Adaptive Fuzzy Super – Twisting Sliding Mode Controller optimized by ABC for Vehicle Suspension System

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Abstract- **In this paper, a second order Sliding Mode Controller (SMC), based on Super – Twisting algorithm, Fuzzy estimator and PID controller is presented for quarter vehicle active suspensions. Because of the chattering that appeared at the output of the system when using first order SMC, second order SMC is preferred. The proposed controller has been derived in order to achieve the convergence and the stability of the system that can improve the comfortable driving and vehicles safety against different road disturbances. The Artificial Bee Colony optimization method has been utilized to find the optimal values of the proposed controller parameters.**

The obtained results of the simulations have been verified the efficiency and the ability of the proposed control scheme to suppress the oscillations and give the stability of the suspension system in the presence of uncertainty and different road disturbances.

Index Terms—**Super - Twisting Sliding Mode Controller, Quarter vehicle model, Active suspension system, PID Controller, Fuzzy estimator, ABC Algorithm.**

I. INTRODUCTION

Suspension system represents an essential part in any vehicle since it separates the body of the vehicle from road disturbances. The suspension system improves the ride quality, maintains the wheels to contact with the road, and enhances the vehicle stability. The components of the suspension system are shock absorbers, the springs, and linkages with the wheels [1].

The suspension system involves three categories, passive, semi - active and active suspension system. The spring and damper are the components of the passive suspension system, where the damper is responsible of dissipation the energy and the spring is responsible of storing these energy. In this kind of suspension system, the coefficient of damping and the stiffness of spring represent a fixed parameters so this acts a weakest point in providing comfortable driving and good handling for the road disturbances [2] . The semi - active suspension system consists also from the same components as the passive suspension, but the coefficient of damping is a variable parameter and can be controlled. During road tests, it has been observed the high frequency hardness feature, so that this suspension system is not convenient to handle this property [3]. In order to achieve tradeoff between comfortable driving, the deflection of suspension and tire, the active suspension system is used since it is the more efficient than the other types. The active suspension system has a hydraulic actuator which dissipates energy from the system, sensors which measure suspension variables, and control unit that can enter desired force of the suspension system based on the passive components. Also active suspension systems provide better dealing with road and increase drive quality [4].

Many researches have been performed to propose a robust controller for active suspension system. In [5], a PI Sliding Mode Control scheme was proposed to deal with the effect of the and uncertainties of the suspension system, where the effect of the selection of the layer thickness in the proposed controller was also presented to satisfy the control stability requirements for the quarter vehicle active suspension system. In [6], Genetic Algorithm based Fuzzy Sliding Mode Controller was designed to tune Distance to the quarter vehicle suspension system, increase the speed response of system and reduce the chattering in the sliding phase in order to establish the vibration isolation of the vehicle body. The weight coefficients of the Sliding Mode Controller (SMC) optimized with genetic algorithm was presented to manipulate the force to improve the control stability requirements of the quarter vehicle active suspension system [7]. Super - Twisting Sliding Mode Controller (STSMC) was presented in quarter vehicle active suspension system to show the effectiveness in vibration isolation of acceleration of passenger seat and responses of vertical displacement when compared with passive suspension system [8]. In [9], STSMC was designed for a quarter vehicle active suspensions to overcome the chattering that appeared in first order SMC. Numerical outcomes showed that the performance of the proposed high order sliding mode controller improve the comfortable driving of passengers.

PID control scheme can be used widely, since it possesses an important function in control applications. But, this control scheme is insensitive to the changes of parameters. The PID controller tuned by particle Swarm optimization method was used for quarter active suspension system and compared with passive suspension system for various road profiles. Also PID controller tuned with Artificial Bee Colony (ABC) optimization technique was used for non – linear full (active suspension system) with traveler seat [10], [11], [12]. Fuzzy logic controller is a robust controller and utilized in many control applications. Fuzzy system can be used to estimate the unknown parameters of the systems [13]. SMC is a robust controller and widely used because of its attractive characteristics of robustness to uncertainties of the system and finite-time convergence. SMC has been applied both theoretical and practical, but it was suffered the chattering problem [14], [15]. Super-Twisting Algorithm represents a well – known second order sliding mode algorithm which is introduced by Levant [16]. STSMC scheme was used to prevent the chattering problem [17].

This paper presents a proposed controller for a quarter vehicle active suspension system to decrease the effects of disturbances on the vehicle body, as well as provide a comfortable driving and better handling with the different road profiles. An Adaptive Fuzzy STSMC to design an effective control strategy is proposed. The Artificial Bee Colony (ABC) algorithm is utilized to give the optimal values of the proposed controller parameters. The hydraulic actuator dynamics have been considered. The PID Controller is used in order to control the force of hydraulic actuator. The STSMC will prevent the chattering and the fuzzy system will estimate the unknown parameters of the system. Simulation results using MATLAB program and SIMULINK toolbox show the superior efficiency of the proposed control scheme in a comparison with passive suspension system.

Section 2 gives the mathematical model of the suspension system. Section 3 presents the proposed controller. In Section 4, the simulation results of the controller are presented. Section 5 shows the conclusion of this paper.

II. MATHEMATICAL MODEL OF QUARTER VEHICLE

In this study, to test the goodness of the proposed control strategy, an active suspension system of quarter vehicle model is used. Many researchers are used this model due to its simplicity and also it gives a good representation for the actual suspension system. The passive components with a hydraulic actuator forms the active suspension systems as appeared in Fig. 1. The non – linear dynamics of a quarter vehicle suspension system are described as:

$$M_s\ddot{x}_s + K_s(x_s - x_u) + C_s(\dot{x}_s - \dot{x}_u) - F_a = 0 \tag{1}$$

$$M_u\ddot{x}_s + K_t(x_u - x_r) - K_s(x_s - x_u) - C_s(\dot{x}_s - \dot{x}_u) + F_a \tag{2}$$

Where,

M_s: Sprung mass of the vehicle body.
K_s: Suspension system linear stiffness.
x_s: Displacement of the vehicle body.
x_u: Displacement of the vehicle tire.
C_s: Linear damping of the suspension system.
F_a: Hydraulic actuator force.
x_r: Road profile.
M_u: Unsprung mass of the vehicle body.
K_t: Linear stiffness of the tire.

The hydraulic actuator consists of a system with four valve piston which is connected in parallel with the passive components of the suspension system and controlled by electro-hydraulic servo-valves. The actuator force can be described as:

$$F_a = A_p P_{sup} \tag{3}$$

The piston area is (A_p) and the pressure of hydraulic supply is (P_{sup}). The dynamic displacement of the spool valve system is modeled as [18]:

$$\dot{x}_v = \frac{1}{\tau}(-x_v + K_c u_{mi}) \tag{4}$$

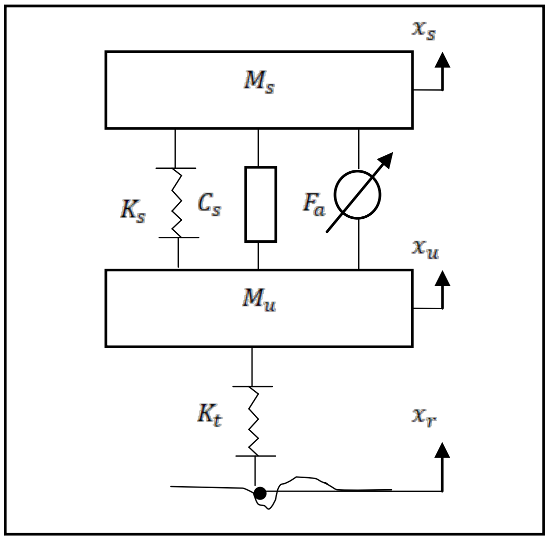

Fig. 1 Quarter vehicle model (Active suspension system)

The time constant for servo-valve is (τ), which represents a mechanical delay time, (K_c) is a gain and (u_{mi}) is the control input. The hydraulic cylinder is described by the dynamic equation:

$$\dot{F}_a = -\sigma A_p^2(\dot{x}_s - \dot{x}_u) - \beta F_a + \gamma_s A_p x_v \sqrt{P_{sup} - \frac{F_a sgn(x_v)}{A_p}} \tag{5}$$

The parameters (σ, β and γ_s) represent the hydraulic actuator parameters which may be time varying. The state space form of the active suspension system (quarter vehicle model) with hydraulic dynamic can be written as:

$$\dot{x} = Ax + BF_a + B_w x_r \tag{6}$$

Consider, the body displacement is $x_1 = x_s$, the velocity of the body is $x_2 = \dot{x}_s$, the unsprung mass displacement is $x_3 = x_u$ and the velocity of the unsprung mass is $x_4 = \dot{x}_u$, so:

$$\dot{x}_1 = x_2 \tag{7}$$

$$\dot{x}_2 = -\frac{K_s}{M_s}x_1 - \frac{C_s}{M_s}x_2 + \frac{K_s}{M_s}x_3 + \frac{C_s}{M_s}x_4 + \frac{1}{M_s}F_a \tag{8}$$

$$\dot{x}_3 = x_4 \tag{9}$$

$$\dot{x}_4 = \frac{K_s}{M_u}x_1 + \frac{C_s}{M_u}x_2 - \frac{(K_s+K_t)}{M_u}x_3 - \frac{C_s}{M_u}x_4 + \frac{K_t}{M_u} - \frac{1}{M_u}F_a \tag{10}$$

The state, control input, and road input matrices are (A,B and B_w) and given as:

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{K_s}{M_s} & -\frac{C_s}{M_s} & \frac{K_s}{M_s} & \frac{C_s}{M_s} \\ 0 & 0 & 0 & 1 \\ \frac{K_s}{M_u} & \frac{C_s}{M_u} & -\frac{(K_s+K_t)}{M_u} & \frac{K_t}{M_u} \end{bmatrix}, B = \begin{bmatrix} 0 \\ \frac{1}{M_s} \\ 0 \\ -\frac{1}{M_u} \end{bmatrix}, B_w = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{K_t}{M_u} \end{bmatrix} \tag{11}$$

The dynamic equation of sprung mass is given by:

$$\ddot{x}_1 = f(x,t) + \frac{1}{M_u} F_a \quad (12)$$

The function $f(x,t)$ is assumed unknown function and to be estimated.

$$f(x,t) = -\frac{K_s}{M_s}x_1 - \frac{C_s}{M_s}x_2 + \frac{K_s}{M_s}x_3 + \frac{C_s}{M_s}x_4 \quad (13)$$

III. THE PROPOSED CONTROLLER DESIGN

In this study, the proposed control scheme has been designed as shown in Fig. 2. It consists of two loops, the outer loop which involves the Fuzzy Super Twisting Sliding Mode Controller (FSTSMC) scheme and PID controller in the inner loop.

The control signal of SMC can be defined as:

$$u_c = u_{eq} + u_{sw} \quad (20)$$

The equivalent control u_{eq} works to keep the variable on the sliding surface with no regard to the effect of the uncertainty of the system and disturbances and it is derived when, $\dot{s} = 0$. The switching control u_{sw} is derived when, $s\dot{s} < 0$ and adopted as:

$$u_{sw} = -K sgn(s) \quad (21)$$

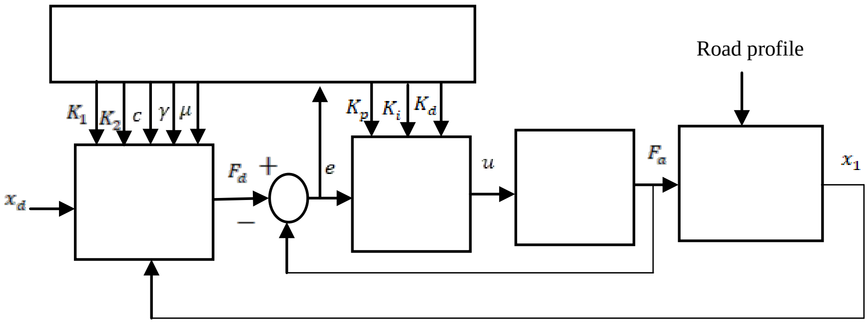

Fig. 2 Block Diagram of Proposed Control Scheme

A. Super – Twisting Sliding Mode Control Design

The sliding mode control (SMC) involves two sections, the first one is responsible of dealing with the system dynamics and the sliding surface and the second one represents the switching control which saves the system dynamics onto sliding surface. The system controller with SMC is inefficient toward the changes of the uncertain parameters and the applied disturbances on the sliding surface, so many researchers have been interested in this feature. The dynamic equation of sprung mass which is given in (12) can be written as:

$$\dot{x}_1 = x_2 \quad (14)$$

$$\dot{x}_2 = f(x,t) + u_c \quad (15)$$

Where, the function $f(x,t)$ is assumed unknown function and to be estimated, the states are x_1 and x_2 , and the control signal is u_c.The sliding surface can be defined as:

$$s = \dot{e}_r + ce_r \quad (16)$$

Where the sliding coefficient is c, and $c > 0$, the tracking error and its derivative are e_r and $\dot{e}_r$ respectively. They are given as:

$$e_r = x_1 - x_d \quad (17)$$

$$\dot{e}_r = \dot{x}_1 - \dot{x}_d \quad (18)$$

The desired signal and the actual signal are x_d which equals to zero and x_1 respectively. The function of sliding mode is chosen as:

$$\dot{s} = \ddot{e}_r + c\dot{e}_r = \ddot{x}_1 - \ddot{x}_d + c\dot{e}_r = f(x,t) + u_c + c\dot{x}_1 \quad (19)$$

Where K is a positive constant.

The purpose of any control design is to provide a proper control law, so that the system output can quickly track the required trajectory in a specified time. In order to solve the problem of tracking taking into account the existence of external disturbances and unknown parameters of the system, a Super – Twisting Sliding Mode Control (STSMC) scheme is presented. This strategy represents a good method to overcome the unknown parameters and the uncertainty of the system.

Also, the serious problem of the control design is the chattering in the output in high frequency, so to limit and reduce the chattering with retention of the conventional SMC advantages a higher order SMC is produced. STSMC is a second order represents a robust scheme which can eliminate the chattering with keeping other properties of SMC. STSMC consists of two sections the first one represents the discontinuous function of the sliding variable and the other one represents the continuous function of the derivative of the sliding variable, and the switching control can be defined as:

$$u_{sw} = -K_1\sqrt{|s|}sgn(s) + m \quad (22)$$

$$\dot{m} = -K_2 sgn(s) \quad (23)$$

The parameters (K_1 and K_2) are positive constant. Based on the STSMC design, the switching controller can be given as:

$$u_{sw} = -K_1\sqrt{|s|}sgn(s) - \int K_2 sgn(s)dt \quad (24)$$

The final control law can be defined as:

$$u_c = -f(x,t) - c\dot{x}_1 - K_1\sqrt{|s|}sgn(s) - \int K_2 sgn(s)dt \quad (25)$$

B. Fuzzy Super – Twisting Sliding Mode Control Design

Fuzzy system is an effective control method and utilized in many practical applications [19], [20], [21], [22], [23]. It is used as an estimator to estimate the unknown parameters of the system and STSMC is used to limit the chattering and the nonlinearity of the system in order to design an adaptive control technique. The Minimum Parameter Learning (MPL) technique is a self-learning method for the parameters in the system [24]. This method uses fuzzy estimator in order to estimate the unknown and uncertain nonlinear parts of the system. The systems can be considered uncertain system, where the dynamics of the system are varying under different conditions. Adaptive Fuzzy Super – Twisting Sliding Mode Control (FSTSMC) represents a good way to adjust the controlled systems. This is essential for the unknown system with altering dynamics. For the unknown function we will change $f(x,t)$ with $\hat{f}(x,t)$ to realize the control law. The fuzzy system output is defined by [24]:

$$\hat{f}(x|\theta_f) = \frac{\sum_{l_1=1}^{5}\sum_{l_2=1}^{5}\sum_{l_3=1}^{5}\sum_{l_4=1}^{5} y_f^{l_1 l_2 l_3 l_4}\left(\prod_{i=1}^{4}\mu_{A_i}^{l_i}(x_i)\right)}{\sum_{l_1=1}^{5}\sum_{l_2=1}^{5}\sum_{l_3=1}^{5}\sum_{l_4=1}^{5}\left(\prod_{i=1}^{4}\mu_{A_i}^{l_i}(x_i)\right)} \tag{26}$$

Where; $\mu_{A_i}^{l_i}(x_i)$ is the membership function of x_i, which is a Gaussian function whose its parameters will not alter over the time and defined as:

$$\mu_{NM}(x_i) = \exp[-((x_i + \tfrac{\pi}{3})/(\tfrac{\pi}{12}))^2],$$

$$\mu_{NS}(x_i) = \exp[-((x_i + \tfrac{\pi}{6})/(\tfrac{\pi}{12}))^2],$$

$$\mu_{Z}(x_i) = \exp[-(x_i/(\tfrac{\pi}{12}))^2],$$

$$\mu_{PS}(x_i) = \exp[-((x_i - \tfrac{\pi}{6})/(\tfrac{\pi}{12}))^2],$$

and $\mu_{PM}(x_i) = \exp[-((x_i - \tfrac{\pi}{3})/(\tfrac{\pi}{12}))^2]$, $y_f^{l_1 l_2 l_3 l_4}$ be a parameter in the group $\hat{\theta}_f \in Rule^{(625)}$, and the fuzzy rules is a column vector for $p_1 = p_2 = p_3 = p_4 = 5$.

$$\xi(x) = \prod_{i=1}^{4} p_i = p_1 \times p_2 \times p_3 \times p_4 = 625.$$

$Rule^{(1)}$: If $x_1$is A_1^1 and x_2 is A_2^1 and x_3 is A_3^1 and x_4 is A_4^1 then $\hat{f}$ is B^1, to…

$Rule^{(625)}$: If $x_1$is A_1^5 and x_2 is A_2^5 and x_3 is A_3^5 and x_4 is A_4^5 then $\hat{f}$ is B^{625}.

The fuzzy sets are $A_1^{l_i}, A_2^{l_i}$, $A_3^{l_i}$, and $A_4^{l_i}$ respectively, $l_i = 1,2,3,4,5$. The equation (26) can be replaced as:

$$\hat{f}(x|\theta_f) = \hat{\theta}^T \xi(x) \tag{27}$$

$$\xi_{l_1 l_2 l_3 l_4}(x) = \frac{\prod_{i=1}^{4}\mu_{A_i^j}(x_i)}{\sum_{l_1=1}^{p_1}\sum_{l_2=1}^{p_2}\sum_{l_3=1}^{p_3}\sum_{l_4=1}^{p_4}\left(\prod_{i=1}^{4}\mu_{A_i^j}(x_i)\right)} \tag{28}$$

Define θ_f^* as a parameter for minimum error as:

$$\theta_f^* = arg \min_{\theta_f \in \Omega_f}\left[\sup_{x \in R^n} |\hat{f}(x|\ \theta_f\ ) - f(x)|\right] \tag{29}$$

Where, Ω_f is a constraint set of θ_f. The term (f) can be described as:

$$f = \theta_f^{*T}\xi(x) + \varepsilon \tag{30}$$

Where ε is an estimation error of the fuzzy system, approximation (f) is:

$$\hat{f}(x|\theta_f) = \hat{\theta}_f^T \xi(x) \tag{31}$$

In this work we consider $f(x,t)$ unknown function so by using Minimum Parameter Learning , express a positive constant as $\phi = \|\theta_f^*\|^2$ and let $\hat{\phi}$ be an estimate of ϕ.

The Lyapunov function will be characterized as:

$$V = \frac{1}{2}s^2 + \frac{1}{2\gamma}\tilde{\phi}^2 \tag{32}$$

Where; $\tilde{\phi} = \hat{\phi} - \phi$ and γ is positive.

$$\dot{V} = s\dot{s} + \frac{1}{\gamma}\tilde{\phi}\dot{\hat{\phi}} \tag{33}$$

The adaptive law is designed as:

$$\dot{\hat{\phi}} = \frac{\gamma}{2}s^2\xi^T\xi - \kappa\gamma\hat{\phi} \tag{34}$$

Where, $\kappa = \frac{2\mu}{\gamma}$ and, $\kappa > 0$. Solving $\dot{V} \le -2\mu V + Q$ to imply that:

$$\lim_{t\to\infty} V = \frac{\phi^2}{2\gamma} + \frac{1}{4\gamma} \tag{35}$$

Where, $Q = \frac{\kappa}{2}\phi^2 + \frac{1}{2}$

The FSTSMC can be designed as:

$$u_c = -\frac{1}{2}s\hat{\phi}\xi^T\xi - c\dot{x}_1 - K_1\sqrt{|s|}sgn(s) - \int K_2 sgn(s)dt - \mu s \tag{36}$$

Where, $\mu > 0$.

The accuracy of convergence based on (c), (γ) and (μ) parameters. The sliding surface converges to zero lastly.

C. PID Control Design

The standard form of PID controller is based on loop feedback mechanism and the estimation of an error signal, an error signal represents the difference between actual signal and a reference signal. The design of the PID controller will provide a stabilization for the system and a suitable system performance [25], also greatly used in the industrial application [10]. The equation of the PID controller is defined as:

$$u = K_p e + K_i \int_0^t e dt + K_d \frac{de}{dt} \tag{37}$$

Where; (u) is the output of PID controller, (K_p, K_i, and K_d) are the proportional, the integral and the differential gains respectively. The error is described as:

$$e = F_d - F_a \tag{38}$$

Where, F_d is the desired force and F_a is the actual hydraulic actuator force.

D. Optimization of the proposed controller parameters using ABC algorithm

The Artificial Bee Colony (ABC) algorithm has been introduced which is firstly proposed by Dervis Karaboga in 2005 [26]. ABC is an optimization algorithm based on the bee colonies behavior, where these colonies are employed or unemployed bees. The employed bees seek about the sources of food and store the data about the goodness of them. The employed bees grant and share the data with the bees of hive about the food sources position. An optimization solution for the problem represents the best food source where, the nectar quantity sources matches to the solution fitness [27]. The unemployed bees consist of scout and onlooker bees. An onlooker bees choose the best food source based on the information founded by employed bees [28], [29]. The scout bees is related with employed bees where if food source becomes deserted, the employed bees will be scout and then search about another sustenance source to become employed again.

The (ABC) method used in this work can efficiently find a suitable gain parameters for the proposed controller. It is aimed to minimize the acceleration and displacement of the vehicle body as well as the amount of suspension. The ABC algorithm can be used to achieve multi objective design criterion and the fitness function may be selected to contain one or more terms of the cost functions like:

$$cost1 = \sum_{i=1}^{N} x_{si}^2 , cost2 = \sum_{i=1}^{N} \ddot{x}_{si}^2 , cost3 = \sum_{i=1}^{N} (x_{si} - x_{ui})^2$$

The ABC technique involves the following steps [26]:

1. Initialize ABC parameters such as (maximum number of cycle, colony dimension, parameter limits).
2. Generate an initial population of Food Sources (FS) individuals randomly. Each solution K_{sou}, $K_{sou} = [1,2,\dots,FS]$ represents a D – dimensional vector corresponded to PID and Super – twisting FSMC parameters where,

$$D = [\, c \;\; \gamma \;\; \mu \;\; K_1 \;\; K_2 \;\; K_p \;\; K_i \;\; K_d \,]$$

3. For each solution, evaluate the fitness function as:

$$fitness = \frac{1}{1+cost} \tag{39}$$

4. Increase the number of cycle counter to 1. The solutions will be modified and altered with a new solution (K_{new}) by employed bees. The better solution which is allocated by each employed bee is assigned in the following equation:

$$K_{new}(s,j) = K(s,j) + \delta(s,j)(K(s,j) - K(n,j)) \tag{40}$$

Where $n = 1, 2, \dots, FS \; (n \neq s)$ and $j = 1, 2, 3, 4, 5, 6, 7, 8$ are randomly selected and $\delta(s,j) \in (0,1)$.

5. Compute the selection probability values (P_s) for each employed bee. According to these probability values, each onlooker bee produce new solution (K_{new}) as in (40). Then calculate the fitness function of each new solution as give in (39), a greedy selection is applied between the new and old solution in order to save the best solution and ignore the other one.
6. If the food source is not improved over an iterations number, that source deserted and changed with another one which is generated randomly by scout bees. This process is made according to the parameter limit.
7. Save the track of the best solution and increase the cycle counter. Steps between (4 and 7) are repeated until reach the maximum number of cycle. If the maximum numbers of iterations have satisfied, stop and return the best solution found.

IV. SIMULATION RESULTS

In this study, the quarter vehicle model of active suspension system has been discussed. The hydraulic actuator and the suspension system parameters are given in Table 1 and Table 2 respectively. The active and passive suspension systems have been compared with various types (three types) of road profiles. The proposed control system of Fuzzy Super – twisting SMC and PID controller is simulated using the MATLAB/SIMULINK.

Table I Physical parameters of the hydraulic actuator

Sample	Value
σ	$4.515 \times 10^{13} N/m$
β	$1 s^{-1}$
γ_s	$1.545 \times 10^{9} N/m$kg
P_{sup}	$10.3425 \times 10^{6} Pa$
A_p	$3.35 \times 10^{-4} m^2$
τ	$1/30\ s$
K_c	$0.001\ m/V$

Table II Physical parameters of the quarter vehicle suspension system

Sample	Value
M_s	$290\ kg$
M_u	$59\ kg$
K_s	$16812\ N/m$
K_t	$190000 N/m$
C_s	$1000\ N/m$

1. Case (1): In this state the applied road profile is a single sine bump which is taken from [30] and it is shown in Fig. 3.

$$x_{r1} = \begin{cases} \frac{0.5h(1+\sin(2\pi vt))}{L} & 0.2 \le t \le 0.7 \\ 0 & otherwise \end{cases} \quad (41)$$

Where; h = 0.05 m, is the altitude of the bump, the bump length is L = 2.5 m, and the vehicle velocity is v = 5 m/s.

2. Case (2): In this road profile, a two sine bumps is assumed as shown in Fig. 4 [31].

$$x_{r2} = \begin{cases} \frac{a(1-\cos(8\pi t))}{2} & 0.5 \le t \le 0.75 \; and \; 3 \le t \le 3.25 \\ 0 & otherwise \end{cases} \quad (42)$$

Where the bump amplitude is a, and a = 0.05.

3. Case (3): The third road profile is supposed to be a random road profile which can be approximated by Power Spectrum Density (PSD) form [32]:

$$\Phi(n) = \Phi(n_o)(\frac{n}{n_o})^{-w} \quad (43)$$

Where, n_0 is the reference spatial frequency and the spatial frequency is n. Choose the road roughness $\Phi(n_o) = 64 \times 10^{-6} m$, $n_0 = 0.127$ and $w = 2$.

The PSD represent the third road profile that is generated by integrating a white noise signal as shown in Fig. 5.
The parameters of the proposed controllers are optimized by ABC algorithm, where the input to the ABC algorithm is the error signal which is utilized to compute the fitness function and the ABC outputs represent the optimal proposed controller parameters. The ranges for the parameters for the control scheme are selected as:

K_p, K_i, and K_d ∈ [-1 500], c and μ ∈ [1 10], γ ∈ [0.001 0.1], K_1 ∈ [40 70], K_2 ∈ [0.0001 0.001].

The parameters chosen for ABC algorithm are given in Table 3. The fitness function is selected as in (39) and (40).

Table III ABC Algorithm parameters

Parameter	Value
Number of optimization parameters	8
Maximum number of iteration cycle	20
Colony dimension	100
Number of Food Source	50

Five Gaussian membership function with centers ($\mu_1 = 1$, $\mu_2 = 0.5$, $\mu_3 = 0$, $\mu_4 = -0.5$, and $\mu_5 = -1$) are used for the inputs of the fuzzy estimator as shown in Fig. 6. The optimal values of the proposed controller after optimization using ABC algorithm are determined as:

$K_p = 389.9063$, $K_i = 47.3237$ and $K_d = 287.1795$.

$c = 5.9231$, $\gamma = 0.0691$ and $\mu = 8.022$.

$K_1 = 59.4735$ and $K_2 = 0.0008$.

The Performance of the suspension system in terms of drive quality, vehicle handling and satisfy the stability requirements is observed. The different road disturbances represent the external input for the system. Active and passive suspension systems have been compared for the quarter vehicle model against three road profiles. The true and estimated forces of control input are given in Fig. 7, Fig. 11, and Fig. 15, where, these figures showed that the estimated force of control input are converged to the true force of control input. The vertical displacement of the vehicle body is shown in Fig. 8, Fig. 12, and Fig. 16, where these figures showed that the transient amplitude of the sprung mass displacement responses are improved, effectively minimized and lastly reached to zero by proposed control method. The suspension deflection is shown in Fig. 9, Fig. 13, and Fig. 17, which are reduced and well improved. The body acceleration is shown in Fig. 10, Fig. 14 and Fig. 18, which are also, gave a good results. So, the FSTSMC strategy has a superior performance and gives a best dynamic properties with minimum chattering at the output responses.

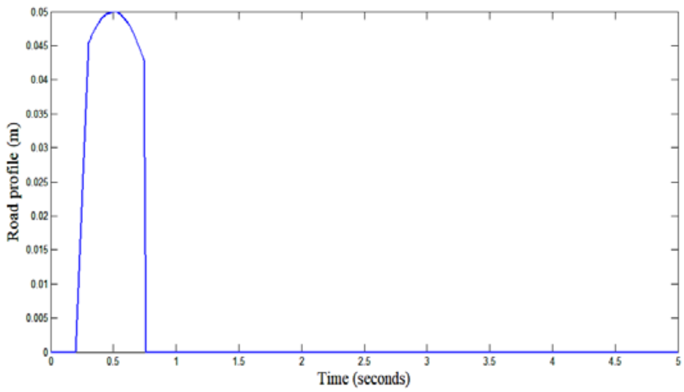

Fig. 3 Road profile 1

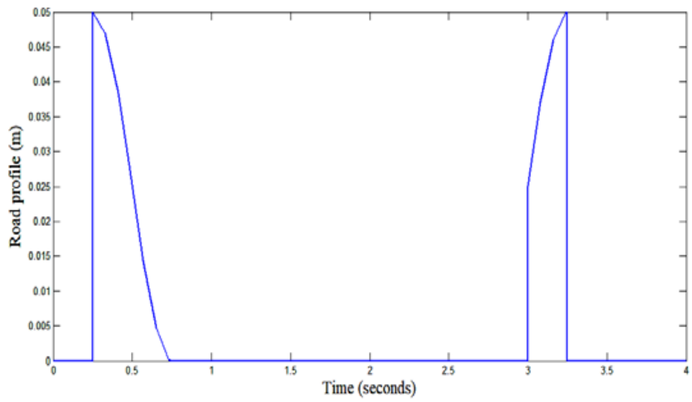

Fig. 4 Road profile 2

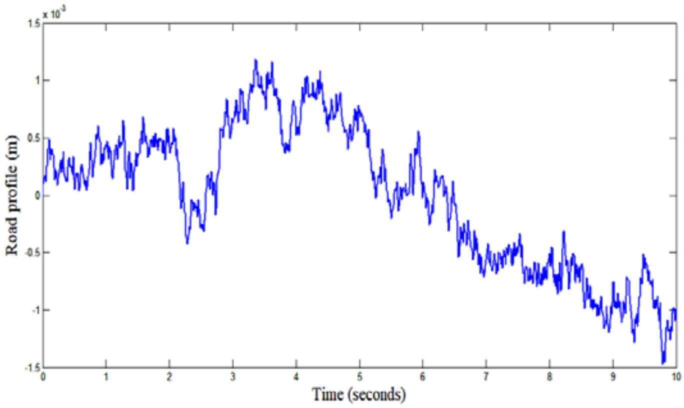

Fig. 5 Road profile 3

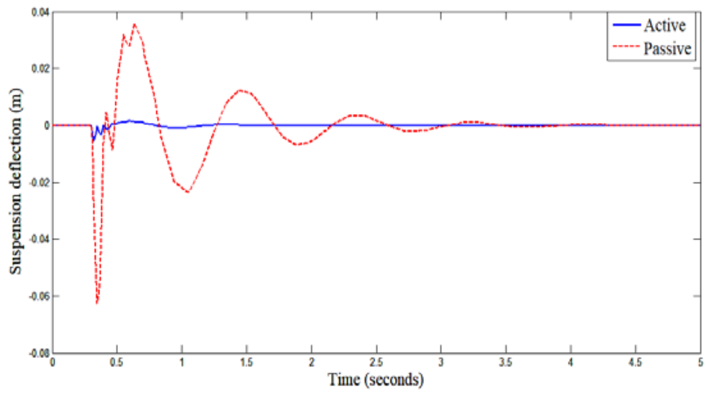

Fig. 9 Suspension Deflection (Case 1)

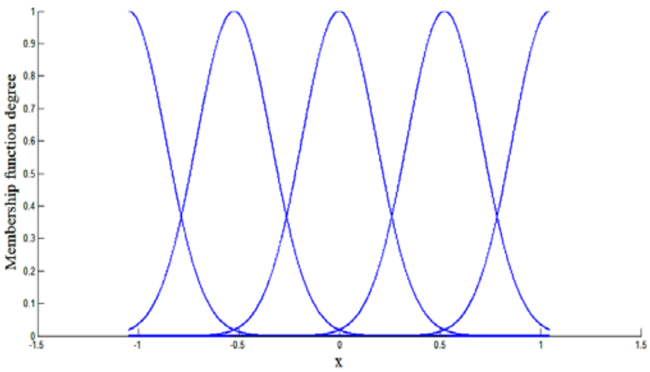

Fig. 6 Membership function of x_i

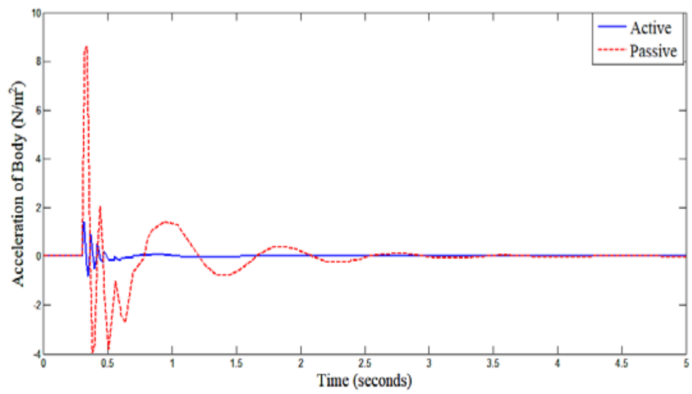

Fig. 10 Body Acceleration (Case 1)

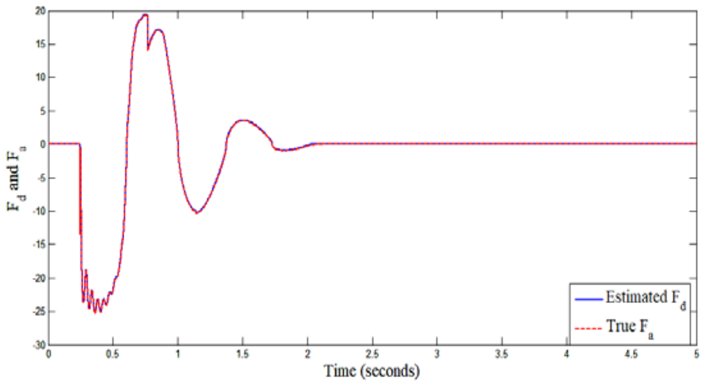

Fig. 7 True and Estimated forces (Case 1)

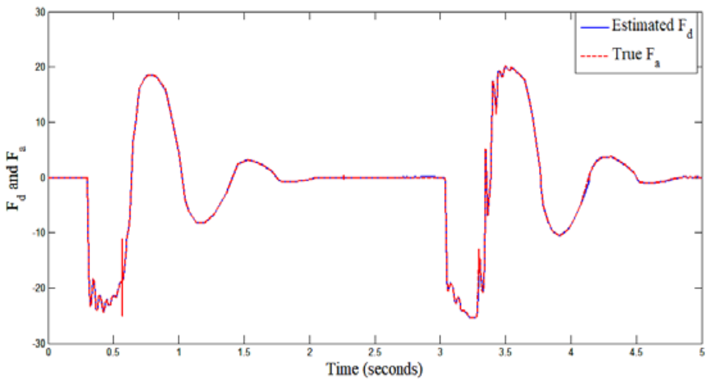

Fig. 11 True and Estimated forces (Case 2)

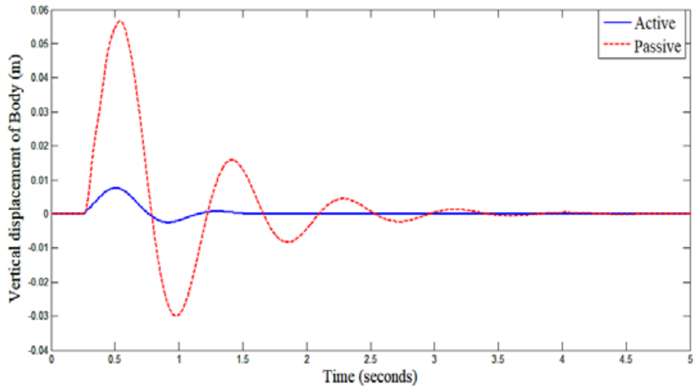

Fig. 8 Body Displacement (Case 1)

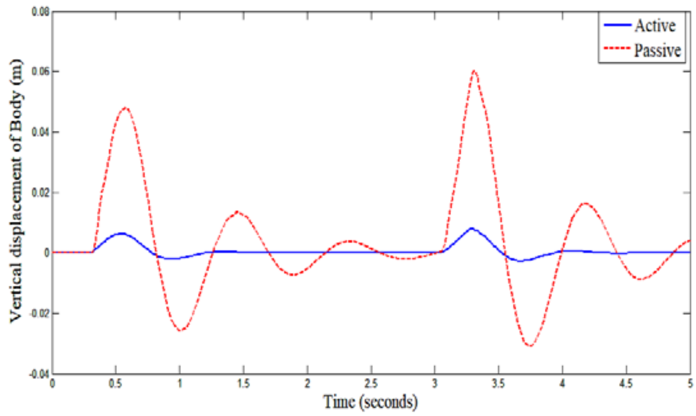

Fig. 12 Body Displacement (Case 2)

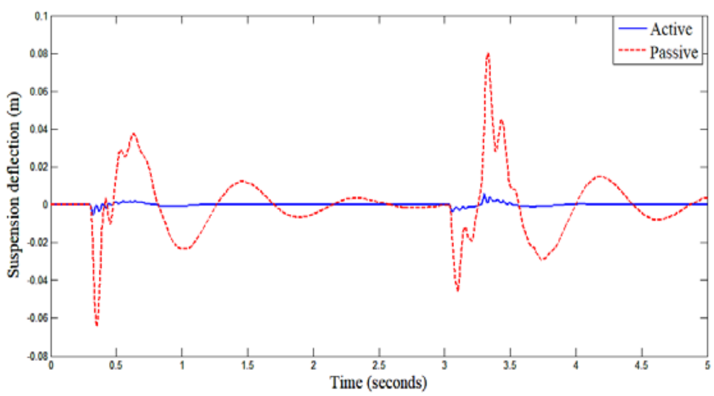

Fig. 13 Suspension Deflection (Case 2)

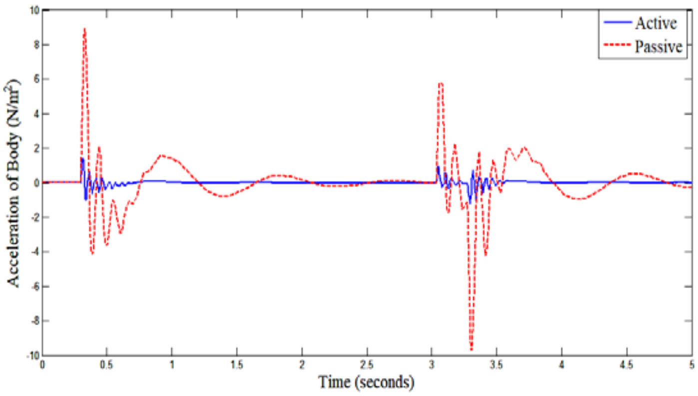

Fig. 14 Body Acceleration (Case 2)

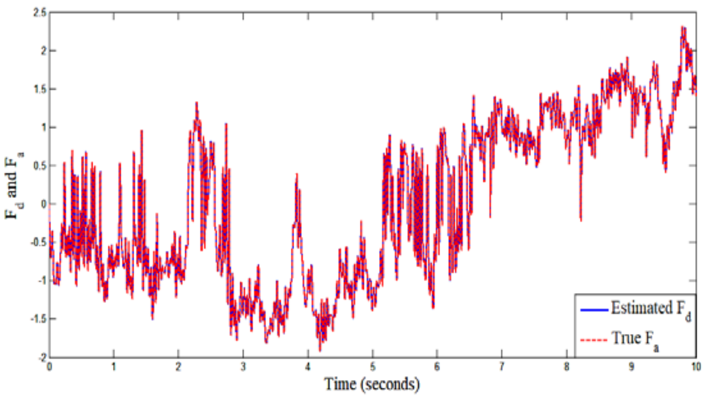

Fig. 15 True and Estimated forces (Case 3)

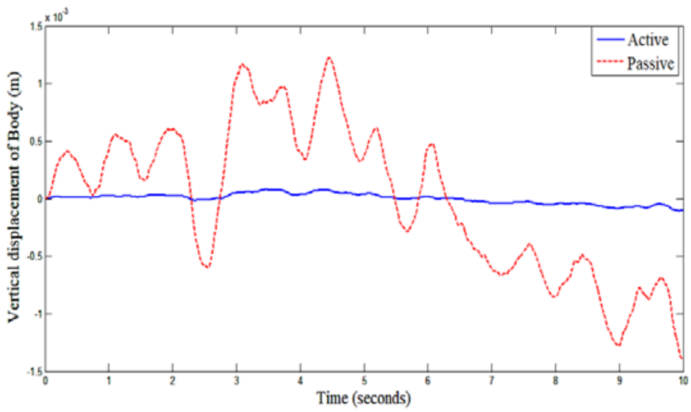

Fig. 16 Body Displacement (Case 3)

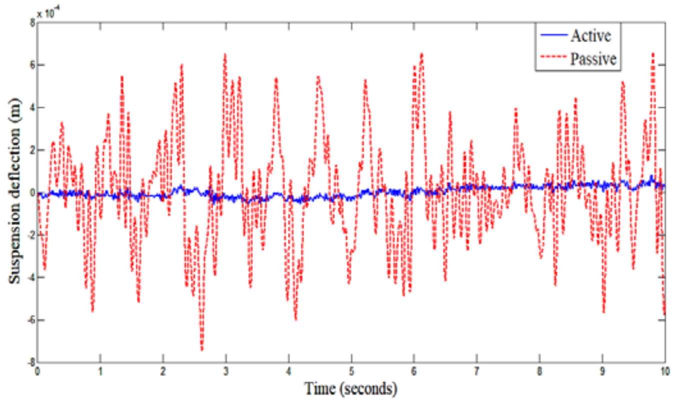

Fig. 17 Suspension Deflection (Case 3)

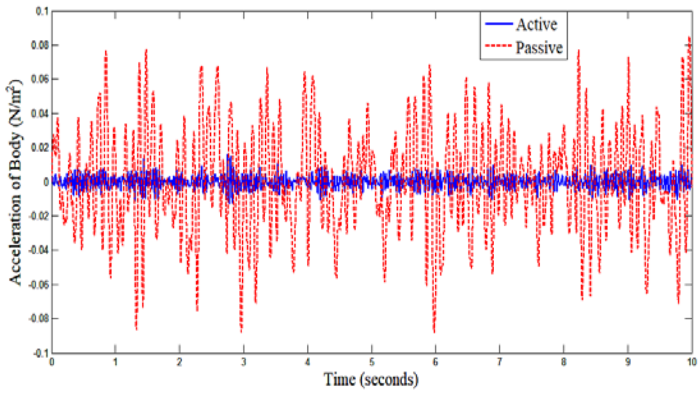

Fig. 18 Body Acceleration (Case 3)

V. CONCLUSION

This work demonstrates the validity of using an optimized super – twisting sliding mode controller with fuzzy estimator for controlling the quarter vehicle. The proposed control scheme gives good results when compared with passive system. Results show that the vertical deflection and acceleration are extremely reduced. A proposed control strategy which combines FSTSMC with PID controller is designed. All parameters of the proposed control strategy are optimized with ABC algorithm. Also the obtained results show that the proposed control scheme can give a better performance of the vehicle and minimize the transient responses amplitude of the suspension system under different road profiles. Obtained results show that the proposed control scheme that use second order SMC gives less chattering with less control effort than SMC, and this is more acceptable since high chattering may harm the actuator.

VI. REFERENCES

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