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Adaptive Neuro-Fuzzy Sliding Mode Control Based Strategy For Active Suspension Control

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***Abstract*—Suspension system of a vehicle is used to minimize the effect of different road disturbances on ride comfort and to improve the vehicle control. A passive suspension system responds only to the deflection of the strut. While, the semiactive system setup can dissipate energy from the system at an appropriate time, in a way or amount that is right for all the variables in the system. The main objective of this work is to design an efficient active suspension control for full car model with 8-Degrees of Freedom (DOF) using adaptive softcomputing technique. So, in this study, an Adaptive Neuro-Fuzzy based Sliding Mode Control (ANFSMC) is used for full car active suspension system to improve the ride comfort and vehicle stability. ANFSMC is adapted in such a way as to estimate online the unknown dynamics and provide feedback response. The detailed mathematical model of ANFSMC has been developed and successfully applied to a full car model. The robustness of the presented ANFSMC has been proved on the basis of different performance indices. The analysis of MATLAB/SMULINK based simulation results reveals that the proposed ANFSMC has better ride comfort and vehicle handling as compared to passive or semi-active suspension systems.**

***Keywords*-Sliding Mode Control; Fuzzy Logic; Neural Network; Active Car Suspension;**

I. INTRODUCTION

Suspension system is a common property of all vehicles. Suspension system isolates the vehicle body from the road disturbances to improve ride comfort and good road-handling. Ride comfort and good road-handling performance of a vehicle are generally analyzed by the damping feature of the shock absorbers. A vehicle suspension may be classified as passive suspension, semi-active suspension and active suspension system. Passive suspension system comprises springs and shock absorbers [1]. The springs are supposed to have a linear feature and shock absorbers exhibit nonlinear affiliation between force and velocity. So, in passive suspension systems, these components have fixed characteristics and have no means for feedback control. Whereas, the semi-active suspension system changes the damping coefficient by the electromagnetic regulator inside the absorber. The important feature of the active suspension system is that a peripheral power source is applied to attain the desired suspension objective. The actuator in an active suspension system is placed as a secondary suspension system for the vehicle. The controller drives the actuator, which depends on the proposed control law. The active suspension system gives the freedom to tune the whole suspension system, and the control force can be initiated locally or globally depending on the system state. Also, the active suspension system has the supplementary advantages, because, in this suspension system the negative damping can be afforded and a large range of forces can be produced at low velocities. Thus, these forces potentially allocate improvement in the system performance. The selection of control strategy is very important in the active suspension system. With proper control strategy, it will present improved coalition between ride comfort and vehicle stability. Initially many researchers assumed that the vehicle models are linear, but these models have some nonlinearities, like dry friction on the dampers. These nonlinearities can affect the vehicle stability and ride comfort.

In the last few years, researchers applied different linear control methods and nonlinear control methods to the vehicle suspension models. The best performance assessments of variable suspension system on a quarter car model are examined by [2], but only gave information about the heave of the model and seems overlooked the rattle space limit. Various linear control techniques are applied on a quarter car model [17], but did not show any information for large gain from road disturbance to car body acceleration. The passenger suspension seat was taken into account in their control technique by [15]. The quarter car model described the passenger suspension seat with nonlinearities like shock absorber damping, bump stops and linkage friction. Since this model does not give the sufficient information about the vehicles angular motion, therefore, vibration control and dynamic behavior of a half-car suspension model is investigated by various researchers in [5]. [6] used optimal control laws to compare the performances of the passive suspension system and active system suspension on quarter-car model, half-car model and full-car model [7]. applied PID controller on an active suspension system. A study on

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nonlinear active suspension system [8], [9] were carried out. Some comparison of PID controlled active suspension and sliding mode controlled active suspension is proposed [10]. Also PI sliding mode control [12] is applied on an active car suspension system. [16] examined the active control of seat vibrations of a vehicle model using various suspension alternatives.

PID control technique is pertained as a conventional law. Since this control technique can be applied broadly and widely, it has performed significant role in control applications. But, this control method is sensitive to parameter changes. If the plant changes its parameters due to uncertainties, then PID controller cannot update their parameters according to plant parameters. On the other hand, the adaptive PID controller can update its parameters according to the models parameters. Because of this, adaptive PID controller has better performance than conventional PID control. But on the hand, adaptive PID control has not much nonlinearity as compared to fuzzy logic control to control the nonlinear system. Fuzzy logic control has been broadly considered and executed in different control systems [13], [14]. A major advantage for this kind of control approach is that a precise depiction of the system is not essential, the system relies on different sensors, vagueness intrinsic in the measurements acquired by the various sensors are obligatory. Fuzzy logic control is perfect for this sort of condition, since exact and accurate input is not necessary. Because, fuzzy logic control is a set of linguistic rules, which signifies human thoughts and arranges the estimation to resolve control approaches for the whole process. Due to these benefits, many researchers prefer to look into this kind of control approach to reduce the tradeoff between the ride comfort and vehicle stability. In [15], the authors used a fuzzy logic controller tuned to increase the ride comfort of the vehicle. A variety of simulations showed that the fuzzy logic control is efficient to give a better ride quality than other common control approaches for example, skyhook control [17].

[18] presented a sliding mode controller for active suspension systems. But this does not address the problem of robustness and models uncertainties. A variety of simulations showed that the fuzzy logic and observer-based fuzzy sliding-mode control is proficient to give a better ride quality than other common control approaches [19]. In this study, ANFSMC control technique is used to enhance the ride comfort and vehicle stability against the road disturbances. The passenger seat is also incorporated in the vehicle model to improve the passengers comfortability.

The paper is divided into 6 sections. Section II, discusses the full vehicles model. Section III discusses the control problem, Section IV, discusses ANFSMC for full car suspension control. Finally, simulation results and conclusion are given in section V and VI, respectively.

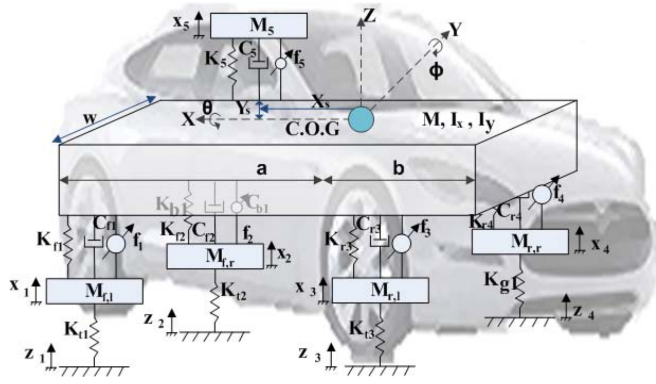

Figure 1. Full Car Model

II. Vehicle's Model

The full car suspension model is known to be a nonlinear system, which has eight-degrees of freedom. It comprises only a sprung mass attached to the four unsprung masses M_{fr}, M_{fl}, M_{rr}, M_{rl} (front-right, front-left, rear-right and rear-left wheels) at each corner. The sprung mass is allowed to have pitch, heave and roll and where the unsprung masses are allowed only to have heave. For simplicity all other motions are ignored for this model. This model has eight degrees of freedom and allocates the body acceleration and upright body displacement, pitch and roll motion of the vehicle body. Full car is professionally ensuring passenger safety and ride comfort. This model is considering only one seat and this is very important to take into consideration other fixed with chassis [16]. The eight degrees of freedom consists of $(x_1,\ x_2,\ x_3, x_4, x_5,\ x_6,\ x_7 = \theta,\ x8 = \phi)$ four wheels displacement, seat displacement, heave displacement, pitch displacement and roll displacement. The model of a full car suspension system is shown in figure 1. The suspensions between the sprung mass and unsprung masses are modelled as nonlinear viscous dampers and spring components and the tyres are modeled as simple nonlinear springs without damping elements. The actuator gives forces that determine the displacement of the actuator between the sprung mass and the wheels. The dampers between the wheels and car body signify sources of conventional damping like friction among the mechanical components. The inputs of full car model are four disturbances coming through the tyres, and the four outputs are the heave, pitch, seat, and roll displacement [20], [21], [22], [23]. The full car model will be used as a good approximation of the whole car.

A. Mathematical Modeling

The general class of nonlinear MIMO system is described by:

$$y^{(r)} = A(x) + \sum_{i=1}^{p}\sum_{i=1}^{s} B_{ij}(x)u_j + \sum_{i=1}^{p}\sum_{i=1}^{s} G_{ij}(x)z_j \qquad (1)$$

Where $x = [y_1, \dot{y}_1, ..., y_1^{r_1-1}, ..., y_p, \dot{y}_p, ..., y_p^{r_p-1}]^T \in R^r$ is the overall state vector, which is assumed available and $r_1+r_2+...+r_p = r$. $u = [u_1, u_2, ..., u_s]^T \in R^s$ is the control input vector, $y = [y_1, y_2, ..., y_p]^T \in R^p$ is the output vector and $z = [z_1, z_2, ..., z_s]^T \in R^s$ is the disturbance vector. $A_i(x),\ i = 1, ..., p$ are continuous nonlinear functions, $B_{ij}(x),\ i = 1, ..., p,\ j = 1, ..., s$ are continuous nonlinear control functions and $G_{ij}(x),\ i = 1, ..., p,\ j = 1,\ ...,\ s$ are continuous nonlinear disturbance functions.
Let

$$A = [A_1(x)\ A_2(x)\ ...\ A_p(x)]^T \tag{2}$$

The control matrix is:

$$B(x) = \begin{bmatrix} b_{11}(x) & \dots & b_{1s}(x) \\ \vdots & \ddots & \vdots \\ b_{p1}(x) & \dots & b_{ps}(x) \end{bmatrix}_{p \times s} \tag{3}$$

The disturbance matrix is:

$$G(x) = \begin{bmatrix} g_{11}(x) & \dots & g_{1s}(x) \\ \vdots & \ddots & \vdots \\ g_{p1}(x) & \dots & g_{ps}(x) \end{bmatrix}_{p \times s} \tag{4}$$

$$y^{(r)} = [y_1^{(r_1)}, y_2^{(r_2)},\ ...,\ y_p^{(r_p)}]^T$$

$$y^{(r)} = A(x) + B(x).u + G(x).z \tag{5}$$

$$A(.) \in R^{p \times p};\ B(.) \in R^{p \times s};\ G(.) \in R^{p \times s}$$

The generic nonlinear car model is,

$$\dot{y} = f(x) + B(x).u + G(x).z \tag{6}$$

$$y = h(x) \tag{7}$$

$f(x) \in R^{(16 \times 16)}$, $B(x) \in R^{(16 \times 4)}$, $G(x) \in R^{(16 \times 4)}$, state vector $x \in R(16 \times 1)$, $u \in R(4 \times 1)$ and $z \in R(4 \times 1)$. The above matrices can be shown in state-space form, with state vector x that is also represented in row matrix form.

$$f(x) = [A_1(x)\quad A_2(x)\quad A_3(x)\quad ...\quad A_{16}(x)]$$

$$x = [x_1\quad x_2\quad x_3\quad ...\quad x_{16}]^T$$

$A_1(x)$ to $A_8(x)$ are velocity states and $A_9(x)$ to $A_{16}(x)$ are acceleration states of four tires, seat, heave, pitch and roll [16]. The disturbance inputs for each tire individually are represented in the form of z matrix.

$$z = [z_1\quad z_2\quad x_z\quad z_4]^T$$

z_n are n disturbances applied to full car model. u_n are n controllers output to full car model, to regulate the car model disturbances. y_n are n states of car. r_n are n desired outputs of the controller.

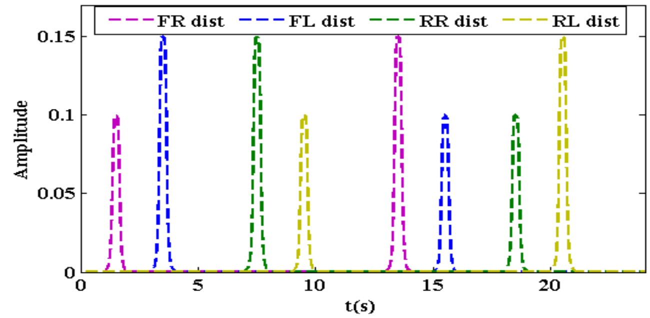

Figure 2. Road Profile

III. Control Problem

The aim of this control strategy is to improve the ride comfort and vehicle stability against the road disturbances. The comfort is examined by the vertical displacement and acceleration felt by the passenger. The lifetime of elements of vehicle is conserved by keeping away from hitting the rattle space limits, i.e., to stay away from the allowable peak to peak displacement of the system. Hence, the controller goal is to minimize the displacement and acceleration of the vehicle body with reference to the open-loop, to avoid the suspension travel should not hitting the rattle space limits. So, the controller performance is good when it reduces the vehicle vibrations under road disturbances. In this study, a road profile is expressed as.

$$z(t) = \begin{cases} a(1 - cos8\pi t) & 1 \leq t \leq 2, 9 \leq\leq 10 \\ & 15 \leq t \leq 16\ and\ 17 \leq t \leq 18 \\ (b1 - cos8\pi t) & 3 \leq t \leq 4, 7 \leq\leq 8 \\ & 13 \leq t \leq 14\ and\ 20 \leq t \leq 21 \\ 0 & otherwise \end{cases}$$

Where $a = 0.10\ m$ and $b = 0.15\ m$ are the amplitudes of two different bumps on the road. These road profiles are very helpful for observing the heave, pitch and roll of the vehicle. Figure 2 shows the road profile.

The control problem is that the suspension travel should be less than the amplitude of disturbance i.e., $0.15\ m$. The maximum displacement of the road profile is $0.15\ m$. The time delay between front and rear wheels is given by:

$$\delta(t) = \frac{(s_1 + s_2)}{V} \tag{8}$$

Where $s_1 = 1.2\ m$ and $s_2 = 1.4\ m$ are the values of distance between front wheels and rear wheels and V is the vehicle velocity, which is uniform.

IV. ADAPTIVE SLIDING MODE CONTROL BASED NEURO-FUZZY STRATEGY

As sliding mode control is a nonlinear control strategy, which can provide robust state feedback for the nonlinear dynamic systems. The sliding mode control forces the systems state to stay on the switching surface. When the system states reach the sliding surface, then, the system stay insensitive to internal constraint oscillations and irrelevant disturbances [28]. The sliding mode control structure should satisfy two requirements, i.e., the closed-loop stability and performance specifications. Consider the following sliding mode surface by using a sign function, i.e.,

$$q = ksgn(s) \tag{9}$$

Where k is a constant and it is the maximal value of the controller output. s is called switching function, because, the control action switches its sign on the two sides of the switching surface $s = 0$. s is defined as [29], [31].

$$s = \dot{e} + \lambda e \tag{10}$$

where $e = y - y_{ref}$, here y and y_{ref} are the actual and desired states. λ is a constant, then $sgn(s)$ is a discontinuous function given as:

$$sgn(s) = \begin{cases} -1 & if \ s < 1 \\ 1 & if \ s > 1 \end{cases}$$

This control strategy will ensure that the system states move towards and stay on the sliding surface, $s = 0$ from any initial condition if the following condition meets

$$s\dot{s} \leq -\tau \mid s \mid \tag{11}$$

where τ is a positive constant that ensures the system trajectories will meet the sliding surface.
The aforementioned sign function for the sliding mode controller structure often cause chattering in practice. Chattering is undesirable, because, it may excite the high frequency response of the system. In order to resolve the chattering issue, a boundary layer around the sliding surface is introduced [32].

$$q = q_s \tag{12}$$

Where

$$q_s = k.sat(\frac{s}{\psi})$$

where ψ is a constant which denotes the thickness of the boundary layer and $sat(\frac{s}{\psi})$ represents the saturation function which is defined as:

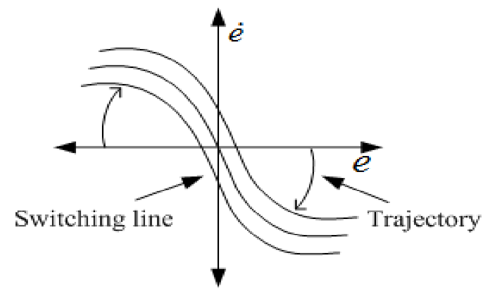

Figure 3. A nonlinear switching surface

$$sat(\frac{s}{\psi}) = \begin{cases} \frac{s}{\psi} & if \mid \frac{s}{\psi} \mid \leq 1 \\ sgn(\frac{s}{\psi}) & if \mid \frac{s}{\psi} \mid > 1 \end{cases}$$

This controller is actually a continuous approximation of the ideal relay control [29], [30]. The result of this control scheme is that invariance of sliding mode control is lost. The system robustness is a function of the width of the boundary layer. A variation of the above controller structure is to use a hyperbolic tangent function instead of a saturation function [33], [34].

$$q = k.tanh(\frac{s}{\psi}) \tag{13}$$

or it can be written as

$$q = k.tanh\left(\frac{\dot{e} + \lambda e + c}{\psi}\right) \tag{14}$$

where ψ and c are the constant and λ is the thickness of the sliding surface. It is proven that if k is large enough, the sliding mode controllers of (9), (12) and (14) are guaranteed to be asymptotically stable. The nonlinear switching curve is shown in figure 3. This nonlinear curve shows that the switching band around the switching line is there to alleviate chattering.

A. ANFSMC Structure

The TSK fuzzy system is fundamentally adaptive and nonlinear in nature, which provides robust performance for the parameter variations and load disturbances. The fundamental idea of the fuzzy modeling was given by Zadeh in [35]. The proposed ANFSMC connects TSK fuzzy logic system with sliding mode functions. In the ANFSMC, linear function or constant in the consequent part of the linguistic rules in TSK fuzzy systems are replaced with hyperbolic tangent function to enhance the estimation power of the neuro-fuzzy system by using the information of hyperbolic tangent function. Each rule in a TSK fuzzy logic control can be a sliding mode controller. The sliding mode controller in each rule can have various forms. The boundary layer and the coefficients of the sliding surface become the coefficients of the rule output function and have their physical meanings. The ith fuzzy sliding mode rule can be expressed as, i.e.,

$$IF\ x_1\ is\ A_{i1}\ AND\ x_2\ is\ A_{i2}\ AND\ ...\ AND\ x_i\ is\ A_{nm} \quad (15)$$
$$THEN\ y_m\ is\ k.tanh(\frac{s}{\psi}),\ i = 1, ..., m$$

Where $x_1, x_2, ..., x_m, y_1, y_2, ..., y_m$ are the input-output variables and A_{ij} is the membership function of ith rule and jth input. The ANFSMC structure is given in figure 4. In the antecedent part, fuzzy reasoning process is performed and in the consequent part of the rule sliding mode control process is performed.

Layer 1: In this layer fuzzy reasoning process is performed. This layer accepts input values. Its nodes transmit input values to the next layer.
Layer 2: In this layer fuzzification process is performed and neurons represent fuzzy sets used in the antecedents part of the linguistic fuzzy rules. The outputs of this layer are the values of the membership functions. Then Gaussian membership function is given by:

$$\eta_j(x_i) = e^{-\frac{(x_i - g_{ij})^2}{\sigma_{ij}^2}}, \quad i = 1, 2, ..., m, \quad j = 1, 2, ..., n \quad (16)$$

Where $\eta_j(x_j)$ shows the membership function, g_{ij} and g_{ij} are the mean and variance of membership function of the jth term of ith input variable, m and n are the number of input signals and number of nodes in second layer, respectively.
Layer 3: In this layer each node represents a fuzzy rule. In order to compute the firing strength of each rule, and min operation is used to estimate the output value of the layer. i.e.,

$$\mu_j(x) = \prod_i \eta_j(x_i) \quad (17)$$

Where $\prod_i$ is the meet operation and $\mu_j(x)$ are the input values for the next layer (consequent layer).
Layer 4: In this layer, hyperbolic tangent function are represented.
Layer 5: This layer estimates the weighted consequent value of a given rule,i.e., the hyperbolic tangent function are multiplied with the third layers output value. Therefore, the output value for this layer is given by:

$$p_l = w_l k.tanh\left(\frac{s}{\psi}\right) \quad (18)$$

Layers 6,7: In these layers, the defuzzification process is performed, i.e.,

$$u = \frac{\sum_{l=1}^{n} \mu_i(x) p_l}{\sum_{l=1}^{n} \mu_i(x)} \quad (19)$$

Where u is the output for the entire network.

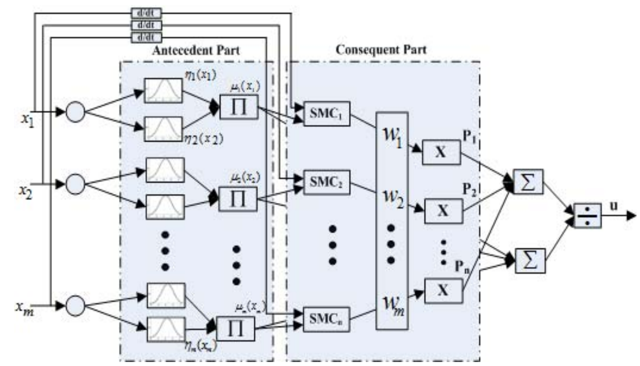

Figure 4. Schematic diagram of back-propagation learning algorithm for ANFSMC

1) Parameters Updating Learning Rules: The ANFSMC learning is to minimize a given function or input and output values by adjusting network parameters. Unknown parameters are mean, g_{ij} and variance, σ_{ij} of membership functions in antecedent part, λ_l, c_l, ψ_l and w_l are the update parameters in the consequent part of the rules. In this study, the gradient descent technique is used reduce the cost function. To minimize the error between the actual output value of the system and the desired value, the gradient descent method can be expressed as:

$$J = \frac{1}{2}\sum_{i=1}^{n}(y_{ref} - y_i)^2 \quad (20)$$

Where y_{ref} and y_i are the desired and current output values of the system, respectively. The update parameters w_l, λ_l, c_l and ψ_l of the consequent part of network, and g_{il} and σ_{il} $(j = 1, 2, ..., n)$ of the antecedent part of the network can be formulated as:

$$w_l(t + l) = w_l(t) - \gamma\frac{\partial J}{\partial w_l} \quad (21)$$

$$\lambda_l(t + l) = \lambda_l(t) - \gamma\frac{\partial J}{\partial \lambda_l} \quad (22)$$

$$c_l(t + l) = c_l(t) - \gamma\frac{\partial J}{\partial c_l} \quad (23)$$

$$\psi_l(t + l) = \psi_l(t) - \gamma\frac{\partial J}{\partial \psi_l} \quad (24)$$

$$g_{ij}(t + l) = g_{ij}(t) - \gamma\frac{\partial J}{\partial g_{ij}} \quad (25)$$

$$\sigma_{ij}(t + l) = \sigma_{ij}(t) - \gamma\frac{\partial J}{\partial \sigma_{ij}} \quad (26)$$

Where γ represents the learning rate.
By using chain rule, the partial derivatives of the $\frac{\partial J}{\partial w_l}$, $\frac{\partial J}{\partial \lambda_l}$, $\frac{\partial J}{\partial c_l}$, $\frac{\partial J}{\partial \psi_l}$, $\frac{\partial J}{\partial g_{ij}}$ and $\frac{\partial J}{\partial \sigma_{ij}}$ can be expressed as:

$$\frac{\partial J}{\partial w_l} = \frac{\partial J}{\partial y}\frac{\partial y}{\partial u}\frac{\partial u}{\partial p_l}\frac{\partial p_l}{\partial w_l} \tag{27}$$

$$\frac{\partial J}{\partial \lambda_l} = \frac{\partial J}{\partial y}\frac{\partial y}{\partial u}\frac{\partial u}{\partial p_l}\frac{\partial p_l}{\partial \lambda_l} \tag{28}$$

$$\frac{\partial J}{\partial c_l} = \frac{\partial J}{\partial y}\frac{\partial y}{\partial u}\frac{\partial u}{\partial p_l}\frac{\partial p_l}{\partial c_l} \tag{29}$$

$$\frac{\partial J}{\partial \psi_l} = \frac{\partial J}{\partial y}\frac{\partial y}{\partial u}\frac{\partial u}{\partial p_l}\frac{\partial p_l}{\partial \psi_l} \tag{30}$$

$$\frac{\partial J}{\partial g_{ij}} = \frac{\partial J}{\partial y}\frac{\partial y}{\partial u}\frac{\partial u}{\partial \mu_j}\frac{\partial \mu_j}{\partial g_{ij}} \tag{31}$$

$$\frac{\partial J}{\partial \sigma_{ij}} = \frac{\partial J}{\partial y}\frac{\partial y}{\partial u}\frac{\partial u}{\partial \mu_j}\frac{\partial \mu_j}{\partial \sigma_{ij}} \tag{32}$$

Where the quantity $\frac{\partial y}{\partial u}$ is approximated by a constant r [26], [27]. By taking the derivative of the above equations, it give

$$\frac{\partial J}{\partial w_l} = er\frac{\mu_j}{\sum_j \mu_j} ktanh\left(\frac{\dot{e} + \lambda_i e + c_i}{\psi_i}\right) \tag{33}$$

$$\frac{\partial J}{\partial \lambda_l} = e^2 r\frac{\mu_i}{\psi_i \sum_i \mu_j} w_i ksec^2h\left(\frac{\dot{e} + \lambda_i e + c_i}{\psi_i}\right) \tag{34}$$

$$\frac{\partial J}{\partial c_l} = er\frac{\mu_i}{\psi_i \sum_i \mu_j} w_i ksec^2h\left(\frac{\dot{e} + \lambda_i e + c_i}{\psi_i}\right) \tag{35}$$

$$\frac{\partial J}{\partial \psi_l} = -er\frac{\mu_i}{\psi_i \sum_i \mu_j} w_i ksec^2h\left(\frac{\dot{e} + \lambda_i e + c_i}{\psi_i}\right) \left(\frac{\dot{e} + \lambda_i e + c_i}{\psi_i^2}\right) \tag{36}$$

$$\frac{\partial J}{\partial g_{ij}} = er\frac{y_j - u}{\sum_i \mu_j}\mu_i(x_i)\frac{2(x_i - gij)}{\sigma_{ij}^2} \tag{37}$$

$$\frac{\partial J}{\partial g_{ij}} = er\frac{y_j - u}{\sum_i \mu_j}\mu_i(x_i)\frac{2(x_i - gij)^2}{\sigma_{ij}^3} \tag{38}$$

Hence the equations $(33-38)$ give the required change for the values of $\frac{\partial J}{\partial w_l}$, $\frac{\partial J}{\partial \lambda_l}$, $\frac{\partial J}{\partial c_l}$, $\frac{\partial J}{\partial \psi_l}$, $\frac{\partial J}{\partial g_{ij}}$ and $\frac{\partial J}{\partial \sigma_{ij}}$ respectively.

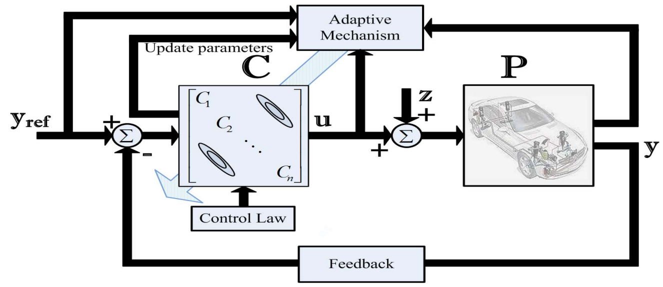

Figure 5. Closed-loop control structure for active suspension

B. Online Adaptive Neuro-fuzzy Sliding Mode Control Algorithm

Figure 5 shows the closed-loop control structure for the proposed ANFSMC control. This structure is employed to update the parameters of the ANFSMC control. The calculations at any instant of time can be described in the following steps.
Step 1: Set the input-output, yref and y of the system.
Step 2: Update the parameters of the ANFSMC scheme, i.e., g_{ij} , σ_{ij} , w_l, λ_l, c_l and ψ_l by using equations $(21-26)$.
Step 3: Calculate the output of the ANFSMC scheme by using equation (19).
Step 4: Finally, output of the controller added with disturbances is given to system.
Step 5: Repeat steps (2-4) until solution converges.

C. Simulation Results

In this study, it is assumed that the vehicle is moving with uniform velocity unless the road disturbances create the undesired oscillations in the vehicle body. The closed-loop structure for full car suspension control is given in figure 5.

In order to fulfill the aim of the active suspension system, the proposed active suspension strategies are successfully implemented on full car suspension system to improve the vehicles stability and passengers comfort. The comfort is examined by the vertical displacement and acceleration felt by the passenger. The controllers goal is to minimize the displacement and acceleration of the vehicle body with reference to the open-loop, so, as to avoid the suspension travel hitting the rattle space limits. So, the controller performance is good when it reduces the vehicle vibrations under road disturbances. In this section, the simulation results of displacement and acceleration of the heave, pitch, roll and seat (with and without controller) are given. These results are compared with passive suspension and semi-active suspension systems. The simulation time for the road profile is 24 $seconds$.
The performance index (PI) used for evaluation of different

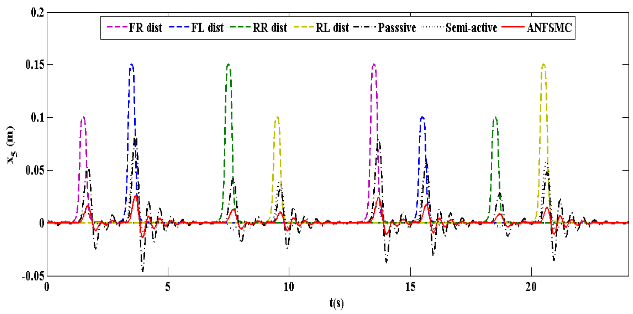

Figure 6. Seat displacement without control

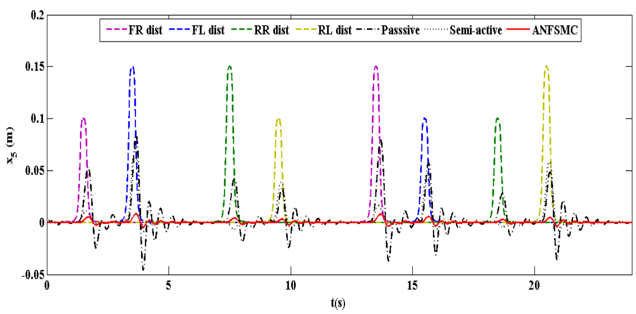

Figure 7. Seat displacement with control

algorithms is given by,

$$PI = \frac{1}{2}\int_{0}^{T}(Z_P^T Q Z_P)dt \tag{39}$$

where, Z_p is the vector for displacement or acceleration, Q is the identity matrix. The Root Mean Square (RMS) value for displacement and acceleration of heave, pitch, roll and seat has been calculated by,

$$z_{disp.}^{rms} = \sqrt{\frac{1}{2}\int_{t=0}^{T}[h(t)]^2} \tag{40}$$

$$\ddot{z}_{disp.}^{rms} = \sqrt{\frac{1}{2}\int_{t=0}^{T}[h(t)]^2} \tag{41}$$

Figures $(6-7)$ show that the response of seat with control (wc) and seat without control (woc) is improved as compared to passive suspension and semi-active suspension system. In passive suspension and semi-active suspension, the maximum value for seat displacement is $0.084\ m$ and $0.062\ m$ while, for the ANFSMC control, the maximum value for seat displacement is $0.025\ m$. Here, the passenger comfort is increased by 71% as compared to passive suspension and 60% as compared to semi-active suspension system. The response of seat with controller is better than seat without controller. Also, the settling time of ANFSMC controller is reduced and steady state response is improved as compared

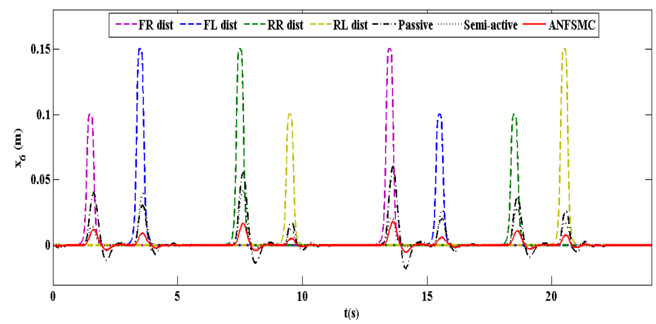

Figure 8. Heave displacement

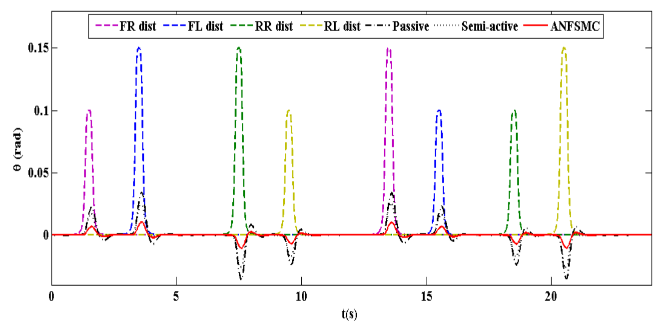

Figure 9. Pitch displacement

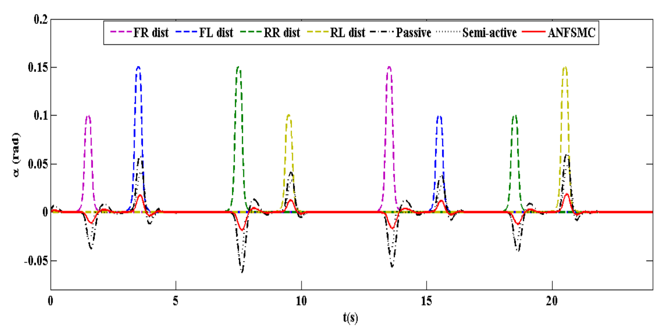

Figure 10. Roll displacement

to passive suspension. The seat with controller increased the passenger comfortability by 10%.

Figures $(9-10)$ show that the response of heave, pitch and roll is improved as compared to passive suspension and semiactive suspension system. In passive suspension and semiactive suspension, the maximum value of displacement for heave is $0.065\ m$ and $0.045\ m$ while, for the ANFSMC control, the maximum value of displacement for heave is $0.019\ m$. Here, the value of heave is improved by 79% as compared to passive suspension and 68% as compared to semi-active suspension system. In passive suspension and semiactive suspension, the maximum value of displacement for pitch is $0.035\ m$ and $0.029\ m$ while for the ANFSMC control, the maximum value of displacement for pitch is

0.013 m. Here, the value of pitch is improved by 70% as compared to passive suspension and 57% as compared to semi-active suspension system. In passive suspension and semi-active suspension, the maximum value of displacement for roll is 0.062 m and 0.045 m while, for the ANFSMC control, the maximum value of displacement for roll is 0.020 m. Here, the value of roll is improved by 68% as compared to passive suspension and 56% as compared to semi-active suspension system. This increased the vehicle stability, ride comfort and passenger comfortability. Also, the settling time of ANFSMC controller is reduced and steady-state response is improved as compared to passive suspension.

Table I
PERFORMANCE COMPARISON

DOFs	Control Algo.	$\ddot{z}^{rms}_{disp.}$	$\ddot{z}^{rms}_{acc.}$	**PI**
Seat (woc)	Passive	0.0166	1.5509	1.2026
	Semi-active	0.0134	1.1970	0.7165
	ANFSMC	0.0074	0.6317	0.1995
Seat (wc)	Passive	0.0166	1.5509	1.2026
	Semi-active	0.0134	1.1970	0.7165
	ANFSMC	0.0071	0.6017	0.1810
Heave	Passive	0.0117	0.5421	0.1470
	Semi-active	0.0099	0.4496	0.1011
	ANFSMC	0.0039	0.2683	0.0360
Pitch	Passive	0.0084	0.4593	0.1055
	Semi-active	0.0079	0.4584	0.1051
	ANFSMC	0.0021	0.2027	0.0265
Roll	Passive	0.0146	0.8168	0.337
	Semi-active	0.0139	0.7066	0.4993
	ANFSMC	0.0034	0.3197	0.0511

Table I shows the RMS values for displacement and acceleration, for the said road profile. It can be seen that maximum improvement has been achieved in case of seat, heave, pitch and roll with ANFSMC strategy.

V. CONCLUSION

In this study, the aim is to give the comfort ride and vehicle stability against the road disturbances. The detailed mathematical modeling of the full car suspension and proposed active suspension controllers are given. Simulation results show that ANFSMC control strategies for active suspension gives better ride comfort and vehicle stability than semi-active and passive suspension system. The parameters of the proposed ANFSMC have been adjusted using online adaptation by minimizing the cost function. The performance of the active control strategy is observed by the seat, heave, pitch and roll motion of the vehicle body. It is also observed that the ANFSMC control strategy is more robust and it improves the ride comfort and vehicle stability.

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