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Design of Fuzzy Super Twisting Sliding Mode Control Scheme for Unknown Full Vehicle Active Suspension Systems Using an Artificial Bee Colony Optimization Algorithm

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Abstract

This article proposes a new intelligent control scheme that uses the Fuzzy Super Twisting Sliding Mode Concept (FSTSMC) and PID controller tuned with the Artificial Bee Colony (ABC) algorithm to control a full vehicle active suspension system with new convergence proof. Suspension systems are utilized to provide vehicle safety and improve comfortable driving. The effects of road roughness transmitted by tires to the vehicle body can be reduced by using suspension systems. In this work super twisting sliding mode is combined with a fuzzy system to design a robust control method. The super twisting sliding mode concept is utilized to limit and minimize the chattering problem and the fuzzy system is used for estimating the unknown parameters and uncertainty in the suspension system components (spring, damper, and actuator). The advantage of such combination is that it can handle the uncertainties and nonlinearities efficiently. The PID controller is used to create the required force to be produced by the actuator. The proposed control scheme consists of four similar sub control schemes, one for each of the four corners of the vehicle. All parameters in the sub control schemes are optimized using the ABC algorithm. The designed control system is applied for a full vehicle model with 8 degrees of freedom. Simulation results show large reduction in the vibration of the vehicle body when passing on disturbance and also show good robustness properties when tested for different road conditions. Passive and active suspension systems using sliding modecontrol (SMC) and the proposed FSTSMC are compared to test the efficiency and the ability of the proposed control scheme to achieve safe and comfortable driving for a random road profile.

KEYWORDS

ABC algorithm, full vehicle active suspensions, fuzzy estimator, PID controller, super twisting sliding mode control

1 | INTRODUCTION

The suspension system of any vehicle is a system that isolates the body of the vehicle and its wheels from external vibration. The primary objectives of suspension systems are to provide comfortable driving and system stabilization under different road conditions, and to reduce the deflection and vertical acceleration transmitted into the cabin of the driver and passengers [1–4]. The quality of suspension systems depends on the kinds of both the scheme of control and the suspension system. The passive suspension system possesses fixed stiffness and damping elements (spring and damper). The spring keeps the energy and the damper dissipates it. Some issues with passive suspension systems include that their elements are unchanged by mechanical parts employed to damp unwanted vibrations. The transmitted vibration of passive systems is low in the range of low frequencies [5].

Semi-active suspension consists of the parameters of the passive suspension system itself, with the exception of the coefficient of damping, which is a variable parameter that can be controlled. The semi-active suspension system can limit transmitted vibration across a range of low and high frequencies [6].

It is more convenient to use an active suspension system to limit the problems of passive and semi-active suspension systems. The active suspension system consists of spring and damper with the addition of an electromagnetic actuator or hydraulic actuator commonly situated in parallel with damper and spring [7,8]. The actuator dissipates the energy from the system, sensors measure suspension variables, and the control unit can enter the desired force. The active suspension is utilized to minimize the effects of road disturbances and road profiles do not alter the efficiency of the active suspension system. Active suspension systems increase driving quality [9].

Many control strategies related to active suspension systems have been designed to improve driving comfort and the capability of road holding against various disturbances [10–15]. The most popular control scheme is the PID controller. In [16], a PID controller is designed and a comparison between semi -active and passive suspension systems is performed. The simulation results show the ability of the PID controller to control the shock absorber variable and reduce the disturbances of the road on the vehicle body. Many optimization algorithms have been used to give the optimal values for a PID controller, such as the Particle Swarm Optimization (PSO) method, which has been used for a quarter vehicle active suspension system [17], and the Artificial Bee Colony (ABC) optimization technique, which has been used for a non-linear full active suspension system with traveler seat [18]. Several methods of tuning the PID controller have been introduced to enhance PID performance for active suspension systems in comparison with passive suspension systems under different road profiles [19]. A Linear Quadratic control (LQR) for full active suspension systems is introduced in [20] and the results of the LQR controller are successful in terms of there is little amplitude amount for tire deflection, suspension travel, and the acceleration of a vehicle, but the LQR controller for full active suspension cannot be implemented in rough road vibrations. Fuzzy Logic Control (FLC) schemes are utilized where the controller design does not require any information from the mathematical model and can overcome the problems of uncertainty effects [21]. FLC is used for suspension systems and the obtained outcomes show the ability of the proposed controller to limit the acceleration of vehicle body [22]. The performance of FLC for a non-linear full semi-active vehicle model has been tested and the outcomes obtained compared with those of a system controlled by a PID controller, with an improvement in driving comfort and road holding with disturbances [23]. The Takagi-Sugeno (T-S) model based FLC design strategy for a non-linear full suspension has been presented, with the numerical outcomes showing that the designed controller can satisfy good performance in spite of the presence of nonlinear dynamics of the actuator, constraints of the control inputs, and the variation of sprung mass [24]. Neurofuzzy controller and neural network for full vehicle nonlinear active suspension systems are introduced in [25]. MATLAB SIMULINK toolboxes are used to simulate the proposed controllers with the controlled model and to display the responses of the controlled model under different types of disturbance. The results show that the neurofuzzy controller is more effective and robust than the neural network based controller. In [26], a non-linear adaptive control and robust H_∞ schemes are proposed for full vehicle suspension systems with 7 degrees of freedom. The controller is successful in that the active controller is able to minimize the accelerations of the vehicle cabin in the directions of pitching, rolling, and heaving. The variations of system parameters and uncertainty are taken into consideration and guaranteed by small gain theorem. The author in [27] is concerned with the structure of reliable controllers for quarter vehicle suspension systems. A simultaneous mixed LQR/H_∞ control method is designed to improve the performance of suspension under breakdown of suspension components or probable sensor fault. The barrier method was utilized to give an approximate solution for solving the auxiliary minimization problem of simultaneous mixed LQR/H_∞ control.

Simulation results explain that the acquired static output feedback controllers can improve the performance of suspension in all failure modes. A Sliding Mode

Controller (SMC) is a robust controller and widely used because of its attractive characteristics of robustness to uncertainties of the system and finite-time convergence. The superiority of SMC had been applied both in theory and practice [28–30]. Ansari and Taparia constructed a quarter vehicle active suspension system using an improved SMC as control scheme with an observer design to estimate the road profile. The goal of designing the vehicle suspension system was to improve driving quality by controlling the forces of suspension to suit the driving and road conditions [31]. A fFuzzy Sliding Mode Controller (FSMC) for quarter active suspension system was proposed with PD sliding mode observer. The FSMC was utilized to minimize the power consumption and limit the vibration influences of SMC force, while the fuzzy controller was used to change the gain of switching control in order to reach the sliding mode surface, and the PD sliding mode observer was presented to estimate uncertainties of model and state variables. Results showed the goodness of the proposed observer and control method [32]. The SMC suffers from the chattering problem in the control signal, which can damage the mechanical parts of the suspension system. Higher Order SMC (HOSMC) was proposed based on a Super Twisting Algorithm (STA) for active suspension system, where the numerical outcomes showed the effectiveness of the proposed HOSMC in improving the comfortable driving of passengers and the ability of the proposed controller to overcome the chattering that appeared in first order SMC [33]. To limit the chattering problem, many studies have utilized the Super Twisting Sliding Mode Controller (STSMC), which represents a familiar higher order sliding mode algorithm introduced by Levant [34]. The finite-time control method is well known for its good robustness and disturbance rejection. The problem of finite-time tracking control is studied for uncertain nonlinear mechanical systems in [35] to achieve finite-time convergence of tracking errors. The stabilization problem via finite-time control method for the suspension systems of vehicle in the presence of hard constraints depending on theTerminal Sliding Mode Control (TSMC) scheme was presented in [36]. The TSMC method is characterized by its finite-time convergence, thus the TSMC method may be helpful in particular applications. A novel second order SMC algorithm was used in order to avoid the main troubles of the TSMC scheme (chattering and singularity). The prepared control method proved its effectiveness by both experiment results and theoretical analysis. The performance and achievement of the suspension system improved through satisfying the stabilization requirements for a perturbed vehicle suspension system by using the finite-time control strategy. Unknown and uncertain disturbances were efficiently compensated by proposing a version disturbance compensator with (finite-time) convergence performance. Furthermore, the prepared compensator was helpful because its ability to be continuous also can fully remove any identical disturbance. In practice, the law of continuous control would not result in chattering. It was proved that active suspension was stabilized with finite time. An example was designed to illustrate the activity of the proposed controller to achieve vehicle driving comfort [37]. In [38], the authors handled concerns related to finite-time control and the constraints of the displacement for the active suspension systems with three degrees of freedom. The Barrier Lyapunov theorem was used to handle the constraint problem of displacement as well as improve the vehicle's safety and driving comfort. The Neural Network (NN) was utilized to estimate the unknown function in the suspension system. Generally, the traditional finite-time controller includes the functions of absolute value or sign and this causes problems with controller buffeting. So, by employing the NN, Barrier Lyapunov theorem, and the finite-time notion, an adaptive finite-time control strategy was built to overcome this problem. The outcomes of two examples explain the goodness of the decided adaptive finite-time/NN control strategy.

In [39], Singh presented STSMC for a quarter vehicle active suspension system with a 3 degrees of freedom model and compared it with the passive suspension system to show the goodness in oscillation isolation of acceleration of passenger seat and responses of vertical displacement to improve passenger comfort under random road disturbance. Selection of the controller parameter values for building the system has been to date the main focus of this stream of research. Tuning of higher order SMC using a super-twisting algorithm with Multi Objective Genetic Algorithm (MOGA) has been proposed to control active suspension system. The controlled and uncontrolled results are compared to explain that building a model with higher order SMC optimized with MOGA is an effective model to depress the effects of oscillations and vibration [40].

This paper presents a proposed controller for a full vehicle active suspension system to decrease the influences of vibrations and disturbances on the vehicle body, as well as provide comfortable driving and better handling for a random road profile. An adaptive fuzzy STSMC approach with a new convergence proof is used to design an effective control strategy. The Artificial Bee Colony (ABC) technique is utilized to give the optimal values of the proposed controller parameters. Hydraulic actuator dynamics have been considered. The PID controller is used in order to control the force of a hydraulic actuator. In this study, the STSMC will prevent the chattering problem that appeared in first order SMC and the fuzzy system will approximate and estimate the unknown parameters of the system. Simulation results using the MATLAB program and SIMULINK toolbox show the superior efficiency of the proposed control scheme in comparison with SMC and passive suspension system.

The paper is organized as follows. The mathematical model of a full vehicle is described in Section 2. Section 3 presents the proposed controller. In Section 4, the simulation results of the proposed controller are presented. Section 5 shows the robustness test and Section 6 gives the conclusion of this paper.

2 | MATHEMATICAL MODEL OF FULL VEHICLE

Many researchers have used a quarter vehicle model for suspension with the suspension system as a linear model with no consideration for the behavior of nonlinearities such as the friction on dampers and springs [41,42]. In this work, the effect of nonlinearity in the suspension system is taken into consideration [43]. The model with eight degrees of freedom of full vehicle nonlinear active suspension system is given in Figure 1.

$$M_s\ddot{Z}_c = -\sum_{j=1}^{4} F_{kj} - \sum_{j=1}^{4} F_{cj} + \sum_{j=1}^{4} F_j \tag{1}$$

$$J_x\ddot{\alpha} = (F_{k1}-F_{k2}-F_{k3}+F_{k4})\frac{d}{2}+F_{c1}-F_{c2}-F_{c3}+F_{c4}\frac{d}{2}+(F_3-F_1+F_2-F_4)\frac{d}{2}+T_x \tag{2}$$

$$J_y\ddot{\varphi} = (F_{k3}+F_{k4})b-(F_{k1}+F_{k2})a+(F_{c3}+F_{c4})b-(F_{c1}+F_{c2})a+(F_1+F_2)a-(F_3+F_4)b+T_y \tag{3}$$

where

J_x, J_y: The inertia about x – axis and about y – axis.
M_s: The mass of sprung.
T_x, T_y: The bending torque and the braking torque.
a, b: The dimension between the gravity center of sprung mass and the front and rear axles.
d: The path width.

For (j = 1,2,3,4), the stiffness and damping coefficients of suspension Front (left – right) and Rear (right – left) are K_{sj}, C_{sj} respectively. The spring force (F_{kj}) and the damper force (F_{cj}) are described as:

$$F_{kj} = K_{sj}(Z_j - W_j) + \vartheta K_{sj}(Z_j - W_j)^3 \tag{4}$$

$$F_{cj} = C_{sj}\left(\dot{Z}_j - \dot{W}_j\right) + \vartheta C_{sj}\left(\dot{Z}_j - \dot{W}_j\right)^2 sgn\left(\dot{Z}_j - \dot{W}_j\right) \tag{5}$$

The generated nonlinear force (F_j) is defined as the form:

$$F_j = F_{hyj} - F_{frj} \tag{6}$$

where, the nonlinear hydraulic actuator and frictional forces are (F_{hyj}, F_{frj}), and (ϑ) is an empirical operator. The force of hydraulic actuator can be written as:

$$F_{hyj} = A_p P_{Lj} \tag{7}$$

The differential equation of the j^{th} actuator's piston pressure (P_{Lj}) can be written as:

$$\dot{P}_{Lj} = -\sigma A_p\left(\dot{Z}_j - \dot{W}_j\right) - \beta P_{Lj} + x_{vj}\gamma\sqrt{P_{sup} - sgn(x_{vj})P_{Lj}} \tag{8}$$

In order to control the servo-valve displacement of spool (x_{vj}), the voltage input signal (u_{pj}) is utilized, where the

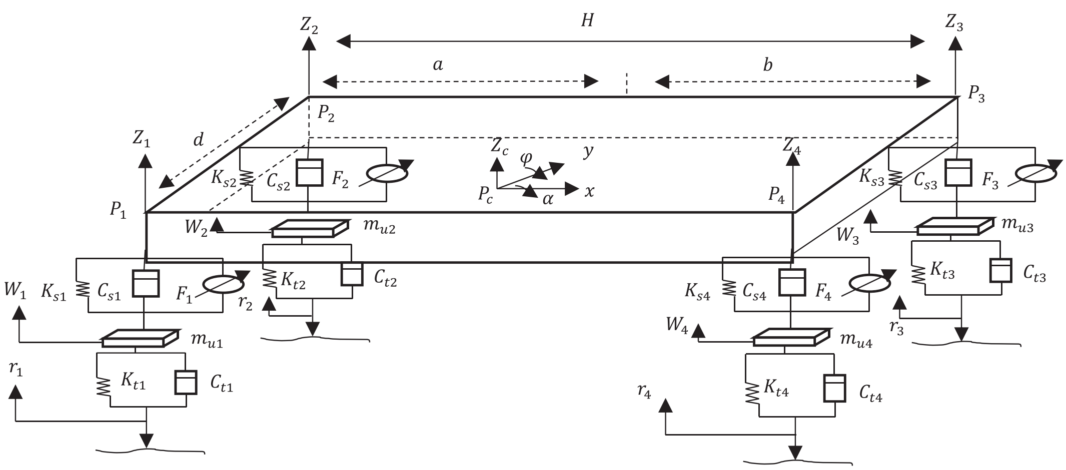

FIGURE 1 Full model

dynamic equation of the servo-valve displacement of spool is described as:

$$\dot{x}_{vj} = \frac{1}{\tau}\left(K_c u_{pj} - x_{vj}\right) \tag{9}$$

where
σ, β and γ: The actuator parameters,
A_p: The piston area.
P_{sup}: The pressure supply of the pump.
K_c: The conversion gains.
τ: The time of the mechanical delay of the system.

The frictional force was taken in consideration, and implemented with signum function approximation as:

$$F_{frj} = \begin{cases} \psi sgn\left(\dot{Z}_j - \dot{W}_j\right) & for \quad \left|\left(\dot{Z}_j - \dot{W}_j\right)\right| \geq 0.01 \\ \psi sin\dfrac{\pi\left(\dot{Z}_j - \dot{W}_j\right)}{0.02} & for \quad \left|\left(\dot{Z}_j - \dot{W}_j\right)\right| < 0.01 \end{cases} \tag{10}$$

where (ψ) is the empirical factor. For ($j = 1,2,3,4$), W_j is the vertical displacement of unsprung mass at Front (left – right), Rear (right – left) respectively and its motion can be defined as:

$$m_{uj}\ddot{W}_j = -K_{tj}\left(W_j - r_j\right) - C_{tj}\left(\dot{W}_j - \dot{r}_j\right) + F_{kj} + F_{cj} - F_j \tag{11}$$

where
m_{uj}: The unsprung mass.
K_{tj}, C_{tj}: The stiffness and damping coefficients of tire at Front (left – right) and Rear (right – left) respectively.
r_j: The road profiles.

According to Figure 1, The equation of plane can be written as:

$$\begin{vmatrix} x - x_1 & y - y_1 & z - z_1 \\ x_2 - x_1 & y_2 - y_1 & z_2 - z_1 \\ x_4 - x_1 & y_4 - y_1 & z_4 - z_1 \end{vmatrix} \tag{12}$$

The points coordinates (P_1,P_2,P_3, P_4 and P_c), can be denoted as (0, 0, z_1), (0, d, z_2), (H, d,z_3), (H, 0, z_4) and (a, $\frac{d}{2}$, z_c). These points will be substituted in (12) to imply:

$$z = \frac{(z_4 - z_1)}{H} + \frac{(z_2 - z_1)}{d} + z_1 \tag{13}$$

The points coordinate (P_3 and P_c) can be substituted in (13), so, the vertical displacement at (Z_3 and Z_c) can be given as:

$$z_3 = -z_1 + z_2 + z_4 \tag{14}$$

$$z_c = gz_1 + 0.5z_2 + \frac{a}{H}z_4 \tag{15}$$

where $g = 0.5 - \frac{a}{H}$. The Rolling and Pitching angles can be written as:

$$\alpha = \frac{\partial z}{\partial y} = \frac{z_2 - z_1}{d} \tag{16}$$

$$\varphi = \frac{\partial z}{\partial x} = \frac{z_4 - z_1}{H} \tag{17}$$

According to (15), (16), and (17), the vertical displacement of the sprung mass (Z_j) with respect to Z_c, α, and φ can be given as [25]:

$$Z_1 = Z_c - 0.5d\alpha - a\varphi \tag{18}$$

$$Z_2 = Z_c + 0.5d\alpha - a\varphi \tag{19}$$

$$Z_3 = Z_c + 0.5d\alpha + a\varphi \tag{20}$$

$$Z_4 = Z_c - 0.5d\alpha + a\varphi \tag{21}$$

3 | PROPOSED CONTROLLER DESIGN

The key objective of controller design for the suspension system of a vehicle is to decrease the vibration in the cabin caused by road harshness and to support street handling. This needs a controller of accurate and fast specifications to achieve these objectives. The following proposed control scheme will be adopted to achieve these requirements efficiently. The proposed control scheme has been designed as shown in Figure 2. It consists of four sub control schemes, one for each of the four wheels. According to Figure 2, the sub control scheme for each wheel consists of FSTSMC in the outer loop and a PID controller in the inner loop. Many applications have used a fuzzy system as a controller or estimator [44–50]. In the outer loop, a fuzzy system is combined with the STSMC, where the fuzzy system is used as an estimator for the unknown parameters in the suspension system and STSMC is used to limit the nonlinearity and the chattering problem at the output. The PID controller in the inner loop is designed to control the actuator and create the required actuator force at each corner.

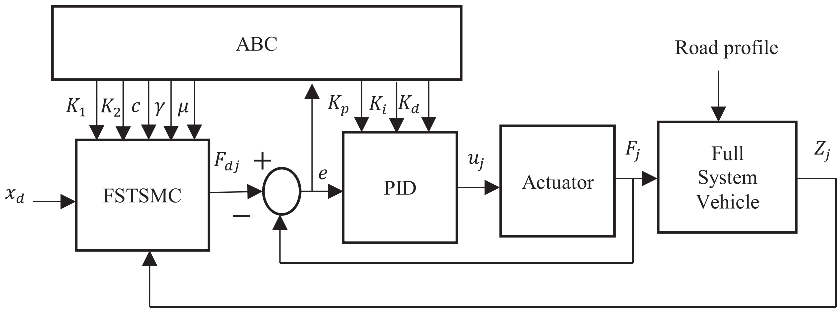

FIGURE 2 Block diagram of proposed control scheme

3.1 | Outer loop design

For each side of the wheel, a fuzzy system is combined with the STSMC. Generally, SMC involves two sections. The first one is responsible for 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 [32], so many researchers are interested in this feature. The dynamic equation at each side ($j = 1,2,3,4$) can be written as:

$$M_{sj}\ddot{Z}_j + F_{kj} + F_{cj} - F_j = 0$$

$$m_{uj}\ddot{W}_j + K_{tj}(W_j - r_j) + C_{tj}\left(\dot{W}_j - \dot{r}_j\right) - F_{kj} - F_{cj} + F_j = 0$$

For each of the four sides, the body displacement is defined as $x_1 = Z_j$, the velocity of the body is $x_2 = \dot{Z}_j$, the unsprung mass displacement is $x_3 = W_j$, and the velocity of the unsprung mass is $x_4 = \dot{W}_j$, so the sate vector is $x = [x_1, x_2, x_3, x_4]^T$. The body vertical motion at each side (j) can be described as:

$$\ddot{x}_1 = f(x,t) + \frac{1}{M_{sj}}F_j \tag{22}$$

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

$$f(x,t) = \frac{1}{M_{sj}}\left(-F_{kj} - F_{cj}\right) \tag{23}$$

The sliding surface can be defined as:

$$s = \dot{e} + ce, \text{ and } c > 0 \tag{24}$$

The convergent rate of x_1 is c. The tracking error can be defined as:

$$e = x_1 - x_d \tag{25}$$

$$\dot{s} = \ddot{e} + c\dot{e} = \ddot{x}_1 - \ddot{x}_d + c\dot{e} = f(x,t) + \frac{1}{M_{sj}}F_{dj} + c\dot{x}_1 \tag{26}$$

The desired signal and the actual signal are x_d which equals to zero and x_1 respectively. The control signal of SMC can be defined as:

$$F_{dj} = u_{eq} + u_{sw} \tag{27}$$

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:

$$u_{sw} = -K\,sign(s) \tag{28}$$

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 this problem taking into account the existence of external disturbances and unknown parameters of the system, a STSMC scheme is presented. This strategy represents a good method to overcome the effect of uncertainty in the system. Also, the serious problem of the control design is the chattering of 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 represents a robust scheme that can eliminate chattering while keeping the other properties of SMC. STSMC consists of two sections. The first one represents the discontinuous function of the sliding variable, and the second one represents the continuous function of the derivative of the sliding variable, and the switching control can be defined as:

$$u_{sw} = -K_1\sqrt{|s|}\,sign(s) + m \tag{29}$$

$$\dot{m} = -K_2 sign(s) \tag{30}$$

The parameters ($K_1$and, K_2) are positive constant. Based on the STSMC, the switching control can be designed as:

$$u_{sw} = -K_1\sqrt{|s|}sign(s) - K_2\int sign(s)dt \tag{31}$$

The final control law can be defined as:

$$F_{dj} = M_{sj}\left(-f(x,t) - c\dot{x}_1 - K_1\sqrt{|s|}sign(s\right) - K_2\int sign(s)dt) \tag{32}$$

To estimate the unknown function, the fuzzy system is used. The Minimum Parameter Learning (MPL) technique is a self – learning method for the parameters in the system, which uses a fuzzy estimator to approximate and estimate the unknown and uncertain nonlinear parts of the system [51]. The systems can be considered uncertain systems, where the dynamics of the system are varying under different conditions. The FSTSMC strategy 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:

$$\hat{f}(x|\theta_f) = \frac{\sum_{l_1=1}^{5}\sum_{l_2=1}^{5}\sum_{l_3=1}^{5}\sum_{l_4=1}^{5} y_f^{l_1l_2l_3l_4}\left(\prod_{i=1}^{4}\mu_{A_i}^{l_i}(x_i)\right)}{\sum_{l_1=1}^{5}\sum_{l_2=1}^{5}\sum_{l_3=1}^{5}\sum_{l_4=1}^{5}\left(\prod_{i=1}^{4}\mu_{A_i}^{l_i}(x_i)\right)} \tag{33}$$

where $\mu_{A_i}^{l_i}(x_i)$, is the membership function of x_i, which is a Gaussian function defined as:

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

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

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

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

and, $\mu_{NM}(x_i) = exp\left[-\left(\left(x_i - \frac{\pi}{3}\right)/\left(\frac{\pi}{12}\right)\right)^2\right]$ and $y_f^{l_1l_2l_3l_4}$ is a parameter in the group $\hat{\theta}_f \in Rule^{(625)}$, and fuzzy rules form column vector for $p_1 = p_2 = p_3 = p_4 = 5$, $\xi(x) = \prod_{i=1}^{4} p_i = p_1 \times p_2 \times p_3 \times p_4 = 625$ are:

$Rule^{(1)}$: 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 $A_1^{l_i}$, $A_2^{l_i}$, $A_3^{l_i}$, and $A_4^{l_i}$ respectively, $l_i = 1,2,3,4,5$. The Equation 33 can be replaced as:

$$\hat{f}(x|\theta_f) = \hat{\theta}^T\xi(x)\,\hat{\mathrm{f}}(\mathrm{x}|\theta_\mathrm{f}) = \hat{\theta}^\mathrm{T}\xi(\mathrm{x}) \tag{34}$$

$$\xi_{l_1l_2l_3l_4}(x) = \frac{\prod_{i=1}^{4}\mu_{A_i}^{l_i}(x_i)}{\sum_{l_1=1}^{p_1}\sum_{l_2=1}^{p_2}\sum_{l_3=1}^{p_3}\sum_{l_4=1}^{p_4}\left(\prod_{i=1}^{4}\mu_{A_i}^{l_i}(x_i)\right)} \tag{35}$$

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

$$\theta_f^* = arg\min_{\theta_f\in\Omega_f}\left[sup\left|\hat{f}\left(x|\ \theta_f \atop x\in R^n\right) - f(x)\right|\right] \tag{36}$$

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

$$f = \theta_f^{*T}\xi(x) + \varepsilon \mathrm{f} = \theta_\mathrm{f}^{*\mathrm{T}}\xi(\mathrm{x}) + \varepsilon \tag{37}$$

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

$$\hat{f}(x|\theta_f) = \hat{\theta}_f{}^T\xi(x)\,\hat{\mathrm{f}}(\mathrm{x}|\theta_\mathrm{f}) = \hat{\theta}_\mathrm{f}^\mathrm{T}\xi(\mathrm{x}) \tag{38}$$

In this work we consider $f(x,t)$ unknown function so by using minimum parameter learning [51], express a positive constant as $\phi = \left||\theta_f^*\right||^2$ and let $\hat{\phi}$ be an estimate of ϕ.

The stability requirements via STSMC can be satisfied, the Lyapunov function was selected based on [52] as:

$$V = 2K_2\sqrt{s} + \frac{1}{2}\left(K_1\sqrt{|s|}sign(s-m)\right)^2 + \frac{1}{2}m^2 + \frac{1}{2\gamma}\tilde{\phi}^2 \tag{39}$$

where $\tilde{\phi} = \hat{\phi} - \phi$ and γ is positive. It can be written in the quadratic form as:

$$V = \Omega^T P\Omega + \frac{1}{2\gamma}\tilde{\phi}^2 \tag{40}$$

and, $\Omega = \left[\sqrt{|s|}sign\,(s) \quad m\right]^T$,

$\mathrm{V} = \frac{1}{2}\mathrm{s}^2 + \frac{1}{2\gamma}\tilde{\phi}^2$, $P = \frac{1}{2}\begin{bmatrix} K_1^2 + 4K_2 & -K_1 \\ -K_1 & 2 \end{bmatrix}$.

$\dot{V} = \left(\dot{\Omega}^T P\Omega + \Omega^T P\dot{\Omega}\right) + \frac{1}{\gamma}\tilde{\phi}\dot{\hat{\phi}}\,\dot{V} = s\dot{s} + \frac{1}{\gamma}\tilde{\phi}\dot{\hat{\phi}}$, which can be written as:

$$\dot{V} = \dot{s}\,sign(s)\left(2K_2 + \frac{1}{2}K_1^2\right) - K_1\sqrt{|s|}\,\text{sign}(s) - \frac{K_1 m\dot{S}}{2\sqrt{|S|}} + 2m\,\dot{m} + \frac{1}{\gamma}\tilde{\phi}\dot{\hat{\phi}}.$$

The adaptive law is designed as:

$$\dot{\hat{\phi}} = \frac{\gamma}{2}s^2\xi^T\xi - \kappa\gamma\hat{\phi} \tag{41}$$

where $\kappa = \frac{2\mu}{\gamma}$ and, $\kappa > 0$ for positive (μ and γ).

The bounds of uncertainties will be substituted in (40), so that $\dot{V} < \frac{K_1}{2\sqrt{|s|}}\left(\Omega^T Q\Omega\right)$, where.

$$Q = \begin{bmatrix} K_1^2 + 2K_2 - \left(\frac{4K_2}{K_1} + K_1\right)\Delta & -K_1 + \frac{\Delta}{2} \\ -K_1 + \frac{\Delta}{2} & 1 \end{bmatrix},$$

the bounded constant $\Delta \geq 0$. For stability requirements the condition that $\dot{V} < 0$ must be satisfied and gains K_1 and K_2 must satisfy the conditions:

$$K_1 > 2\Delta \quad \text{and}\, K_2 > \frac{K_1\Delta^2}{8(K_1 - 2\Delta)} \tag{42}$$

The FSTSMC law can be designed as:

$$F_{dj} = M_{sj}\left(-\frac{1}{2}s\hat{\phi}\xi^T\xi - c\dot{x}_1 - K_1\sqrt{|s|}sign(s)\right) - K_2\int sign(s)dt - \mu s) \tag{43}$$

The accuracy of convergence is based on values of (c), (K_1), and (K_2). The sliding surface converges to zero lastly.

3.2 | Inner loop design

The design of the PID controller will provide a stabilization for the system and a suitable system performance, also widely used in industrial applications [16]. In this study, the PID controller has been designed for each side of the four wheels. The equation of the PID controller is defined as:

$$u_j = K_p e + K_i\int_0^t edt + K_d\frac{de}{dt} \tag{44}$$

where (u_j) 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:

$$e = F_{dj} - F_j \tag{45}$$

3.3 | Optimization of the proposed controller parameters using ABC algorithm

The Artificial Bee Colony (ABC) algorithm was first proposed by Dervis Karaboga [53]. ABC is an optimization algorithm based on the behavior of bee colonies, where these colonies are employed or unemployed bees. The employed bees search for the sources of food and store the information about the quality of the food source. The employed bees give and share the information about the food source position with the bees of the hive.

Each food source represents an optimization solution for the problem where the nectar quantity sources matches to the solution fitness [54]. The unemployed bees are onlooker bees and scout bees. The onlooker bees choose the best food source based on the information found by employed bees [55]. The scout bees are related to employed bees where, if a food source becomes deserted, the employed bees will be scouts and then search for another source of sustenance to become employed again.

The (ABC) method used in this work can efficiently find a suitable gain parameter for the proposed controller. It is aimed to minimizethe 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:

$$cost_1 = \sum_{i=1}^{4}\sum_{i=1}^{N} Z_{ji}^2,\; cost_2 = \sum_{i=1}^{4}\sum_{i=1}^{N}\ddot{Z}_{ji}\,2,$$

$$cost_3 = \sum_{i=1}^{4}\sum_{i=1}^{N}\left(Z_{ji} - W_{ji}\right)^2.$$

The ABC technique involves the following step [49].

1 Initialize ABC parameters such as (maximum number of cycles, colony dimension, parameter limits).
2 Generate an initial population of food sources (FS) individuals randomly. Each solution K_{sou}, $K_{sou} = [1,2, \ldots, FS]$ represents a D – dimensional vector corresponding to PID and FSTSMC parameters where,

$$D = \left[c\ \mu\ K_1\ K_2\ K_p\ K_i\ K_d\right]$$

3 For each solution, evaluate the fitness function as:

$$fitness = \frac{1}{1 + cost} \tag{46}$$

4 Increase the number of cycles 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:

$$K_{new}(s,j) = K(s,j) + \Delta(s,j)(K(s,j) - K(n,j)) \tag{47}$$

where $n = 1,2,...,FS$, $(n \neq s)$ and $j = 1, 2, 3, 4, 5, 6, 7$ are randomly chosen 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 produces new solution (K_{new}) as in (47). Then calculate the fitness function of each new solution as given in (46), a greedy selection is applied between the new and old solution in order to save the best solution and ignore the other one.

TABLE 1 The hydraulic acutator parameters

Sample	Value
σ	4.515×10^{13}
β	1
γ	1.545×10^{9}
Psup	10342500 (pa)
Ap	3.35×10^{-4} (m^2)
Kc	0.001 (m/V)
τ	1/30 (s)

TABLE 2 Full vehicle numerical values

Parameter	Value
M_s	1460 (kg)
J_x	460 (kg.m^2)
J_y	2460 (kg.m^2)
d	1.51 (m)
a	1.011 (m)
b	1.803 (m)
K_{s1}, K_{s2}	19960 (N/m)
K_{s3}, K_{s4}	17500 (N/m)
C_{s1}, C_{s2}	1290 (N.s/m)
C_{s3}, C_{s4}	1620 (N.s/m)
m_{u1}, m_{u2}	40 (kg)
m_{u3}, m_{u4}	35.5 (kg)
M_{sj}	290 (kg)
K_{t1}, K_{t2}, K_{t3}, K_{t4}	175500 (N/m)
C_{t1}, C_{t2}, C_{t3}, C_{t4}	14.6 (N.s/m)
ϑ, ψ	0.1

6 If the food source is not improved over an iterations number, that source is 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 till an extreme number of cycles is reached. If the maximum numbers of iterations is satisfied, stop and return to the best solution found.

4 | SIMULATION RESULTS

In this study, the performance of the proposed controller for full vehicle active suspension system is studied. The applied road profile is a random signal. The parameters of the hydraulic actuator and the full vehicle suspension

TABLE 3 ABC algorithim parameters

Parameter	Value
Number of optimization parameters	7
Maximum number of iteration cycle	50
Colony dimension	100
Number of food source	50

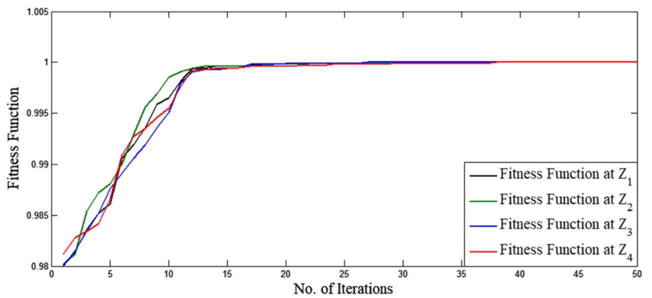

FIGURE 3 Relationship between fitness function and iteration at each Z_j [Color figure can be viewed at wileyonlinelibrary.com]

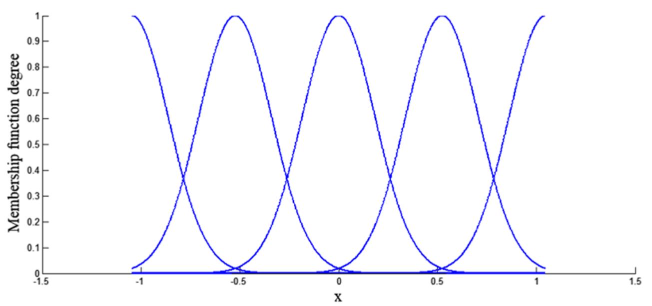

FIGURE 4 Membership function of X_i [Color figure can be viewed at wileyonlinelibrary.com]

TABLE 4 The optimal paramerters of proposed controller

Controller	K_p	K_i	K_d	K_1	K_2	c	μ
C_1	117.8791	260.0891	339.2469	1419	374.6	2.229	9.0183
C_2	228.8833	115.0288	197.1531	1509.6	980	7.491	4.0075
C_3	481.5074	243.9378	183.0858	552.8	891.7	1.9609	7.2887
C_4	272.9497	311.6541	493.979	1359.7	1293	6.8838	2.7803

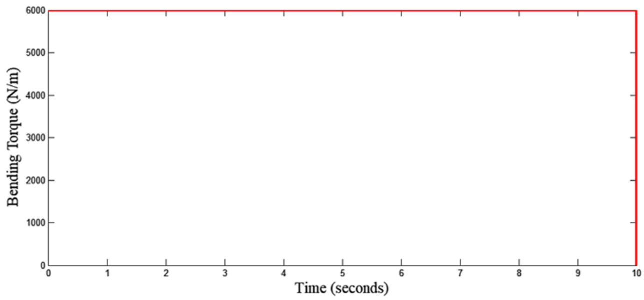

FIGURE 5 Bending torque [Color figure can be viewed at wileyonlinelibrary.com]

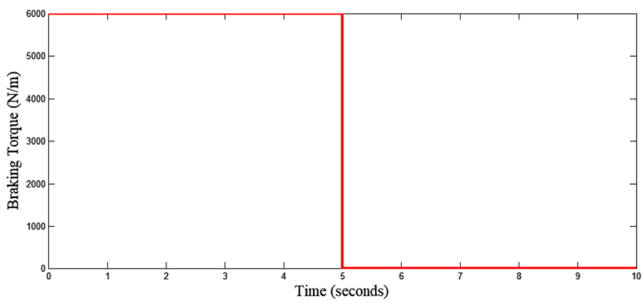

FIGURE 6 Braking torque [Color figure can be viewed at wileyonlinelibrary.com]

system are given in Table 1 and Table 2. The proposed control scheme of FSTSMC and PID controller is simulated using MATLAB/SIMULINK and compared with the SMC controller and with a passive suspension system. Four FSTSMC and PID controllers are used, one for each suspension system in each side.

The ABC algorithm outputs are the optimal values of proposed controller parameters. The chosen values for the ABC algorithm are given in Table 3. The parameters of the control scheme for each side should be designed. From the stability conditions in (24, 42, 43, and 44), and in order to reduce the optimization time, the following ranges: c and $\mu \in [1\ 10]$; K_1 and $K_2 \in [1\ 2000]$; K_p, K_i, and $K_d \in [1\ 500]$ for the parameters of the four sub control schemes are chosen. (By several simulation tests for the system the suitable ranges are estimated). The fitness function is determined as in (46). It can be seen that the fitness function converges to nearly one as shown in Figure 3 after (25) iterations.

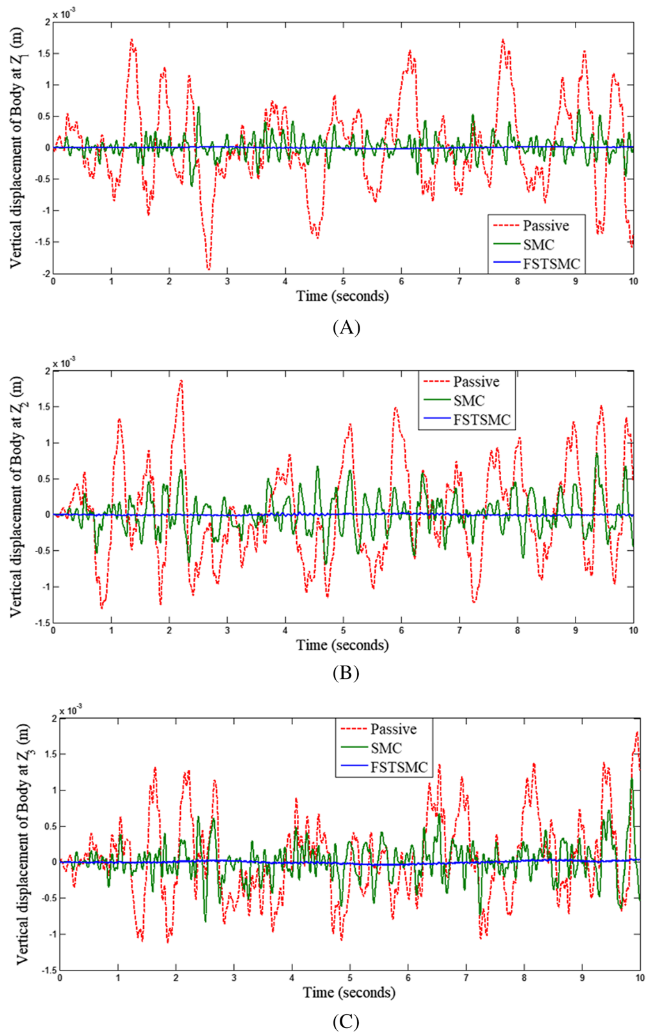

FIGURE 7 Displacement and rotation of body at: (a): Z_1, (b): Z_2, (c): Z_3, (d): Z_4, (e): Z_c, (f): α, (g): φ [Color figure can be viewed at wileyonlinelibrary.com]

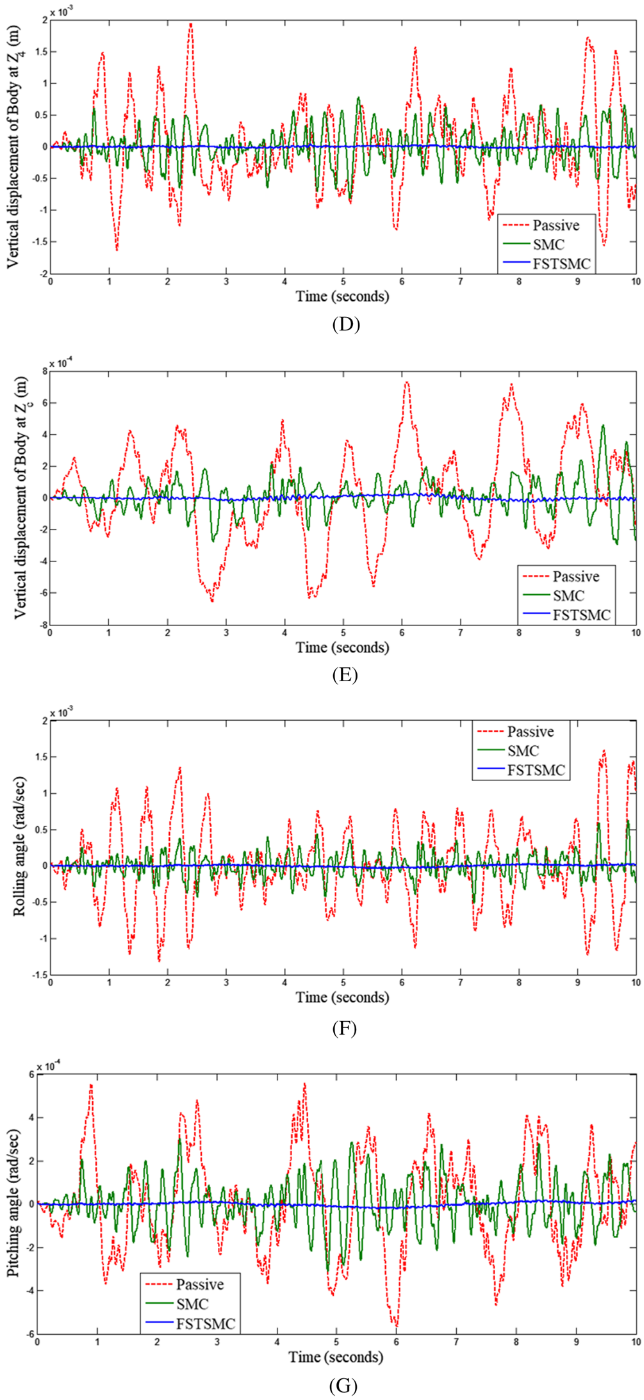

FIGURE 7 (Continued)

The centers of the five Gaussian membership function in the inputs of the fuzzy estimator are ($\mu_1 = 1$, $\mu_2 = 0.5$, $\mu_3 = 0$, $\mu_4 = -0.5$, and $\mu_5 = -1$) as shown in Figure 4. After ABC optimization algorithm, the optimal values for parameters of each proposed controller (C_1, C_2, C_3 and C_4) are given in Table 4. The applied bending torque and braking torque (T_x, T_y) to the full vehicle active suspension system are shown in Figures 5 and 6.

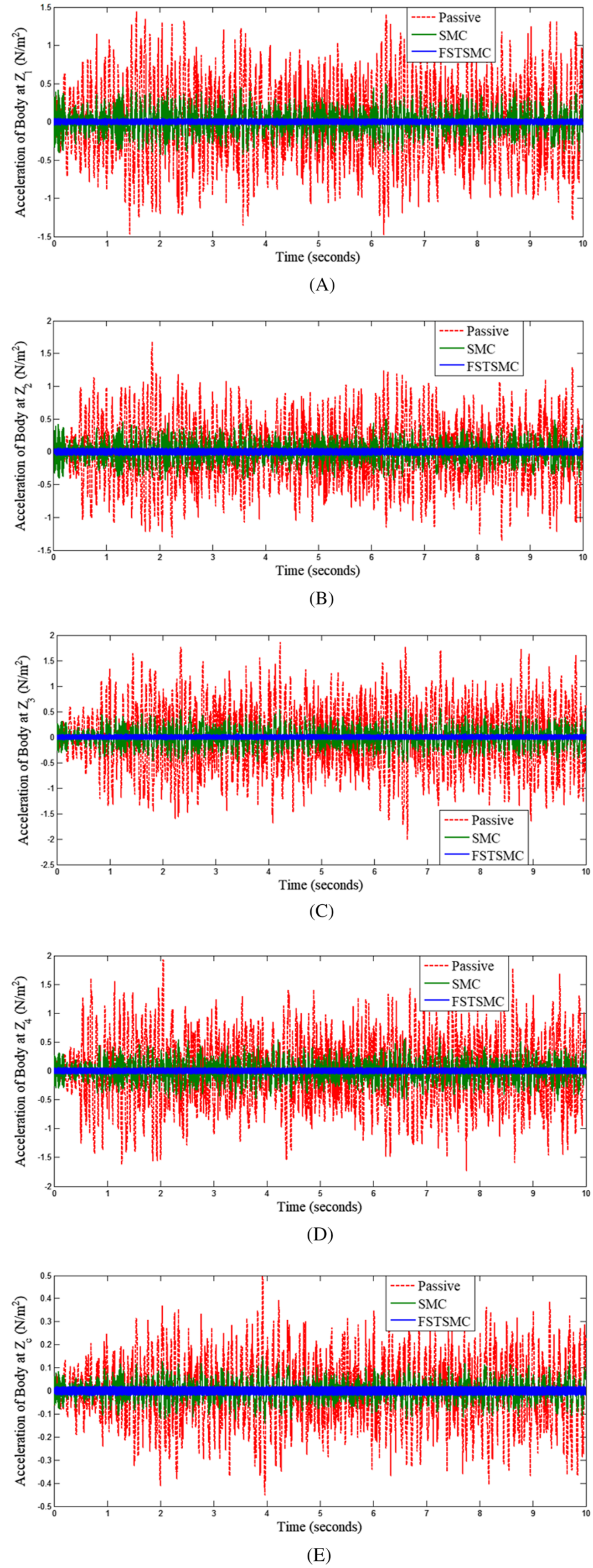

FIGURE 8 Acceleration of body at: (a): Z_1, (b): Z_2, (c): Z_3, (d): Z_4, (e): Z_c [Color figure can be viewed at wileyonlinelibrary.com]

The output responses of full vehicle active suspension system with proposed control scheme FSTSMC and PID optimized with ABC are compared with the passive suspension system and SMC, as shown in Figure 7, where the transient amplitude of the sprung mass displacement responses is improved, effectively minimized, and lastly reached to zero by the proposed control method. The body acceleration is shown in Figure 8, which also shows good results. The acceleration is extremely minimized in all four sides and in the center of the vehicle with the proposed control scheme. The outputs of PID controllers (u_1, u_2, u_3, and u_4) are utilized to control the hydraulic actuator and produce force (F_j) for each suspension system, as shown in Figure 9. After applying bending and braking torques, the responses of full vehicle suspension system output via the proposed control scheme are compared with the passive suspension system and with SMC, as shown in Figures 10 and 11, where these figures show that the transient responses amplitude of the sprung mass displacement have been effectively minimized. The FSTSMC strategy has a superior performance and gives good dynamic properties with minimum chattering in the output responses of all figures with and without applying the bending and braking torques.

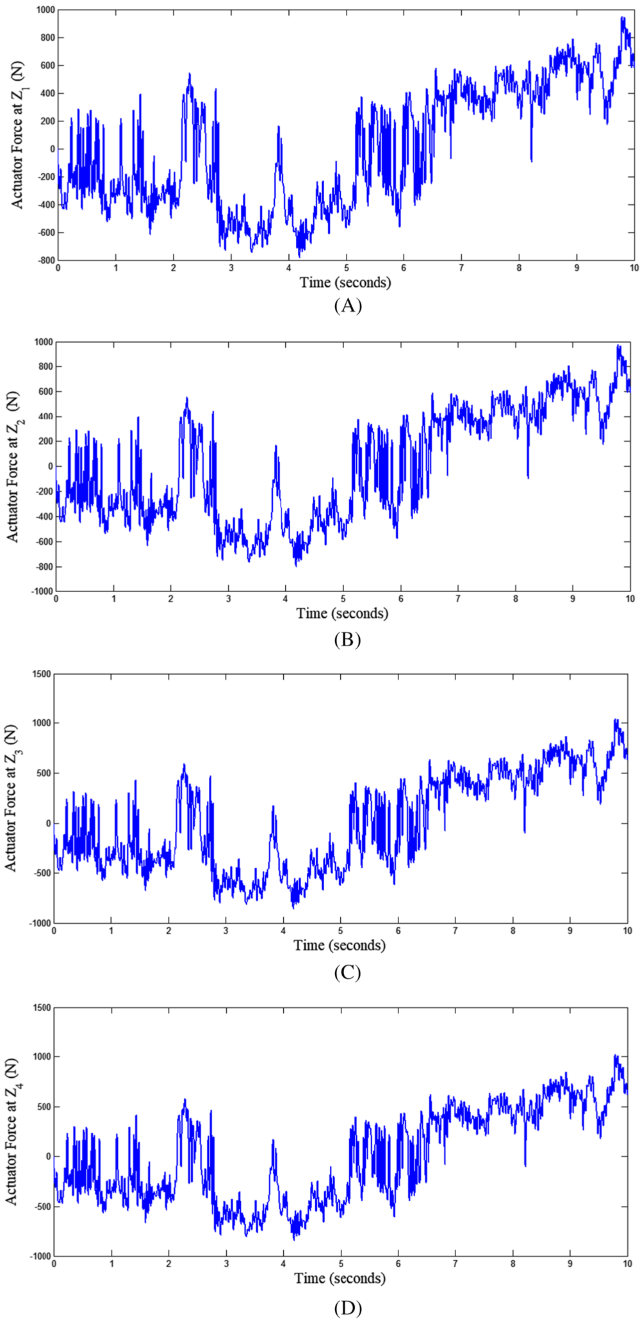

FIGURE 9 Actuator force for random road profile at: (a): Z_1, (b): Z_2, (c): Z_3, (d): Z_4 [Color figure can be viewed at wileyonlinelibrary.com]

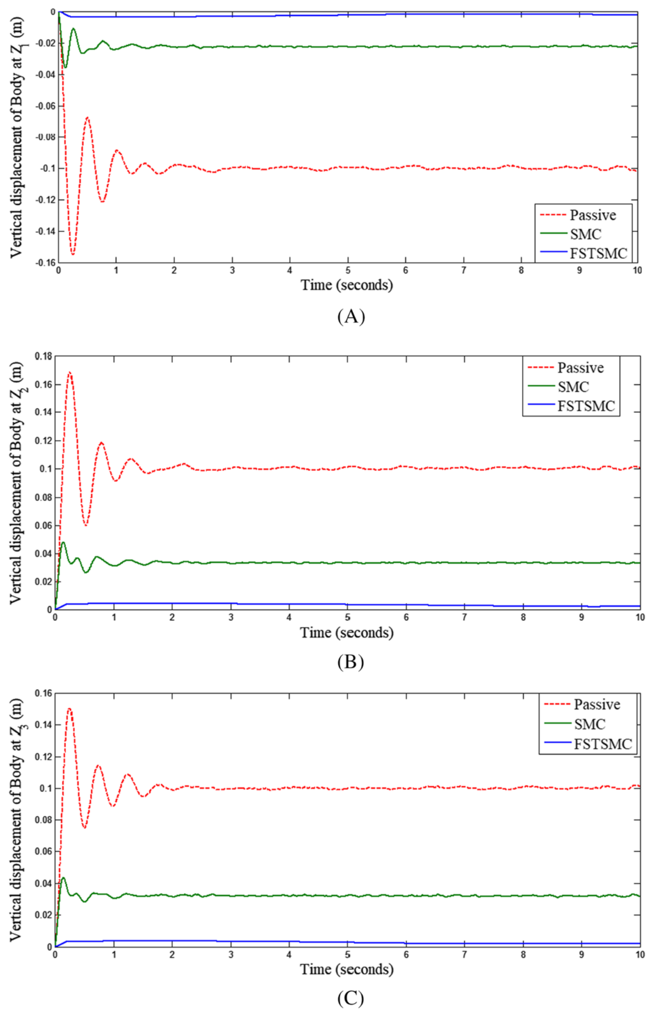

FIGURE 10 Displacement and rotation of body (with bending torque) at: (a): Z_1, (b): Z_2, (c): Z_3, (d): Z_4, (e): Z_c, (f): α, (g): φ [Color figure can be viewed at wileyonlinelibrary.com]

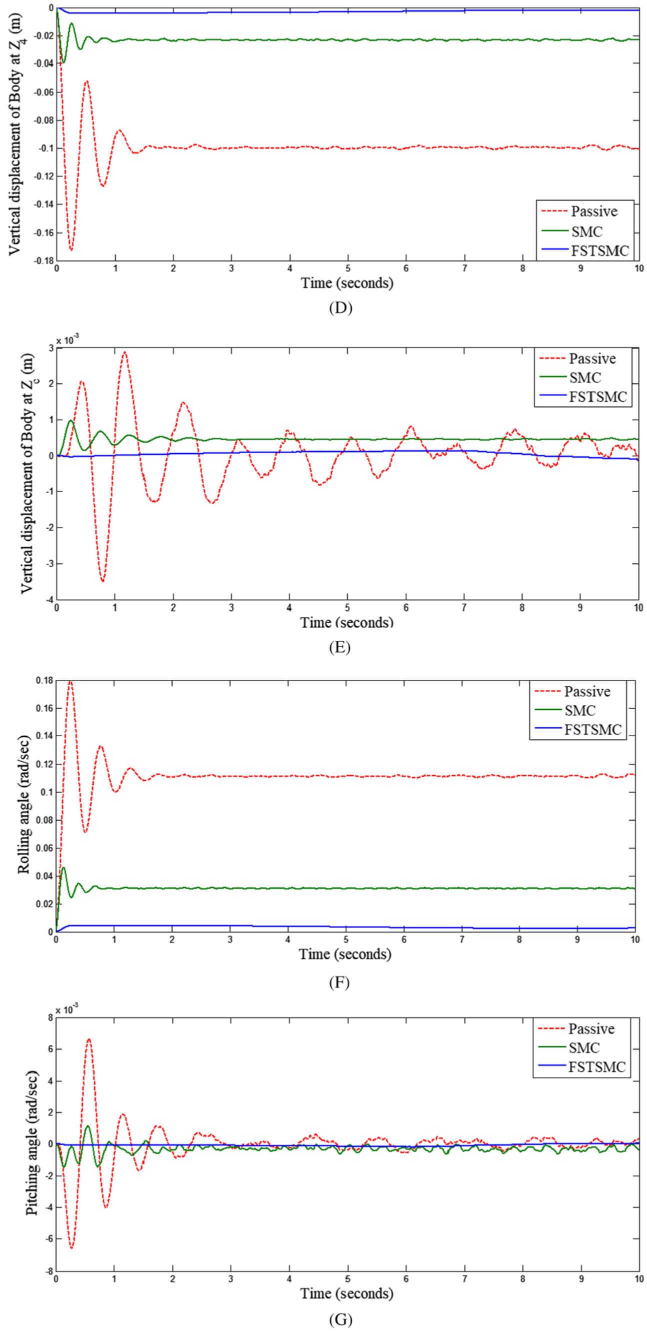

FIGURE 10 (Continued)

5 | ROBUSTNESS TEST

In order to test the goodness of the proposed controller, a robustness term is defined using (48). The robustness term can evaluate the performance of the proposed controller, where it will be still stable even by applying different disturbances with different amplitudes or different frequencies as an input to the suspension system. In this work the robustness test has been made for the proposed control scheme and compared with the SMC and passive suspension system. The system is subjected to random road profile with different amplitudes from (0.01 m to 0.07 m) and fixed frequency. The cost function is applied as:

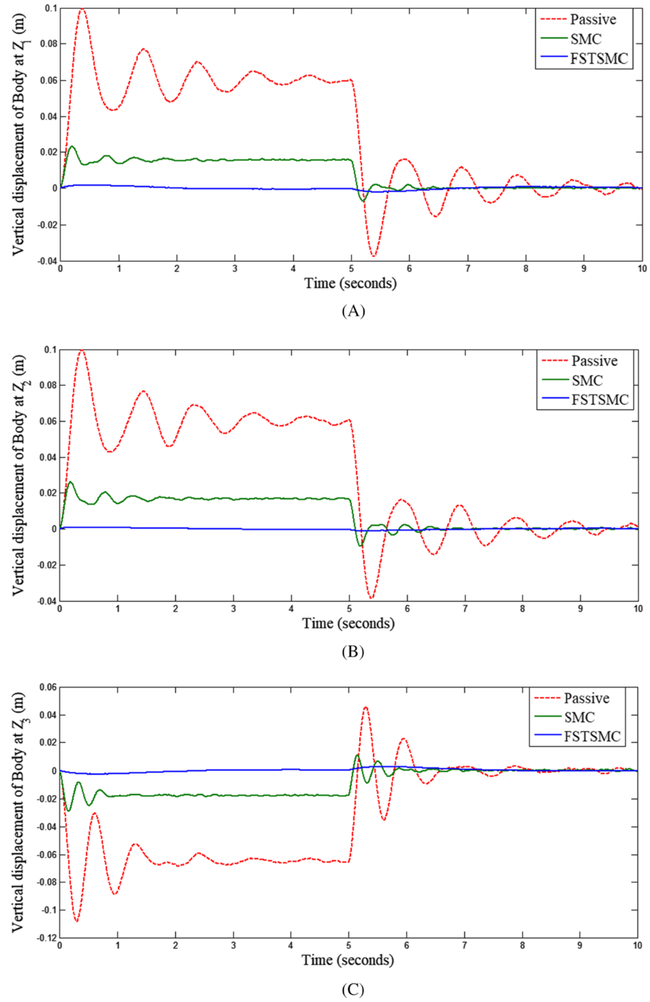

FIGURE 11 Displacement and rotation of body (with braking torque) at: (a): Z_1, (b): Z_2, (c): Z_3, (d): Z_4, (e): Z_c, (f): α, (g): φ [Color figure can be viewed at wileyonlinelibrary.com]

$$cost = \sum_{j=1}^{N} \sum_{j=1}^{4} \left(Z_{ij} - Z_{dij}\right)^2 \tag{48}$$

where, the vertical displacement of each (j) corner in the full active suspension system is Z_{ij} and Z_{dij} is the desired vertical displacement, which is equal to zero, so the cost function can be implemented as:

$$cost = \sum_{j=1}^{N} \sum_{j=1}^{4} \left(Z_{ij}\right)^2 \tag{49}$$

The obtained outcomes from the MATLAB program in Table 5 show that the cost function in the proposed

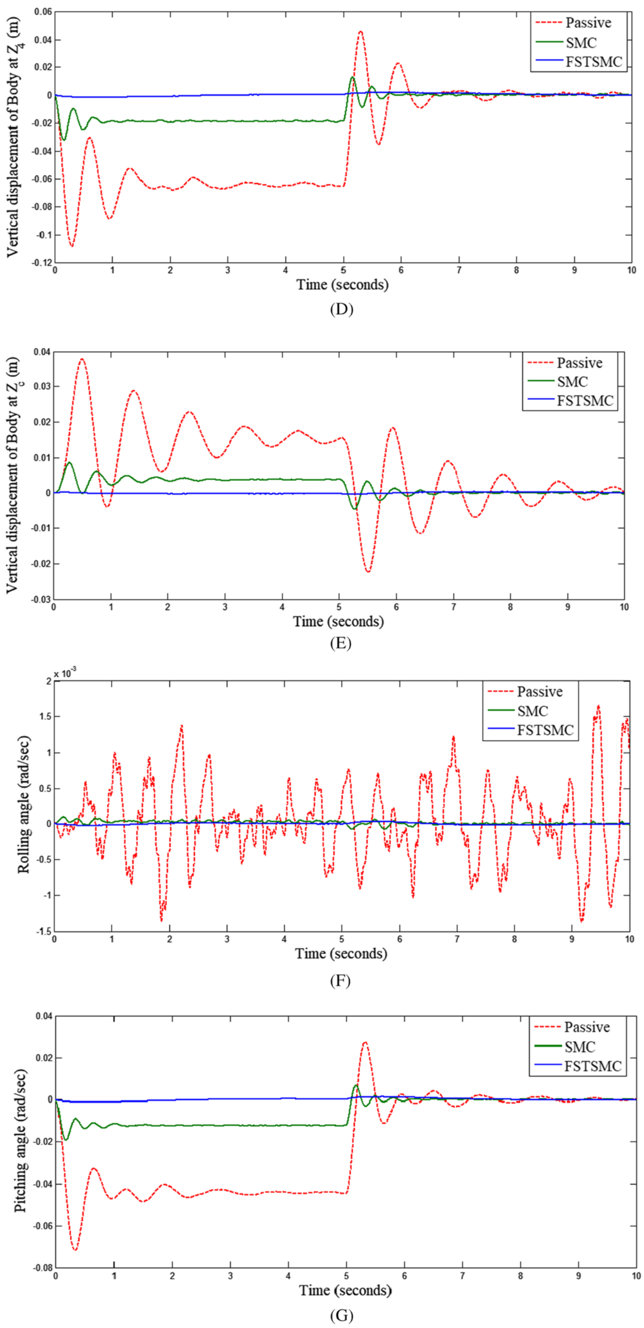

FIGURE 11 (Continued)

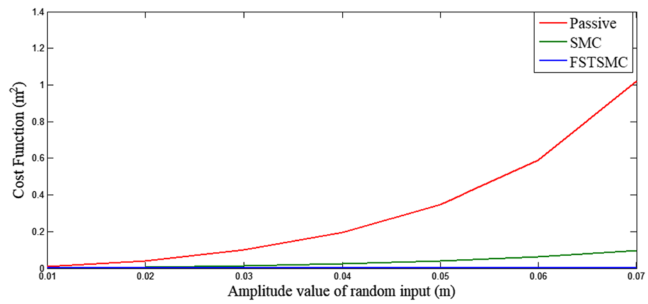

FIGURE 12 Cost function with different amplitudes of random wave [Color figure can be viewed at wileyonlinelibrary.com]

controller is improved for the random input with different amplitudes. The cost function for different amplitudes of random signal wave input for passive and active (SMC and FSTSMC) vehicle is shown in Figure 12.

6 | CONCLUSION

This work presents and demonstrates the validity of using an optimized fuzzy estimator with STSMC for controlling a full vehicle. The proposed control scheme gives very effective and good results when compared with SMC and a passive system. Results show that the vertical deflection and acceleration are extremely minimized in all four sides and in the center of the vehicle. All parameters of the proposed control strategy, which combines FSTSMC with a PID controller, are optimized with the ABC algorithm. The obtained results show that the proposed control scheme can give good performance and also minimize the transient responses amplitude of the suspension system under random road profile with and without bending and braking torques. Also, the outcomes show that the proposed control scheme gives less chattering, and this is more acceptable since high chattering may harm the actuator. The robustness test of the proposed control strategy is also accomplished by

TABLE 5 The cost function of PASSIVFE, SMC, and FSTSMC

Amplitude (m)	Cost (m^2) [passive]	Cost (m^2) [SMC]	Cost (m^2) [FSTSMC]
0.01	0.0085	0.0011	7.3737e-05
0.02	0.0394	0.0054	9.6926e-04
0.03	0.0998	0.0128	7.5255e-04
0.04	0.1967	0.0241	4.1248e-04
0.05	0.3462	0.0403	0.0012
0.06	0.5882	0.0631	0.0028
0.07	1.0235	0.0956	0.0059

applying random road profile with different amplitudes and fixed frequency. Simulation results show good robustness properties for FSTSMC when it is compared with passive and with SMC under the same conditions. The main contributions of this article are the design of an optimal robust FSTSMC scheme with PID controller using a ABC algorithm with a new convergence proof and application of the proposed control scheme to full vehicle model with eight degrees of freedom with an excellent result. The work can be extended and applied for full vehicle with passenger seat model with ten degrees of freedom in future work.

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