega}_r) + \left(\frac{1}{b}\right) d_1(\theta,\omega)\right|$$

The value h is computed according to the reference throttle angle to be considered in the simulations and to the parameters in Table 1, is taken as

$$h = 2.4$$

where the nominal parameters are taken as the mean of the maximal and minimal values. Hence

$k = 2.5 > h$

and finally, the integral term parameters are chosen as

$\eta = 5 \quad and \quad \epsilon_2 = 0.01$

In the following section the simulations are done based on the electronic throttle model as presented in eq. (1), i.e., the DC. motor mathematical model is included without ignoring the inductance L.

SIMULATIONS RESULT

Through the simulations the system parameters are taken as:

$\alpha = 90,\ \beta = 146,\ \gamma = 47,\ \delta = 62,$
$\theta_o = 0.095,\ km = 0.02,\ R = 1.6,$
$J = 0.0016,\ N = 20\ , L = 0.0009.$

The first set of simulations is devoted to verify the effectiveness of the proposed controller in eq. (9), namely the controller ability to constrain the error according to the inequalities (21) and (23) without and with the integral term respectively. In these simulations, it is required for the throttle angle to follow a desired opening angle equal to 60^o. The effectiveness is measured here by the steady state error of the switching function $s(t)$. Therefore, we remove the integral term from the control law and plot first the switching function as shown in Fig. 4. As can be seen in this figure the steady state error of $s(t)$ is less than ϵ_1 ($\epsilon_1 = 0.1047$) and consequently the steady state error for the throttle angle is less than $0.5\ deg$ as can be verified in Fig. 5. In addition, the control action is plotted in Fig. 6 which satisfies inequality (4).

Next, we add the integral term to the control law, and plot $s(t)$ with time in Fig. 7. The steady state error is eliminated completely and the throttle angle will follow the desired angle without any steady state error (Fig. 8).

This is the first contribution of adding the nonlinear integral term to the controller formula. The second contribution may be explored when an external torque is added to the model (see **(Pan et al., 2008)** for the realization of adding the external disturbance). The external torque may be taken here as:

$$T_e = 25 * J * \sin\,(100\pi * t)$$

The presence of a periodic disturbance causes an oscillation in system response especially when there is no integral term. This feature for the controller when added the integral term is will clarified in Fig. 9 which plots the switching function with and without the integral term. It can also be noted that the integral term eliminate greatly the high frequency oscillation of the switching function and of course with a better accuracy. In fact the steady state error for the switching function will not exceeding 0.01 as selected in previous section and for the throttle angle (Fig. 10) it will be less than $0.05\ deg$. In addition, the plot of the control action with and without the integral term is found in Fig. 11. It can be noted that the actuation voltage is smoother with the integral term and still satisfying the voltage constrain in the inequality (4).

In the second simulations set, the throttle angle is forced to follow a typical opining angle reference. The reference angle consists of a train of different opining throttle angle and with different time durations **(Horn, 2008)**. The throttle angle response and the error between the throttle and the reference angle are plotted in Fig. 12. Also, in Fig. 13, the control action (the voltage) is plotted, where the maximum voltage still satisfying constraint (4).

As a final simulations test, the reference angle will be in a periodic form, as follows:

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

In Fig. 14, the ability of the proposed controller is clarified where the throttle angle is forced to follow the reference angle θ_r. In addition the switching function and the control effort are plotted in Fig. 15. It can be seen that the switching function value is constrained by ± 0.03 which exceed ϵ_2. To make the switching function constrained by 0.01 the value of η is taken equal to 75 and $\epsilon_2 = 0.005$. The simulation result in Fig. 16 shows that the switching function is constrained by ± 0.01 after a period of time not to exceed 0.1 second with a maximum control action less than 8 voltage. This situation reveals the flexibility of the proposed controller and its ability to attenuate the disturbances to a small bound without a great increase in the control action value.

CONCLUSIONS

In this paper a continuous sliding mode controller for a throttle valve system was proposed. The saturation function was used in the proportional and the integral terms in the control law to force the state to slide along the switching manifold. A Lyapunov function was used to prove the stability of the continuous sliding mode controller. The addition of a nonlinear integral term helps greatly in preventing the chattering that may occurs due to the requirement of a high precession opening angle; also in reducing the steady state error that occurs due to the presence of the external disturbances, friction and the nonlinear spring model. The simulation results prove first the effectiveness of the proposed controller when constraining the steady state error to lie within ± 0.05 *deg*.. Secondly, eliminating the high oscillatory actuation as could be verified clearly from the simulations result. Finally, the last simulation set showed the ability of the proposed controller in controlling the steady stat error by adjusting the integral term parameter η and ϵ_2. This was done without a large increase in the control action and without violating the maximum voltage constraint.

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Table 1: Electronic Throttle Valve Parameters.

Param -eters	Minimal Value	Maximal Value	Units
α	69	95	$1/s^2$
β	143.5	157	rad/s^2
γ	32	54	$1/s$
δ	57	76	rad/s^2
θ_o	0.095	0.095	rad
k_m	0.02	0.02	$V.s/rad$
R	1.3	1.7	Ω
L	0.8×10^{-3}	1×10^{-3}	H
J_m	4×10^{-6}	4×10^{-6}	$kg.m^2$
J_{th}	2×10^{-6}	50×10^{-6}	$kg.m^2$
N	20	20	

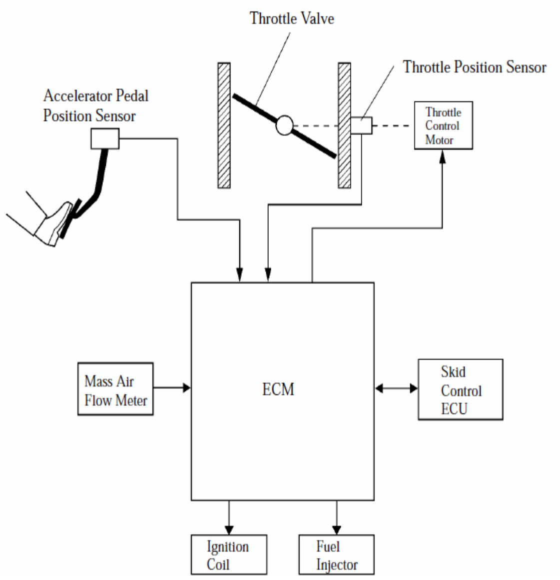


Fig. (1) Electronic control module system diagram**(http://www.ncttora.com/fsm/2003/4runner/215/ncf/1gr-fe_etcs-i.pdf.)**

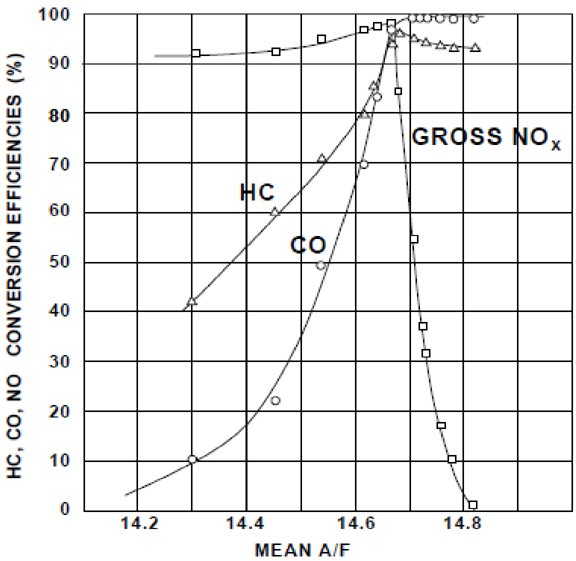


Fig. (2) Steady state conversion efficiency **(Jun-Mo Kang et al., 1999)**.

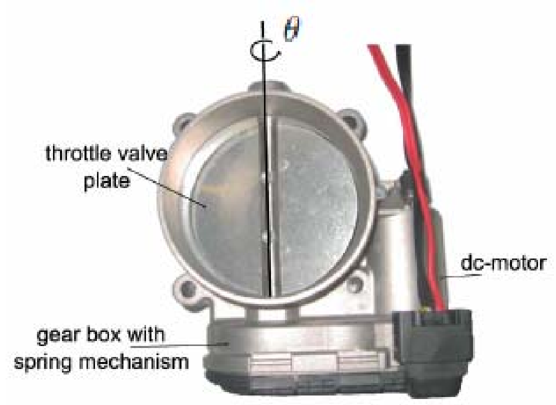


Fig. (3) Electronic Throttle Valve **(Reichhartinger et al., 2009)**

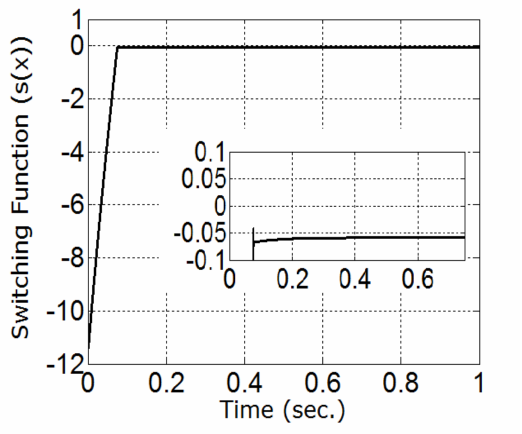


Fig. (4) $s(x)$ vs. time without the integral term.

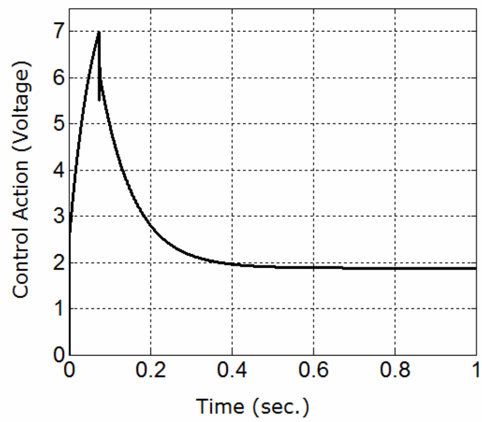
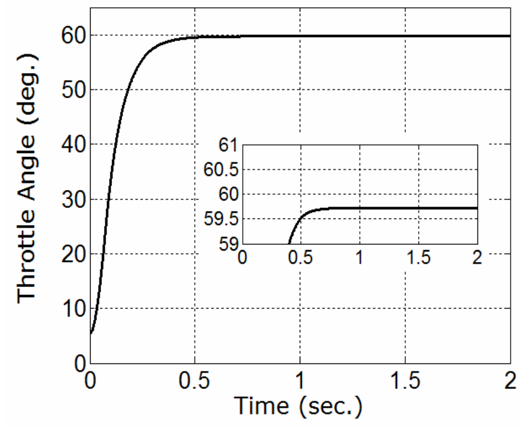
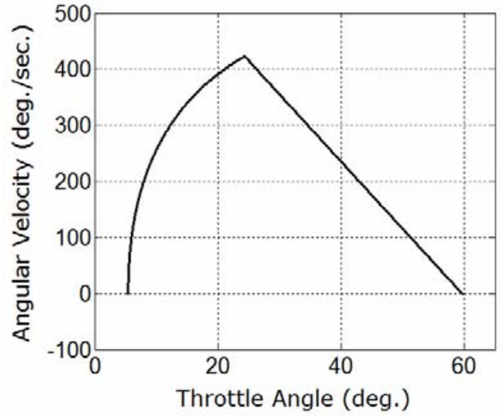


Fig. (6) Control Action vs. time without the integral term.



(a)



(b)

Fig. (5) Simulation without the integral term a) the throttle angle vs. time b) the phase plot.

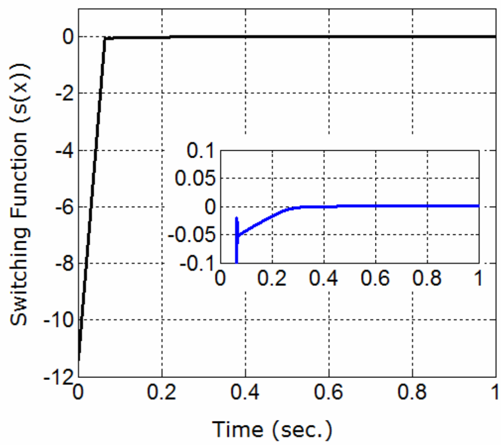


Fig. (7) $s(x)$ vs. time with the integral term.

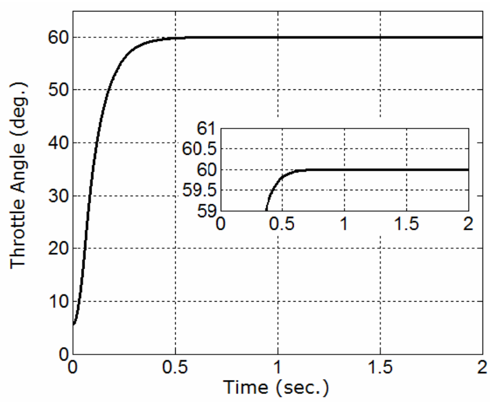
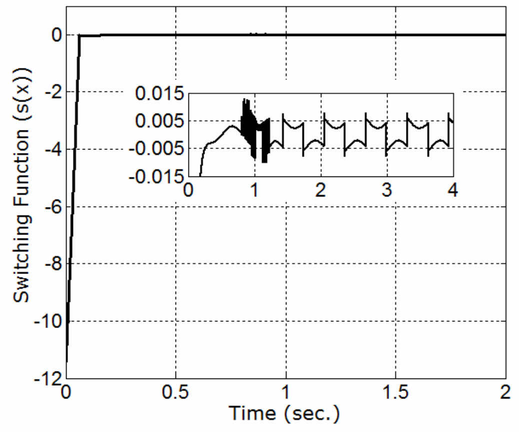
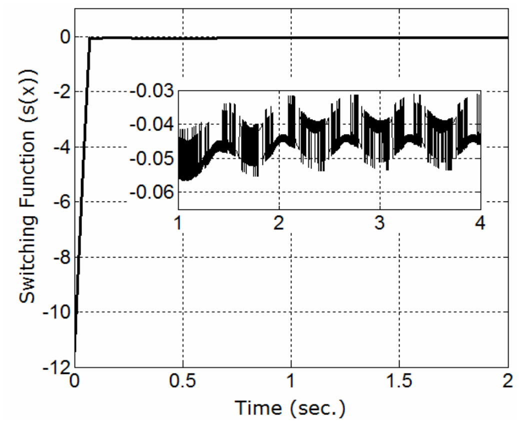


Fig. (8) The throttle angle vs. time with the integral term.



(a)



(b)

Fig. (9) $s(x)$ vs. time a) with the integral term b) without the integral term.

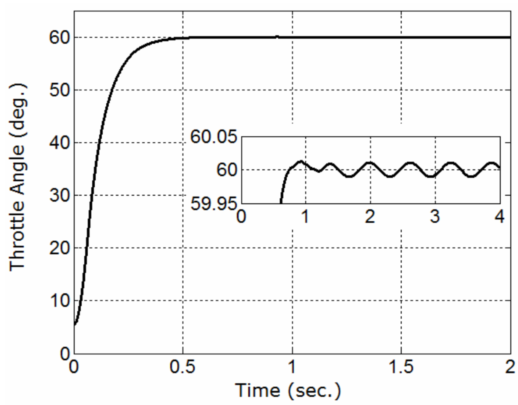
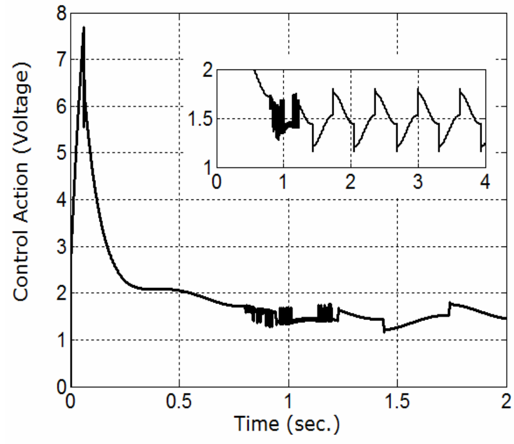
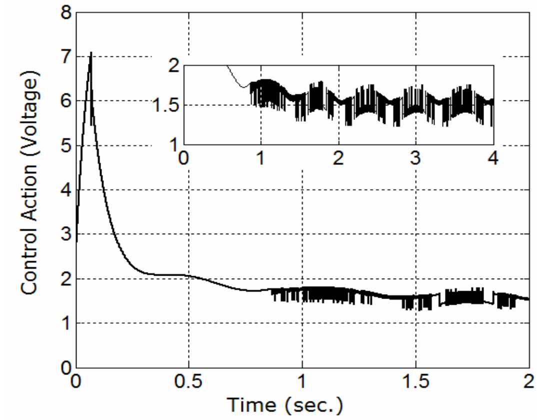


Fig. (10) The throttle angle vs. time with the integral term.

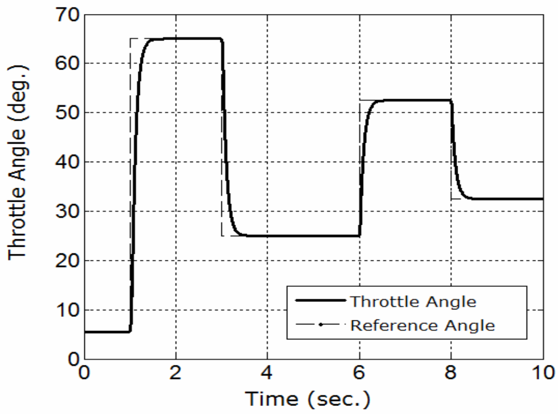


(a)

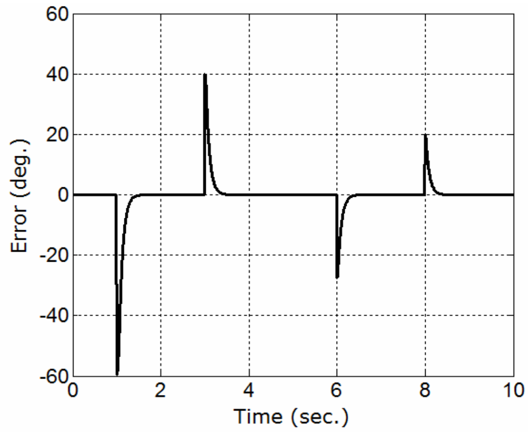


(b)

Fig. (11) Control Action vs. time a) with the integral term b) without the integral term.



(a)



(b)

Fig. (12) a) Throttle angle vs. time b) the error $(\theta - \theta_r)$ vs. time.

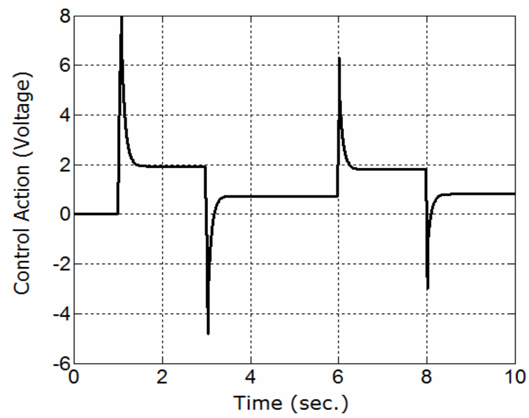
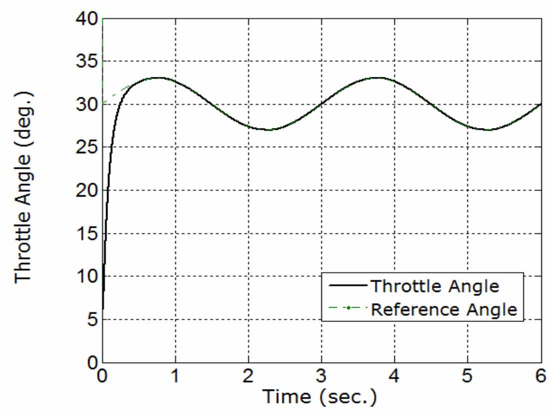
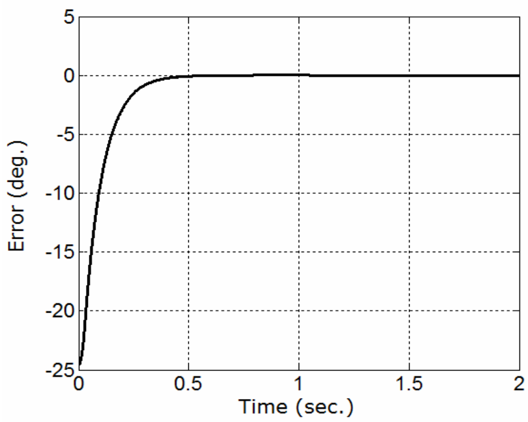


Fig. (13) Control Action vs. time.

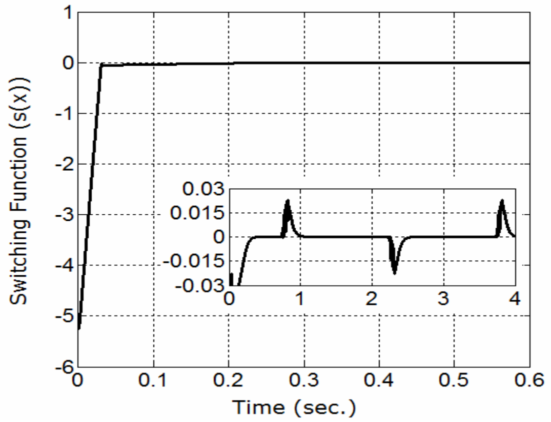


(a)

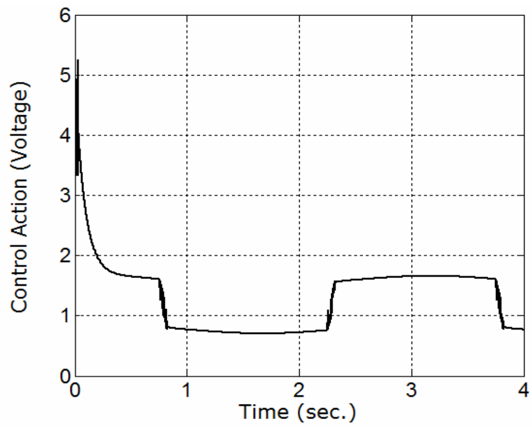


(b)

Fig. (14) a) Throttle angle vs. time b) the error $(\theta - \theta_r)$ vs. time.

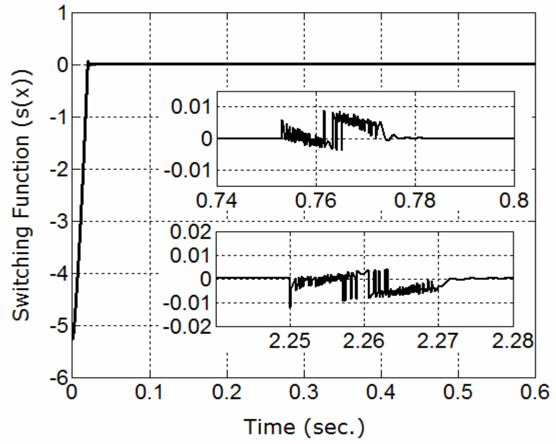


(a)

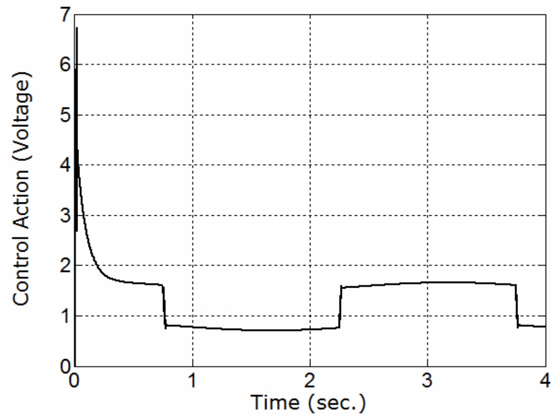


(b)

Fig. (15) a) $s(x)$ vs. time b) control Action vs. time.



(a)



(b)

Fig. (16) Simulation with $\eta = 75$ and $\epsilon_2 = 0.005$ a) $s(x)$ vs. time b) control Action vs. time.