



Contents lists available at <http://qu.edu.iq>

Al-Qadisiyah Journal for Engineering Sciences

Journal homepage: <http://qu.edu.iq/journaleng/index.php/IQES>



Design of Hydraulic Exciter Using Nonstandard Backstepping with Integral Sliding Mode

Mujtaba A. Flayyih^a, Mohsin N. Hamzah^b, Jafar M. Hassan^c

^a Al-Mustaqbal University College, Hillah, Iraq.

^b University of Technology, Baghdad, Iraq.

^c Al-Turath University College, Baghdad, Iraq.

ARTICLE INFO

Article history:

Received 10 November 2020

Received in revised form 31 December 2020

Accepted 31 December 2020

Keywords:

Hydraulic simulator,

Quarter car,

Suspension system,

Sliding Mode control.

ABSTRACT

The hydraulic system plays an important role in the design of many mechanical devices due to its high power to weight ratio. In this work, a theoretical study of a hydraulic stimulator was performed to conduct suspension tests in a quarter car in the laboratory. Due to high nonlinearity in the hydraulic systems, a robust controller based upon integral sliding mode (ISM) with nonstandard backstepping was used to perform the tracking process of the hydraulic actuator to the required road profiles. The controller design divided into two parts: the first part deals with the generation of the ideal control force which satisfied all the tracking requirements by using ISM. The second part of the controller is the replacing of the ideal control force with the actual hydraulic force with all parameters by using nonstandard backstepping control. The effectiveness of the exciter control system was performed by using two road profiles i.e. Bump road profile and random road profile.

© 2020 University of Al-Qadisiyah. All rights reserved.

1. Introduction

Active vibration control is one of the most advanced technologies to improve functionality and safety of mechanical structures [1], [2]. Hydraulic systems have been widely used in a large number of fields applications to perform the activity of these vibration systems due to their small size to power ratios and their ability to apply very large force and torque. Hydraulic systems play an important role in transportation, earth moving equipment, aircraft and industry machinery with heavy duty applications [3], [4]. The hydraulic system is highly nonlinear system due to different parameters such pressure flow characteristic, variations of control volumes and associated stiffness, etc. Merritt [5]. From the control point of view, the hydraulic actuator have two main functions: the first one is the force tracking control Niksefat et al. [6] and the second one was

the position control Ali et al. [7]. In this work, the hydraulic system designed as an exciter for testing the quarter car suspension system model in the laboratory i.e. for generation road profiles for the system. Therefore, the literature focused on the positioning control of the hydraulic system. There are many researches used the linearized dynamic equation of the hydraulic servo-system such as Eong L. et al. [8], Dean H. et al. [9] and others. This linear system is not suitable to give real results when applying control methods because of physical of the hydraulic system which is highly nonlinear. Another trends of studies used linear control methods such as G. P. Liu et al. [10], Bobrow et al. [11] etc. G. P. Liu et al. [10] performed analytical and experimental study for two hydraulic actuators to track the position of the system by using an

* Corresponding author.

E-mail address: mujtaba_abdulkadhim@mustaqbal-college.edu.iq (Mujtaba A. Flayyih)

<https://doi.org/10.30772/qjes.v13i4.724>

2411-7773/© 2021 University of Al-Qadisiyah. All rights reserved.



This work is licensed under a [Creative Commons Attribution 4.0 International License](https://creativecommons.org/licenses/by/4.0/).

optimal-tuning nonlinear PID control strategy of two hydraulic actuators to track the position of the system. In the same context, Bobrow et al. [11] adopted the optimal linear quadratic control to solve the problem of a servo single rod hydraulic cylinder system. These linear control methods such as PID controller, pole placement, optimal controls etc. which are not sensitive for disturbances and nonlinearities of the hydraulic system. On the other hand, Xiuping Yuan et al.[12] performed an analytical study of the single rod hydraulic actuator and made certain the structure of the Fuzzy controller, input or output variable, the quantitative variable, the proportional variable and fuzzy reasoning theorem based on the manipulative experiment. Backstepping control method is one of the robust control methods and used by many researchers such as Prut Nakkarat et al. [13] and others. They employed an experimental and theoretical study of the electrohydraulic actuator. The researchers derived directly the dynamic equation based upon force variables instead of position variables. They used proportional integral observer to estimate the force rate and pressure. Sliding mode controller is the one of the well algorithms used in hydraulic systems since it has its outstanding characteristics in compensating for the nonlinearities and disturbances of electromechanical systems, therefore many of researchers applied these methods on electrohydraulic systems, for example Y Liu et al. [14], Kai Guo et al. [15] and others. Y Liu et al. [14] designed a standard sliding mode controller for the electrohydraulic system connected to the flexible load, i.e. spring and mass. They introduced first-order weighted error function as sliding surface. the experimental results presented the effectiveness of the position control of this system. While, Kai Guo et al. [15] performed a theoretical and experimental study for force tracking of the electrohydraulic actuator. They used cascade controller based on an extended disturbance observer to track desired position trajectory for electro-hydraulic single-rod actuators in the presence of both external disturbances and parameter uncertainties. The researchers used sliding mode controller to compensate the disturbance estimation error estimated. In the same context, they used the backstepping techniques with Lapenove stability for tracking the actuator pressure. they presented that the proposed controller gives excellent tracking performance in the presence of the disturbances and nonlinearities.

The main objective of the present work is to design and implement an ISMC with non-standard backstepping for the exciter system with hydraulic actuator. The first step in this controller is to design an ideal control force to perform the required single using ISMC. The current ISMC is an update to the conventional sliding mode, on which the reaching phase was eliminated by initiating the system state at sliding surface, therefore, the robustness is per formed from the first instant. The feature of the ISMC is that it keeps the order of the system where the uncertainties and perturbations rejected from the system model and make the system as an ideal model with known nominal parameters. In addition, ISMC can also be used as a perturbation’s estimator, which solves one of the main drawbacks of the sliding mode control, which is the chattering problem Utkin et al. [16]. The second step is the inclusion of the nonlinear single- rod hydraulic actuator as controller force by using the non-standard backstepping which based on the sliding mode control to perform stability. Also, the observer sliding mode control was used as a differentiator of the ideal control force in the backstepping step due to the nonlinearities and the uncertainties appears in the derivative of the ideal control force.

2. Mathematical Modelling

Mathematical model is defined as a set of equations which describe the dynamic characteristic of the system there are differed way to represent the model depending on the application of the system and circumstances, one mathematical model may be better suited than the other models for specific. The exciter model system used in this paper consists of mass (m) which tracked the required path generated by the actuator, **Fig. (1)**. Therefore, the mathematic model presented as:

$$m\ddot{z} = u - mg - c\dot{z} \tag{1}$$

Where z is the displacement of the mass and u is the ideal control force, g is the acceleration of gravity and c is the damping coefficient of the actuator.

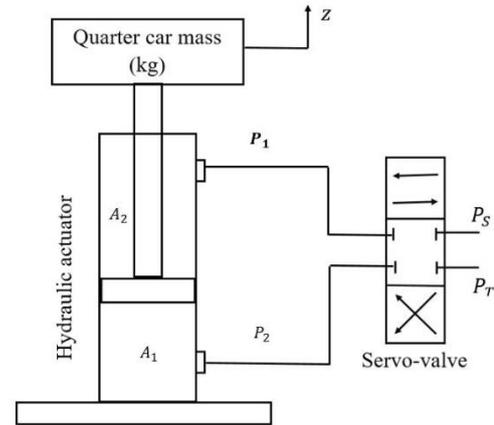


Figure 1. schematic of the model

The force which be used to generate path of the system is produced by hydraulic actuator. Therefore, it is necessary to explain the dynamics equation of the actuator and servo valve, firstly the actuator dynamics equation is:

$$F_h = P_1A_1 + P_2A_2 \tag{2}$$

Where F_h is the actuator hydraulic force, P_1 is pressure at the chamber (1) of the cylinder, P_2 is pressure at the second chamber of the cylinder, A_1 is the cross sectional area of the chamber (1) while A_2 the cross sectional area of the chamber (2). The derivation of P_1 & P_2 will be started from the continuity equation through the cylinder which be written as in [5], [17]

$$\begin{aligned} \frac{V_1}{\beta} \dot{P}_1 &= -A(\dot{z}) + Q_1 - Q_{L1} - Q_{LE1} \\ \frac{V_2}{\beta} \dot{P}_2 &= A(\dot{z}) - Q_2 + Q_{L1} - Q_{LE2} \end{aligned} \tag{3}$$

Where $V_1 = V_{01} + A z$, $V_2 = V_{02} + A z$ are the total volume of the chambers, V_{01} and V_{02} are the extend and retract chamber volumes when (z) equal to zero, β is the effective bulk modulus of the hydraulic oil, Q_{L1} is the internal flow leakage of the cylinder, Q_{LE1} and Q_{LE2} are the external flow leakage of the chambers, Q_1 is the supply flow rate to the forward chambers and Q_2 is the return flow rate from the return chamber. The forward and return flow rates can be described as [7]:

$$\begin{aligned} Q_1 &= k_v x_v \{s_g(x_v)(P_s - P_1) + s_g(-x_v)(P_1 - P_t)\} \\ Q_2 &= k_v x_v \{s_g(x_v)(P_2 - P_a) + s_g(-x_v)(P_s - P_2)\} \end{aligned} \tag{4}$$

Define function

$$Sg(*) = \begin{cases} 1 & \text{if } * \geq 0 \\ 0 & \text{if } * < 0 \end{cases} \quad (5)$$

Where k_v is the servo valve flow coefficient, x_v is the displacement of the servo valve, P_s is the supply pressure and P_t is the pressure of the tank.

Practically while the hydraulic actuator is running, the servo valve displacement x_v can be handled using the input voltage u_h that corresponds to the different required forces. The servo valve dynamic equation can be approximate as Fialho et al. [18]

$$\dot{x}_v = \frac{1}{\tau} (-x_v + u_h)$$

Where τ is the time constant of the servo-valve. It is being showed that the dynamic of the servo-valve is much faster than the actuator dynamic when the time constant value is very small. Thus, the equation (5) can be simplified as an algebraic equation $-x_v + u_h = 0$ Deshpande et al. [19]. Therefore, the equation (3) can be rewritten as:

$$\begin{aligned} Q_1 &= u_h H_1 \\ Q_2 &= u_h H_2 \end{aligned} \quad (7)$$

$$H_1 = k_v \{s_g(u_h)(P_s - P_1) + s_g(-u_h)(P_1 - P_t)\} \quad \text{and} \quad H_2 = k_v \{s_g(u_h)(P_2 - P_a) + s_g(-u_h)(P_s - P_2)\}.$$

The hydraulic force of the actuator simplified as:

$$\dot{F}_h = \dot{P}_1 A_1 - \dot{P}_2 A_2$$

$$\dot{F}_h = \frac{\beta}{V_1} (-A_1(\dot{z}) + u_h H_1) A_1 - \frac{\beta}{V_2} (A_2(\dot{z}) - u_h H_1) A_2$$

$$\begin{aligned} \dot{F}_h &= \\ & -(\dot{z}) \left\{ \frac{\beta A_1^2}{V_1} + \frac{\beta A_2^2}{V_2} \right\} + \\ & u_h \left\{ \frac{\beta H_1 A_1}{V_1} + \frac{\beta H_2 A_2}{V_2} \right\} \end{aligned} \quad (8)$$

The states variable of the dynamic system are written as:

$$\dot{x}_1 = z$$

$$\dot{x}_2 = (a_1 x_3 - g + a_1 x_2)$$

$$\dot{x}_3 = -(x_2) a_3 + u_h a_4 \quad (9)$$

$$\text{Where } a_1 = \frac{1}{m}, a_2 = \frac{c}{m}, a_3 = \left\{ \frac{\beta A_1^2}{V_1} + \frac{\beta A_2^2}{V_2} \right\} \text{ and } a_4 = \left\{ \frac{\beta H_1 A_1}{V_1} + \frac{\beta H_2 A_2}{V_2} \right\}$$

Where x_1, x_2 represents the displacements and velocity of the moving mass and x_3 represents the dynamic equation of the pressure difference of the single rod hydraulic actuator.

3. Controller Design

3.1. ISMC of the ideal control force

The main objective of the control system in this work is tracking process i.e. the mass follows of the required path which is supplied to control system as an input signal. Therefore, the first step in the control design is to drive an error function between the required path and the input required signal as follows:

$$e(t) = x_1 - x_r$$

Where the x_1 required displacement and the x_r is the ideal signal supplied to the system. Now let the state variables of the equation (12) as:

$$e(t) = x_1 - x_r = e_1(t)$$

$$\dot{e}_1(t) = \dot{x}_1 - \dot{x}_r = e_2(t)$$

$$\dot{e}_2(t) = \dot{x}_2 - \dot{x}_r$$

Therefore, the equation (13) presents as:

$$\dot{e}_1(t) = x_2 - \dot{x}_r$$

$$\dot{e}_2(t) = (a_1 u - g + a_2 x_2) - \dot{x}_r$$

The electrohydraulic system includes several types of uncertainties and perturbations arise from variations of the system parameters such as variation of mass dynamics and the variations of hydraulic system, therefore the dynamic system of the electrohydraulic is rewritten as:

$$\dot{e}_1(t) = x_2 - \dot{x}_r \quad (6)$$

$$\dot{e}_2(t) = (a_1 u - g + a_2 x_2) - \dot{x}_r + d(e, t) + \Delta a_1 u + \Delta a_2 x_2$$

Where $e_i, (i = 1 \text{ to } 2) \in R^2, a_1 \in R^1, u \in R, d(e, t) \in R, d(e, t)$ unmodeled dynamics and the nonsmoothed nonlinearities in the system and $\Delta a_1 u$, and $\Delta a_2 x_2$ represents uncertain parameters of the system dynamic. In the same context, let the design control law is:

$$u = u_n + u_s$$

Where the u_n and u_s represented the nominal and discontinuous controls of the system respectively. The nominal control used to stabilize the nominal system dynamics with the desired characteristics Such as the dynamics of the nominal system become as:

$$\dot{e}_1(t) = x_2 - \dot{x}_r$$

$$\dot{e}_2(t) = (a_1 u_n - g + a_2 x_2) - \dot{x}_r$$

While the discontinuous control u_s worked to rejected the perturbation terms and the nonlinearities of the dynamic system in equation (15). After that, the design of the integral sliding mode and the perturbation terms can be formulating as follows:

$$\dot{e}_1(t) = x_2 - \dot{x}_r$$

$$\dot{e}_2(t) = (a_1 u_n + a_1 u_n - g + a_2 x_2) - \dot{x}_r + \delta(e, u)$$

Where $\delta(e, u) = d(e, t) + \Delta a_1 u + \Delta a_2 x_2$ is the perturbation term and represented the parameters variations, unmodeled dynamics, non-smooth nonlinearities and external disturbances and it assumed to be obtain matching condition of the dynamic system Kim et al. [20].

$$\delta(e, u) = a_3 \hat{\delta}(e, u) \quad (16)$$

The design procedure of the ISMC begins with the definition of the sliding variables(x):

$$s(x) = s_o(e) + z \quad (17)$$

Where $s(e), s_o(e)$ and $z \in R^1$. In the same context, the sliding variable contains two parts. The first one $s_o(e)$ will be designed similar to the conventional sliding mode i.e. as a linear combination of the system states. The term of the z represented the integral term and determined as below. From the sliding mode control theory [20], the derivation from the switching surfaces s and its time derivative should be having an opposite sign in the nearby area of the switching surface $s = 0$, i.e.

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

$$\text{Or } s \dot{s} < 0 \quad (18)$$

Elimination of the reaching phase which is distinguishing property of the ISMC achieved by choosing the initial condition z same that the initial condition of the sliding mode control s which is zero. This mean that the dynamic of the system is in sliding surface from first instantaneity by selecting $z(0) = -s_o(0)$.

In this design the $s_o(e)$ is equal to the state variable e_2 , therefore, \dot{s} is differentiated as below:

$$\dot{s} = \dot{e}_2 + \dot{z} = (a_1 u_s + a_1 u_n - g + a_2 x_2) - \dot{x}_r + \quad (19)$$

$$\delta(e, u) + \dot{z}$$

The integral term will be chosen similar to the derivative of the ISMC in [1] i.e.

$$\dot{z} = (a_1 u_n - g + a_2 x_2) - \ddot{x}_r \tag{20}$$

Thus \dot{s} becomes:

$$\dot{s} = a_1 u_s + \delta(e, u) \tag{21}$$

And accordingly, the sliding condition becomes:

$$\dot{s} s = s (a_1 u_s + \delta(e, u)) \tag{22}$$

By selecting the conventional sliding mode control

$$u_s = -\rho(x) \text{sign}(s)$$

Then the equation (24) becomes:

$$\dot{s} s = s (a_1 (-\rho(e) \text{sign}(s)) + \delta(e, u))$$

Since $s \cdot \text{sign}(s) = |s|$ then

$$\dot{s} s = -a_1 \rho(e) |s| + s \delta(e, u)$$

$$\dot{s} s \leq -a_1 \rho(e) |s| + |s| |\delta(e, u)|$$

$$\dot{s} s \leq |s| \{-a_1 \rho(e) + |\delta(e, u)|\}$$

$$\dot{s} s \leq -|s| \{a_1 \rho(e) - |\delta(e, u)|\}$$

$$\dot{s} s \leq -a_1 |s| \left\{ \rho(e) - \frac{|\delta(e, u)|}{a_1} \right\} \tag{24}$$

It was assumed that $a_1 > 0$. The discontinue gain $\rho(x)$ that will perform the inequality in the equation (27) to ensure the right-hand side of this equation to be negative value i.e.:

$$\rho_o(x) > \frac{|\delta(x, u)|}{g_2}$$

Or $\rho_o(x) = k_o + \frac{|\delta(x, u)|}{a_4}$, where the $k_o > 0$

(25)

by referring to the equation (24), the discontinuity in the right-hand side due to the u_s will needed the finite time T to reach to the origin and then the system dynamic become in the sliding motion. Based on the the dynamic system will be achieved by using the equivalent control which performed as follows: when $s(t) = 0, \forall t \leq T$, [21]also the $\dot{s}(t) = 0$ and with $\delta(x, u)$ satisfied the matching condition in equation (19), the equivalent control can be calculated from the equation (24) as shown:

$$0 = u_s + \delta(x, u)$$

$$\Rightarrow [b u_s]_{eq} = -\delta(x, u) \tag{26}$$

By substituting the equivalent control of the equation (29) in to the control law of the ISMC with perturbations i.e. equation (18), the system is reduced to the nominal form as in equation (17) with the dimension equal to n. For this reason, the ISMC is considered as full order sliding mode control because of the dimension of equations (15) & (17) are equal.

Since the remaining system is only the nominal system as in equation (10) the next step in the design of this control method is the determination of the nominal part of the system:

$$\dot{e}_1(t) = x_2 - \dot{x}_r$$

$$\dot{e}_2(t) = (a_1 u_n - g + a_2 x_2) - \ddot{x}_r \tag{27}$$

Therefore, the control design of the nominal system is:

$$u_n = \frac{1}{a_1} (-n_1 e_1 - n_2 e_2 + g + a_2 x_2 - \ddot{x}_r) \tag{28}$$

As the result the nominal system dynamic is:

$$\dot{e}_2 = -n_1 e_1 - n_2 e_2, \tag{29}$$

Where n_1 & n_2 are assigned based on the required system characteristic.

3.2. Backstepping of the hydraulic force

Backstepping control is a well method to the control problems nonlinear systems such as active suspension system. A virtual control assumed to follow the intermediate variable, which performed to achieve the objectives of the control design. The hydraulic actuator force F_h considered virtual control force to track the ideal control u that designed by ISMC to perform the position tracking of the mass by the electrohydraulic system. The error between the virtual variable F_h and the active control element u will be approach gradually to zero (28) thus:

$$ee(t) = F_h - u \tag{30}$$

Where $ee(t)$ is the error function between the ideal and virtual function. To derive the control law that will make the value $s(t)$ goes to zero, a nonstandard Backstepping [22] is utilized here. First we have needed to derive the error function dynamics error function by differentiating equation (32)

$$\begin{aligned} \dot{ee}(t) &= \dot{F}_h - \dot{u} \\ &= \dot{x}_3 - \dot{u} \\ &= -(x_2) a_3 + u_h a_4 - \dot{u} \end{aligned} \tag{31}$$

The hydraulic system has many unmodelled parameters, no smoothed nonlinearities and uncertain parameters, therefore the equation (34) is rewritten as:

$$\dot{ee}(t) = -(x_2) a_3 + u_h a_4 - \dot{u} + d(x, t) \tag{32}$$

Where $d(x, t)$ unmodeled dynamics the nonsmoothed nonlinearities system and represents uncertain parameters of the system dynamic.

Let the design control law is:

$$u_h = u_{h_n} + u_{h_d} \tag{33}$$

Where the u_{h_n} and u_{h_d} represented the nominal and discontinuous controls of the system respectively. The nominal control used to stabilize the nominal system dynamics with the desired characteristics Such as the dynamics of the nominal system becomes as:

$$\dot{ee}(t) = -(x_2) a_3 + u_{h_n} a_4 - \dot{u} \tag{34}$$

In the other hand, the discontinuous part used to reject the uncertainties in the system model i.e. the discontinuous control is performed as:

$$\dot{e}(t) = u_{h_d} a_4 + d(x, t) \tag{35}$$

Let the $u_{h_d} = -\rho_o(x) \text{sign}(ee(t))$

$$\begin{aligned} e(t) \dot{e}(t) &= ee(t) (-\rho_o \text{sign}(ee(t)) a_4 \\ &\quad + d(x, t)) \\ &= -\rho_o(x) |ee(t)| a_4 + |d(x, t)| ee(t) \\ &\leq -\rho_o(x) |ee(t)| a_4 + |d(x, t)| |ee(t)| \\ &\leq |e(t)| a_4 \left\{ -\rho_o(x) + \frac{|d(x, t)|}{\rho_o} \right\} \end{aligned} \tag{36}$$

It was assumed that $g_2 > 0$. The discontinue gain $\rho_o(x)$ that will perform the inequality in the equation (19) to ensure the right-hand side of this equation to be negative value i.e.:

$$\rho_o(x) > \frac{|\delta(x, u)|}{g_2}$$

Or $\rho_o(x) = k_o + \frac{|\delta(x, u)|}{a_4}$, where the $k_o > 0$

3.3. State Differentiator Design

In Eq. (31), the proposed control law assumes ideal control force u and its derivative are available. The first state u is available since it determined from the first step of the control design, i.e. ISMC. The second state \dot{u} is the time derivative of u . Hence, we need to obtain it using an observer. A robust sliding mode differentiator (SMD) is proposed here to get \dot{u} by knowing u only. Kokotović et al. [23] give the sliding mode differentiator;

$$\dot{\eta} = \alpha * \tan^{-1}(\gamma u) \quad \{ \quad (38)$$

Where σ is the SMD variable, α and ρ are differentiator parameters. The third equation in (39) $\tau \dot{v} + v = \alpha * \tan^{-1}(\gamma u)$ is a low pass filter (LPF) with time constant τ , where the output of the LPF, v , is the estimated derivative \dot{u} . According to Deshpande et al. [19], the bound on the steady state estimation error is given by:

$$|v(t) - \dot{u}| \leq 2 \tau \rho \tan\left(\frac{\pi}{2\alpha} h\right) \quad (39)$$

Where $h = \sup_t |u|$

4. Result and discussion

As mentioned above, the hydraulic system exciter in this work used to simulate the road disturbances on the quarter car suspension system. Therefore, two cases of road profiles studied to perform this purpose Mishary [24] **Fig. (2):**

Case 1: Bump Road Profile

The mathematical model of this profile is **Fig. (2 a)**, [18]

$$x_r = \begin{cases} d_o(1 - \cos(\omega_r t)), & t \leq 1 \\ 0, & t > 1 \end{cases}$$

Where d_o represents the peak amplitude and ω_r , a constant frequency in the disturbance model which depends on the car velocity and on the width of the disturbance on the road, which are set as. $d_o = 0.1 m$ and $\omega_r = 2\pi rad$.

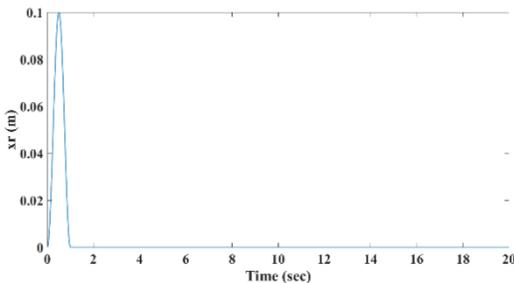
Case 2: Random Road Profile

The mathematical model of this profile is **Fig. (2 b)**, [19]

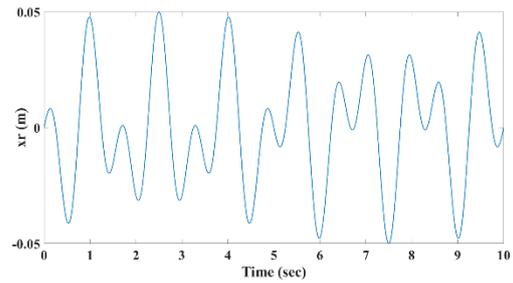
$$x_r = 0.05 \cos(2\pi t) \sin(0.06 \pi t)$$

Table 1. the model parameters of the active suspension

Parameter	Value	parameter	value
m	290 kg	c	1000 N m/sec
k_v	1e-9	β	1.4e9
A	0.001311 m^2	P_s	10342502 N/m^2
V_0	0.01311 m^3		



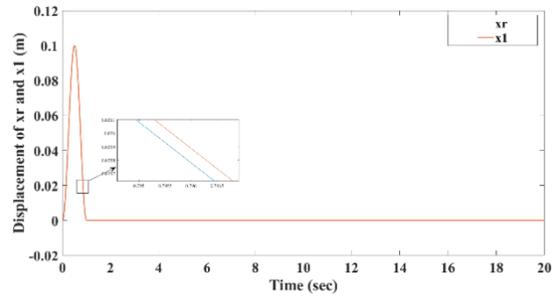
(a)



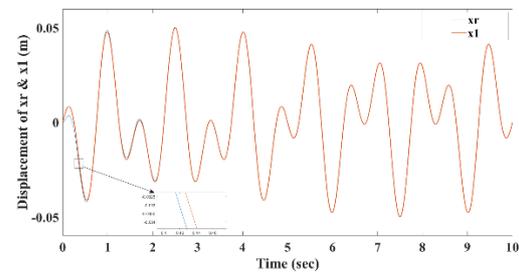
(b)

Figure 2. the simulated road disturbance (m)

Fig. (2) presents the vertical displacement of the quarter car mass and the required profile for two cases. It shows the effectiveness of the proposed control method of the single rod hydraulic actuator to track the required profile with high accuracy. Since the proposed control method in this work is divided into two steps, it is necessary to show the features of the ISMC and nonstandard backstepping to perform tracking process. For ISMC, i.e. generation of the ideal control force, the sliding variable is the main parameter should be discussed. **Fig. (3)** shows that the sliding variable (s) equals to zero from the first instant i.e. $s = 0$ for $t \geq 0$. This behavior of sliding surface (s) which begins and continues with zero along simulations time presented the effectiveness of the of ISMC over the conventional sliding mode control which reaches the stability over reaching phase not from the first instant.

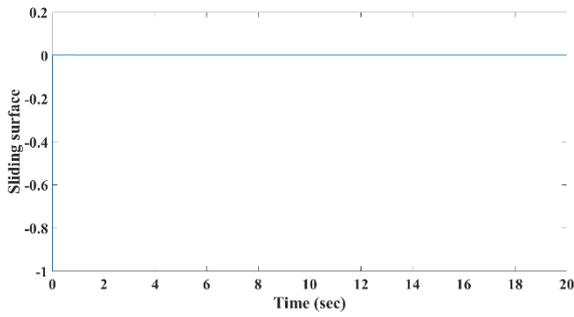


(a)

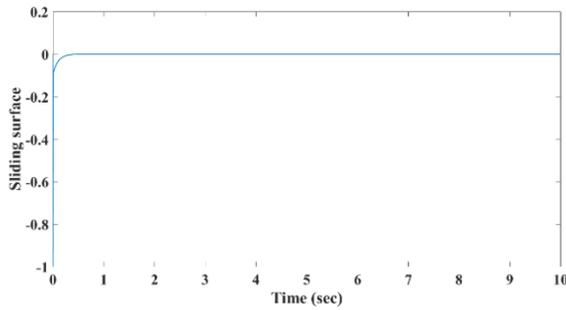


(b)

Figure 2. Vertical displacement of the hydraulic actuator vs required displacement



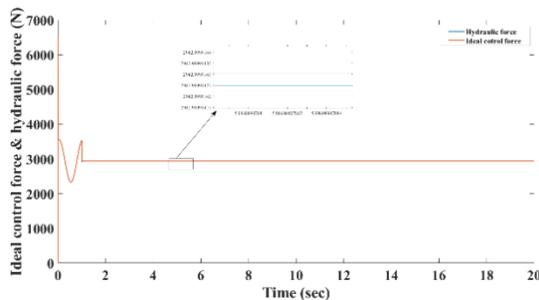
(a)



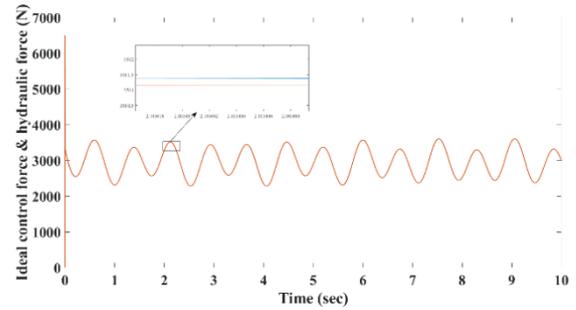
(b)

Figure 3. sliding surfaces

As mentioned in the mathematical modeling, the hydraulic model is added to the system by nonstandard backstepping with sliding mode. Fig. (5) presented the ideal control force which was designed to satisfy the suspension system requirements with actual control force i.e. hydraulic force. This Figure shows the high convergence between them over all simulation time which indicated the powerful of the nonstandard backstepping for the inclusion of the hydraulic force in the overall system model. The controller design ends with determination of the voltage which used as input to the servovalve. The servovalve adjusts the flow rate and pressure to the actuator chamber. Fig. (6) shows the voltage curve of the two cases of the road profiles.

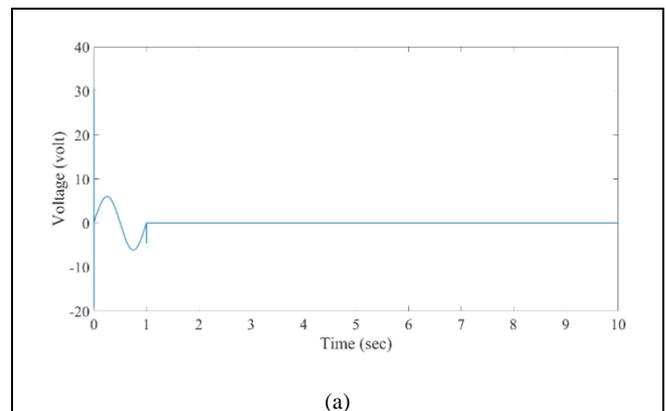


(a)

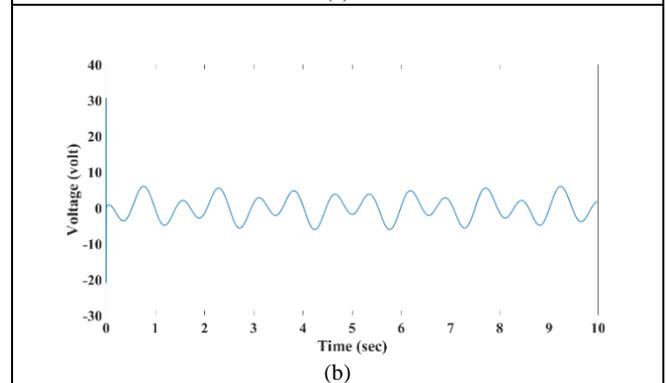


(b)

Figure 4. hydraulic force vs. ideal control force



(a)



(b)

Figure 5. input voltage of the hydraulic actuator

5. Conclusions

This work, represents a design of simulator for testing the quarter car active suspension system with generation different road profiles to simulate the actual road. The following conclusion have been drawn:

The robust ISMC gives better results in the design of virtual control force which performs all simulator requirements. The ISMC performs the stability from the first instant which considered the main feature as compared with the conventional sliding mode control. The second step of the controller design which non-standard backstepping control has good convergence between the virtual control force and the actual hydraulic control force.

REFERENCES

- [1] J. Liu, X. Chen, J. Gao, and X. Zhang, "Multiple-source multiple-harmonic active vibration control of variable section cylindrical structures: A numerical study," *Mech. Syst. Signal Process.*, vol. 81, pp. 461–474, 2016, doi: <https://doi.org/10.1016/j.ymsp.2016.02.065>.
- [2] Y. Xu, H. Hua, and J. Han, "Modeling and controller design of a shaking table in an active structural control system," *Mech. Syst. Signal Process.*, vol. 22, no. 8, pp. 1917–1923, 2008, doi: <https://doi.org/10.1016/j.ymsp.2008.02.003>.
- [3] G. Shen, S.-T. Zheng, Z.-M. Ye, Q.-T. Huang, D.-C. Cong, and J.-W. Han, "Adaptive inverse control of time waveform replication for electrohydraulic shaking table," *J. Vib. Control*, vol. 17, no. 11, pp. 1611–1633, Nov. 2010, doi: [10.1177/1077546310380431](https://doi.org/10.1177/1077546310380431).
- [4] W. Shen, J. Wang, and S. Wang, "The control of the electro-hydraulic shaking table based on dynamic surface adaptive robust control," *Trans. Inst. Meas. Control*, vol. 39, no. 8, pp. 1271–1280, Mar. 2016, doi: [10.1177/0142331216636188](https://doi.org/10.1177/0142331216636188).
- [5] H. E. Merritt, *Hydraulic control systems*. John Wiley & Sons, 1980.
- [6] N. Niksefat and N. Sepehri, "Design and experimental evaluation of a robust force controller for an electro-hydraulic actuator via quantitative feedback theory," *Control Eng. Pract.*, vol. 8, no. 12, pp. 1335–1345, 2000, doi: [https://doi.org/10.1016/S0967-0661\(00\)00075-7](https://doi.org/10.1016/S0967-0661(00)00075-7).
- [7] S. A. Ali, A. Christen, S. Begg, and N. Langlois, "Continuous–Discrete Time-Observer Design for State and Disturbance Estimation of Electro-Hydraulic Actuator Systems," *IEEE Trans. Ind. Electron.*, vol. 63, no. 7, pp. 4314–4324, 2016, doi: [10.1109/TIE.2016.2531022](https://doi.org/10.1109/TIE.2016.2531022).
- [8] I.-Y. LEE, T.-H. KIM, K.-M. Kim, and A. KITAGAWA, "A New Linearized Equation for Modelling a Servo Valve in Hydraulic Control Systems," *Proc. JFPS Int. Symp. Fluid Power*, vol. 2002, no. 5–3, pp. 899–904, 2002, doi: [10.5739/isfp.2002.899](https://doi.org/10.5739/isfp.2002.899).
- [9] D. H. Kim and T.-C. Tsao, "A Linearized Electrohydraulic Servo Valve Model for Valve Dynamics Sensitivity Analysis and Control System Design," *J. Dyn. Syst. Meas. Control*, vol. 122, no. 1, pp. 179–187, Feb. 1998, doi: [10.1115/1.482440](https://doi.org/10.1115/1.482440).
- [10] G. P. Liu and S. Daley, "Optimal-tuning nonlinear PID control of hydraulic systems," *Control Eng. Pract.*, vol. 8, no. 9, pp. 1045–1053, 2000, doi: [10.1016/S0967-0661\(00\)00042-3](https://doi.org/10.1016/S0967-0661(00)00042-3).
- [11] J. E. Bobrow and K. Lum, "Adaptive, high bandwidth control of a hydraulic actuator," *Proc. Am. Control Conf.*, vol. 1, pp. 71–75, 1995, doi: [10.1109/acc.1995.529210](https://doi.org/10.1109/acc.1995.529210).
- [12] X. Yuan, Z. Fang, and H. Li, "Research on fuzzy control for the electro-hydraulic servo system," *Proc. - 2010 7th Int. Conf. Fuzzy Syst. Knowl. Discov. FSKD 2010*, vol. 2, no. Fskd, pp. 918–921, 2010, doi: [10.1109/FSKD.2010.5569123](https://doi.org/10.1109/FSKD.2010.5569123).
- [13] P. Nakkarat and S. Kuntanapreeda, "Observer-based backstepping force control of an electrohydraulic actuator," *Control Eng. Pract.*, vol. 17, no. 8, pp. 895–902, 2009, doi: [10.1016/j.conengprac.2009.02.011](https://doi.org/10.1016/j.conengprac.2009.02.011).
- [14] M. Engineering, "Technical note Sliding mode control for a class of hydraulic position servo," pp. 0–12.
- [15] K. Guo, J. Wei, J. Fang, R. Feng, and X. Wang, "Position tracking control of electro-hydraulic single-rod actuator based on an extended disturbance observer," *Mechatronics*, vol. 27, pp. 47–56, 2015, doi: [10.1016/j.mechatronics.2015.02.003](https://doi.org/10.1016/j.mechatronics.2015.02.003).
- [16] V. I. Utkin and H. C. Chang, "Sliding mode control on electro-mechanical systems," *Mathematical Problems in Engineering*, vol. 8, no. 4–5, pp. 451–473, 2002, doi: [10.1080/10241230306724](https://doi.org/10.1080/10241230306724).
- [17] B. Yao, F. Bu, J. Reedy, and G. T.-. Chiu, "Adaptive robust motion control of single-rod hydraulic actuators: theory and experiments," *IEEE/ASME Trans. Mechatronics*, vol. 5, no. 1, pp. 79–91, 2000, doi: [10.1109/3516.828592](https://doi.org/10.1109/3516.828592).
- [18] I. Fialho and G. J. Balas, "Road adaptive active suspension design using linear parameter-varying gain-scheduling," *IEEE Trans. Control Syst. Technol.*, vol. 10, no. 1, pp. 43–54, 2002, doi: [10.1109/87.974337](https://doi.org/10.1109/87.974337).
- [19] Deshpande, V. S., Bhaskara, M., & Phadke, S. B. (2012). Sliding mode control of active suspension systems using a disturbance observer. Proceedings of IEEE International Workshop on Variable Structure Systems, 70–75. <https://doi.org/10.1109/VSS.2012.6163480>
- [20] E.-S. Kim, "Nonlinear indirect adaptive control of a quarter car active suspension," in *Proceeding of the 1996 IEEE International Conference on Control Applications IEEE International Conference on Control Applications held together with IEEE International Symposium on Intelligent Control*, 1996, pp. 61–66, doi: [10.1109/CCA.1996.558605](https://doi.org/10.1109/CCA.1996.558605).
- [21] Y. Shtessel, C. Edwards, L. Fridman, and A. Levant, *Sliding Mode Control and Observation*, Springer S. Springer Science+Business Media New York 2014, 2014.
- [22] G. B. F. P. Usai, *Modern Sliding Mode Control Theory*, First. Verlag Berlin Heidelberg: Springer, Berlin, Heidelberg, 2008.
- [23] P. Kokotović and M. Arcak, "Constructive nonlinear control: a historical perspective," *Automatica*, vol. 37, no. 5, pp. 637–662, 2001, doi: [https://doi.org/10.1016/S0005-1098\(01\)00002-4](https://doi.org/10.1016/S0005-1098(01)00002-4).
- [24] M. H. Mishary, "Sliding Mode Observer for States and Perturbation," university of technology, 2016.
- [25] V. S. Deshpande, B. Mohan, P. D. Shendge, and S. B. Phadke, "Disturbance observer based sliding mode control of active suspension systems," *J. Sound Vib.*, vol. 333, no. 11, pp. 2281–2296, 2014, doi: [10.1016/j.jsv.2014.01.023](https://doi.org/10.1016/j.jsv.2014.01.023).