Neuro-fuzzy control of vehicle active suspension System

Haifa Souilem, Nabil Derbel

Abstract: Adaptative Neuro-fuzzy controller (ANFIS) is applied in order to control vibration of vehicle's suspensions for full suspension system which comes from road roughness. Moreover, the full vehicle system has seven degrees of freedom on the vertical direction of vehicle's chassis, on the angular variation around X- axis and on the angular variation around Y- axis. The approach of the proposed controller is to minimize vibrations which are made on the road roughness. On the other hand, standard PID controller is also used to control whole vehicle's suspension system for comparison. Consequently, random road roughnesses are used as disturbance of control system. Simulation results show that this control exhibited an improved ride comfort and good road holding ability and indicated that the proposed control system has superior performance at adapting random road disturbance for vehicle's suspension.

Keywords: PID controller, ANFIS controller, disturbances,ride comfort.

I. INTRODUCTION

A number of researchers have suggested control methods for vehicle suspension systems. The active suspension systems have come into commercial use, especially in the passenger car industry.

These modern systems offer improved comfort and road holding in varying driving and loading conditions compared to the matching properties achieved with traditional passive means. Most of the new systems are fitted in to large luxurious cars. However, these systems would be at their most advantageous in small size passenger cars and off-road vehicles. Some researchers designed a linear controller for a quarter or half vehicle [1],[2],[3],[4],[5],[6],[7],[8],[9]. In reference[10] the authors used a robust controller for a full vehicle linear active suspension system using the mixed parameter synthesis. A sliding mode technique is designed for a linear full vehicle active suspension system[11] . In reference[12] the authors presented the development of an integrated control system of active front steering and normal force control using fuzzy reasoning to enhance the full vehicle model handling performance. A fuzzy logic based fast gain scheduling controller is proposed for control nonlinear suspension systems for quarter vehicle system[13] . In fact, nonlinearity inherently exists in damper and spring models[14],[15],[16]. Therefore, the nonlinear effect should be inevitably taken into account to design the controller for practical active suspension system.

This paper will be developed a novel neuro fuzzy (NF) controller for full vehicle nonlinear active suspension systems. The full vehicle model will be investigated to take into account the seven degrees of freedom. A neurofuzzy model combines the features of a neural network and fuzzy logic model. A large class of neuro-fuzzy approaches utilizes the neural network learning algorithms to determine parameters of the fuzzy logic system[17] . The neuro-fuzzy system is more efficient and more powerful than either neural network or fuzzy logic system [18] which has been widely used in control systems, pattern recognition, medicine, expert system, etc.[19].

In this work, an adaptive neuro-fuzzy control system for whole vehicle's vibration control is proposed. The paper first describes the full vehicle suspension model under consideration. Second, the proposed control system and standard PID controller are outlined in Section III. Third, the results of proposed adaptative neuro fuzzy inference system (ANFIS) and PID control system are given and discussed. Finally, the effectiveness of the proposed control method is concluded in Section V.

II. FULL VEHICLE MODEL

To describe the vertical dynamics of a road vehicle which runs at a constant speed along an uneven road, 7 degrees of freedom (DOF) mathematical vehicle model is used and is shown in Fig. 1.

The model, including the relative displacement of unsprung masses and the front left suspension mass ($z_1' - z_1$), the rear left suspension mass ($z_2' - z_2$), the rear right suspension mass ($z_3' - z_3$), the front right suspension mass ($z_4' - z_4$) and the displacement of vertical motion of the

The authors are with the Control, Energy and Management Lab
National School of Engineers of Sfax BP.W,
3038, Sfax, Tunisia

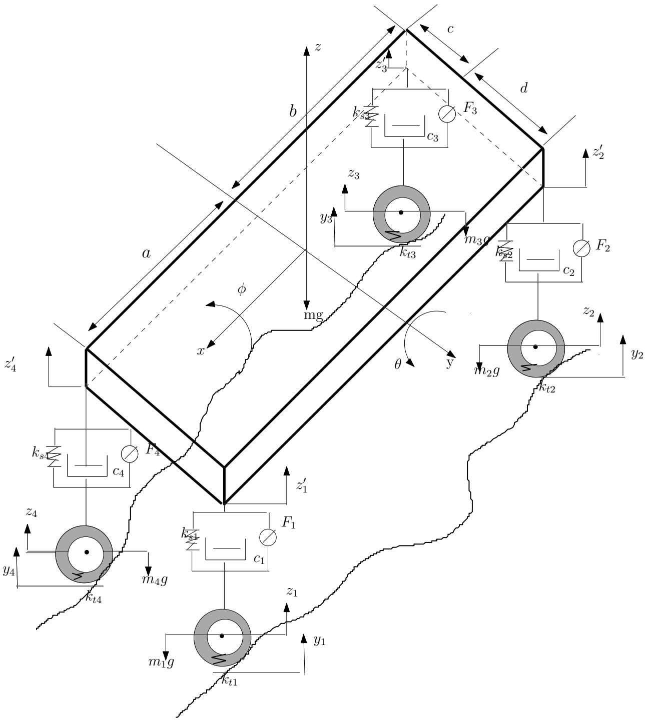

Figure 1. Full vehicle model

vehicle body (z) and including the angular displacement of roll (ϕ) and pitch (θ) motion of the vehicle body.

According to dynamic analysis, the equations for the full vehicle model are given by:

$$
\begin{aligned}
c_{t1}(\dot{z}_1 - \dot{y}_1) &= F_{t1} &(1)\\
c_{t2}(\dot{z}_2 - \dot{y}_2) &= F_{t2} &(2)\\
c_{t3}(\dot{z}_3 - \dot{y}_3) &= F_{t3} &(3)\\
c_{t4}(\dot{z}_4 - \dot{y}_4) &= F_{t4} &(4)
\end{aligned}
$$

In these equations, c_{t1}, c_{t2}, c_{t3}andc_{t4} denote the damping coefficients of the left front tyre, the left rear tyre, the right rear tyre and the right front tyre, respectively. z_1, z_2, z_3 and z_4 represent the vertical displacement of the left front wheel-axle, the left rear wheel-axle, the right rear wheel-axle and the right front wheel-axle, respectively. $y_1, y_2, y_3$and$y_4$denote the road disturbance input for left front wheel, the left rear wheel, the right rear wheel and the right front wheel, respectively. F_{t1}, F_{t2}, F_{t3}andF_{t4}delineate the left front wheel force, the left rear wheel force, the right rear wheel force and the right front wheel force, respectively

$$
\begin{aligned}
m_1\ddot{z}_1 &= k_{t1}(y_1 - z_1) + k_{s1}(z_1' - z_1) + c_1(\dot{z'}_1 - \dot{z}_1)\\
&+ F_1 + m_1 &(5)\\
m_2\ddot{z}_2 &= k_{t2}(y_2 - z_2) + k_{s1}(z_2' - z_2) + c_2(\dot{z'}_2 - \dot{z}_2)\\
&+ F_2 + m_2 g &(6)\\
m_3\ddot{z}_3 &= k_{t3}(y_3 - z_3) + k_{s3}(z_3' - z_3) + c_3(\dot{z'}_3 - \dot{z}_3)\\
&+ F_3 + m_3 g &(7)\\
m_4\ddot{z}_4 &= k_{t4}(y_4 - z_4) + k_{s4}(z_4' - z_1) + c_4(\dot{z'}_4 - \dot{z}_4)\\
&+ F_4 + m_4 g &(8)
\end{aligned}
$$

where $g = 9.81m/s^2$ is a gravitational acceleration. $m1, m2, m3$and$m4$ are the left front suspension mass, the left rear suspension mass, the right rear suspension mass and the right front suspension mass, respectively. $c1, c2, c3$and$c4$ denote the damping coefficients of the left front suspension, the left rear suspension, the right rear suspension and the right front suspension, respectively.k_{t1}, k_{t2}, k_{t3}andk_{t4}represent the left front tyre stiffness, the left rear tyre stiffness, the right rear tyre stiffness and the right front tyre stiffness, respectively. F_1, F_2, F_3andF_4delineate the left front active suspension force, the left rear active suspension force, the right rear active suspension force and the right front active suspension force, respectively

$$
\begin{aligned}
m\ddot{z} &= k_{s1}(z_1 - z_1') + k_{s2}(z_2 - z_2') + k_{s3}(z_3 - z_3')\\
&+ k_{s4}(z_4 - z_4') + c_1(\dot{z}_1 - \dot{z'}_1)\\
&+ c_2(\dot{z}_2 - \dot{z'}_2) + c_3(\dot{z}_3 - \dot{z'}_3)\\
&+ c_4(\dot{z}_4 - \dot{z'}_4) - F_1 - F_2 - F_3 - F_4 + mg \quad (9)
\end{aligned}
$$

where $m$is the vehicle body mass and z the vertical displacement of the vehicle body. k_{s1}, k_{s2}, k_{s3}andk_{s4} represent the stiffness of the left front suspension, the stiffness of the left rear suspension, the stiffness of the right rear suspension and the stiffness of the right front suspension, respectively

$$
\begin{aligned}
J_x\phi\ddot{z} &= -[k_{s3}(z_3 - z_3') + c_3(\dot{z}_3 - \dot{z'}_3) + k_{s4}(z_4 - z_4')\\
&+ c_4(\dot{z}_4 - \dot{z'}_4)]c[k_{s1}(z_1 - z_1') + c_1(\dot{z}_1 - \dot{z'}_1)\\
&+ k_{s2}(z_2 - z_2') + c_2(\dot{z}_2 - \dot{z'}_2)]d\\
&+ (F_3 + F_4)c - (F_1 + F_2)d \quad (10)
\end{aligned}
$$

In which, c is CG distance from right axle, d is CG distance from left axle, J_x is roll moment of inertia and φ is the roll angle

$$
\begin{aligned}
J_y\theta\ddot{z} &= -[k_{s1}(z_1 - z_1') + c_1(\dot{z}_1 - \dot{z'}_1) + k_{s4}(z_4 - z_4')\\
&+ c_4(\dot{z}_4 - \dot{z'}_4)]a[k_{s2}(z_2 - z_2') + c_2(\dot{z}_2 - \dot{z'}_2)\\
&+ k_{s3}(z_3 - z_3') + c_3(\dot{z}_3 - \dot{z'}_3)]b\\
&+ (F_1 + F_4)a - (F_2 + F_3)b \quad (11)
\end{aligned}
$$

where a is CG distance from front axle, b is CG distance from rear axle, J_y is pitch moment of inertia and $\theta$is the pitch angle.

$$
\begin{aligned}
z_1' &= z - (a\theta - d\phi) &(12)\\
z_2' &= z + (b\theta - d\phi) &(13)\\
z_3' &= z + (b\theta + c\phi) &(14)\\
z_4' &= z - (a\theta + c\phi) &(15)
\end{aligned}
$$

The differential equations can be written in state-space notation when the state vector X is defined as

$$
\begin{aligned}
\dot{X} &= AX + BQ &(16)\\
Y &= CX + DQ &(17)
\end{aligned}
$$

where Y is the output vector, Q is the input vector, A is the state matrix, B is the input matrix, C is the output matrix,

ISSN: 1998-4464

D is the feedforward matrix.

$$X = \begin{bmatrix} z_1 - z_1' \\ z_2 - z_2' \\ z_3 - z_3' \\ z_4 - z_4' \\ y_1 - z_1 \\ y_2 - z_2 \\ y_3 - z_3 \\ y_4 - z_4 \\ \dot{z}_1 \\ \dot{z}_2 \\ \dot{z}_3 \\ \dot{z}_4 \\ \dot{z} \\ \dot{\phi} \\ \dot{\theta} \end{bmatrix}, \quad (18)$$

$$Y = \begin{bmatrix} \ddot{z} \\ \ddot{\phi} \\ \ddot{\theta} \\ z_1 - z_1' \\ z_2 - z_2' \\ z_3 - z_3' \\ z_4 - z_4' \\ y_1 - z_1 \\ y_2 - z_2 \\ y_3 - z_3 \\ y_4 - z_4 \end{bmatrix}, \quad (19)$$

$$Q = \begin{bmatrix} \dot{y}_1 \\ \dot{y}_2 \\ \dot{y}_3 \\ \dot{y}_4 \\ g \\ F_1 \\ F_2 \\ F_3 \\ F_4 \end{bmatrix}, \quad (20)$$

III. CONTROL SYSTEMS

Two different control structures are used to control vibration of full vehicle model. These the PID controller and the developed(ANFIS) control system. The PID controller is used to compare the developed ANFIS control system. The next two subsections of this section present the PID controller and the developed ANFIS control system.

3.1. PID controller

PID controller consists of proportional $P(e(t))$,integral $I(e(t))$ and derivative $D(e(t))$ parts. Assuming that each amplitude is completely decouped and controlled independently from other amplitudes, the control input $u(t)$ is given by:

$$u(t) = K_p e(t) + K_I \int e(t)dt + K_D \frac{de(t)}{dt} \quad (21)$$

In equation, e(t) is the control error:

$$e(t) = x_d(t) - x_a(t) \quad (22)$$

where $x_d(t)$ is the desired response and$x_a(t)$ is the actual response.K_p is called the proportional gain, K_I the integral gain and K_D the derivative gain.Zeigler-Nicholas methods are used to determine the optimum PID gain parameters.

3.2. Adaptative neuro-fuzzy inference control system

The ANFIS is one of the methods to organize the fuzzy inference system with given input/output data pairs. The ANFIS is a combination of a fuzzy logic controller and a neural network, which makes the controller self tuning and adaptive. If we compose these two intelligent approaches, it will be achieve good reasoning in quality and quantity. This technique gives the fuzzy logic capability to adapt the membership function parameters that best allow the associated fuzzy inference system to track the given input/output data.

The formal analogy between a fuzzy inference system and a multilayer neural network associated with optimization algorithms is used from the retro-propagation gradient algorithm have winded up in what is called a STFIS Network.
The following figure shows a control architecture which uses only one network as a controller, the learning of which is done directly by the backpropagation of the output.

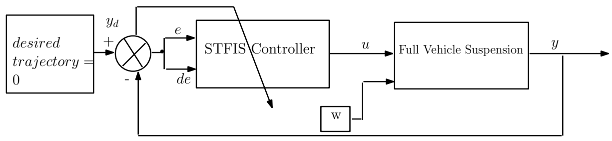

Figure 2. STFIS controller architecture

3.2.1. Presentation of (ANFIS)

A sugeno type fuzzy system is determined in three stage [20]:

1. Given an input x a membership degree μ is obtained from the antecedent part of rules.

2. A truth value degree α_i is obtained,associated to the premises of each rule R_i:**if**$x_1$is X_1 and**if**x_2 is X_1 **then** u is w_i

3. An aggregation stage to take in to account all rules by $u = \sum_{i=1}^{r} \alpha_i w_i / \sum_{i=1}^{r} \alpha_i$

These stages can be traduced by the 4 layers structures shown in fig.2.Each layer, connected with others by adjustable parameters,having a specific function.

ISSN: 1998-4464

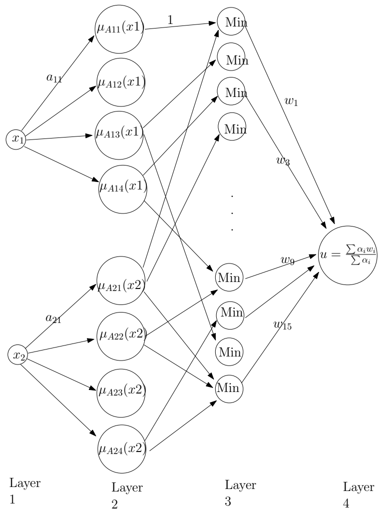

Figure 3. Self Tunable Fuzzy Inference System

3.2.2. Algorithm modification weight regression

In this work, we propose to generate the fuzzy control rules by an optimization method. The optimization of adjustable parameter is accomplished with a version of the classic gradient retro-propagation algorithm. The aim is to minimize cost function E:

$$E = \frac{1}{2}\Delta\epsilon^2 \qquad (23)$$

where, ϵ is the difference between set point and process output. The basic eqations of the algorithm are:

$$w_{ij}^n(t) = w_{ij}^n(t-1) + \Delta w_{ij}^n(t) \qquad (24)$$
$$\Delta w_{ij}^n(t) = -\eta\delta_i^n\alpha_j^{n-1} + b\Delta w_{ij}^n(t-1) \qquad (25)$$

where,$w_{ij}^n(t)$:i^{th} parameter between i of layer n and j^{st} unit of layer $n-1$; η:learning again;t:training iteration;b:moment parameter;$\eta\delta_i^n$:error term (i^{th} neurone of layer n);α_j^{n-1}:output of j^{th}unit of layer $n-1$.

The quality of solution obtained using this algorithm depends on input learning signals, algorithm control parameters and learning duration(number of iterations).
The procedure is entirely done on line on the actuator.The table of rules (weightsw_i) can be initially empty or filled with an a priori knowledge. The actuator acquires by its systems output measures, calculates the error to the back-propagated, updates the triggered rules on-line.The weights of the table of decision are then adjusted locally and progressively. The cost function is given by;

$$J = E + \lambda\Sigma w_i^2 \qquad (26)$$

where E is the classic quadratic error,w are the parameters (weights) to optimize parameters and λ is a constant that controls the growth of parameters. The second term in J is known as weight decay and used usually in the context of classification problems. This technique has been analyzed in the framework of learning theory and it was shown that it is very simple manner to implement a regularization method in a neural network in order to optimize the compromise between the learning error and the generalization error.Due to the classic back-propagation algorithm, the parameters as modify as:

$$w(t+1) = w(t) + \eta(\frac{-\partial J}{\partial w}) \qquad (27)$$

This algorithm easily includes the effect of the second term of the cost function J and by taking $\beta = 2\lambda\eta$ (regression coefficient) we obtain:

$$w(t+1) = w(t) + \eta(\frac{-\partial J}{\partial w}) - \beta\Delta w(t) \qquad (28)$$

Since a fuzzy inference system is concerned, we adapt this formula by multiplying β by the firing term of the rule, namely $\alpha_i/\sum\alpha_i$ is the truth value of the premise part of the triggered rule.
If we limit the optimization only on the conclusions parameters w_{1ij}^4. Then, we get:

$$\begin{aligned}\Delta w_{1j}^4(t) &= -\eta\delta_1^4\alpha_j^3 + b\Delta w_{1j}^4(t-1)\\ &- \frac{\alpha_j^3 2\eta\lambda w_{1j}^4(t-1)}{\sum_k \alpha_k^3}\end{aligned} \qquad (29)$$

where, $\delta_1^4 = y1 - y/\sum_j\alpha_j^3$;$y1$:effective output value;$y$:desired output.

IV. SIMULATION AND RESULTS

Simualation for controller of active half car suspension model is done by using MATLAB simulink. Two type of controllers are applied, they are PID controller which is tuned by Zigler-Nicholas and ANFIS controller.We have considered a total mass of the body equal to $m_b = 1020Kg$.

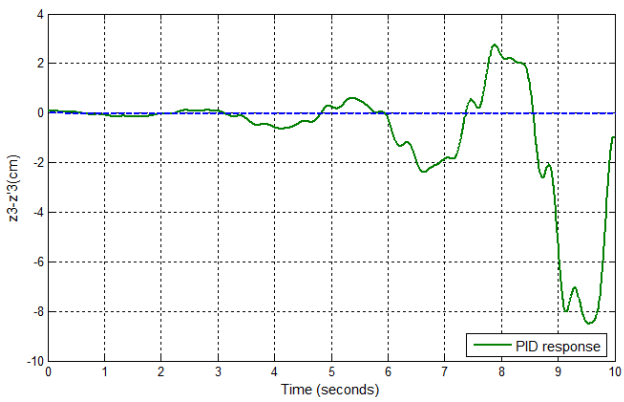

Figure 4. PID response of the rear right active suspension system of the vehicle for random road roughness input signal

Some series simulations were carried out to show the performance of the proposed neuro-fuzzy control system. Fig. 7 indicates response of the front left active suspension

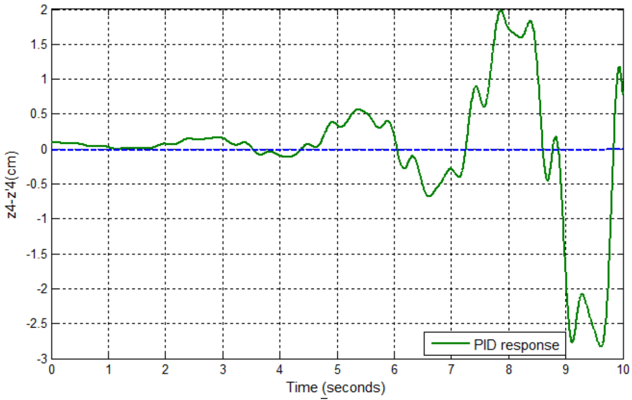

Figure 5. PID response of the front right active suspension system of the vehicle for random road roughness input signal

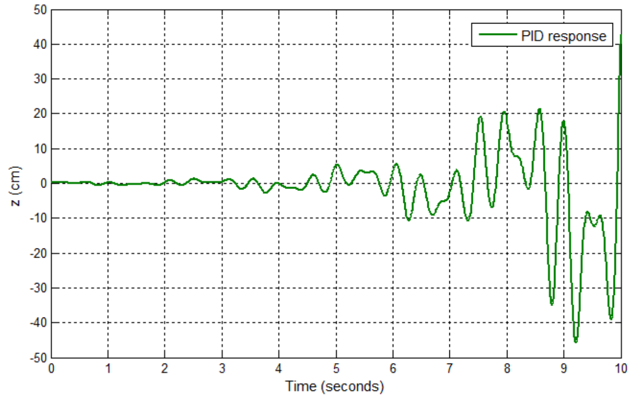

Figure 6. PID response of the displacement of vertical motion of the vehicle body for random road roughness input signal

system of the vehicle for random road profile without any controller. As seen in Fig. 7, because of any controller, the uncontrolled response does not follow the desired random road profile. Fig. 23 shows response of the front left active suspension system of the vehicle for random road profile using the standard PID controller. As can be seen figure, the result of the PID controller does not follow the desired random road profile signal. The result of the STFIS control system for the front left active suspension system of the vehicle given in Fig. 14. As depicted from Fig. 14, the proposed control system has good performance at adapting random road roughness. Fig.8,Fig.24 and Fig.15, give the results of without any controller, the PID controller and the proposed STFIS control system for the rear left active suspension system. As shown in Fig. 24, both the desired random road profile and the PID controller result are not followed. The different control system applied to control the rear right active suspension system of the vehicle and the results are shown in Fig.4,Fig.9 and Fig.16. It has been stated earlier that the STFIS control system exactly follows the desired random road profile. The result of without any controller and the PID controller for the front right active suspension system is shown in Fig. 10 and Fig.5. It is clear to see from graphs, there are differences between desired and PID controller. The results of this controller are also poor to control the front right active suspension system. Fig. 17 indicates the result of the STFIS control system and is not differences between the desired input signal and this developed control system results. Response of without any controller, the PID controller and the STFIS control system are given in Fig. 11,Fig.6 and Fig.18. As seen in relevant figure, there is a best approximation of the STFIS control system for the displacement of vertical motion of the vehicle. Fig. 12 and Fig.21. are presented response of the angular displacement of roll of the vehicle for random road profile without any controller and the PID controller. The results proved the PID controller is not suitable for controlling vehicle system vibrations. The results of the STFIS control system for the angular displacement of roll of the vehicle are given Fig. 19. As depicted from figure, there are zero errors between the STFIS structure and desired random profile signal. Lastly, Fig. 13, Fig.22 and Fig.20 represent the results of the control system for the angular displacement of pitch of the vehicle. From the simulation results, the developed control system has superior performance for vehicle vibration parameters.

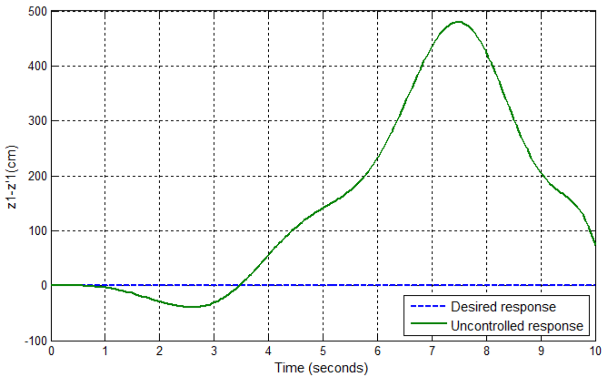

Figure 7. Uncontrolled response of the front left active suspension system of the vehicle for random road roughness input signal

V. CONCLUSIONS

In this paper, adaptative neuro-fuzzy control system for whole vehicle active suspension system parameters has been designed. The full vehicle model is considered seven degrees of freedom system. The results have been compared with PID controller and the corresponding system without controller. From these results, the neuro-fuzzy controller has capability of minimizing the control objectives better than the PID controller. The simulation results confirms the effectiveness and robustness of the proposed ANFIS control system. Finally, the performance of the ANFIS control system is better than standard PID controller

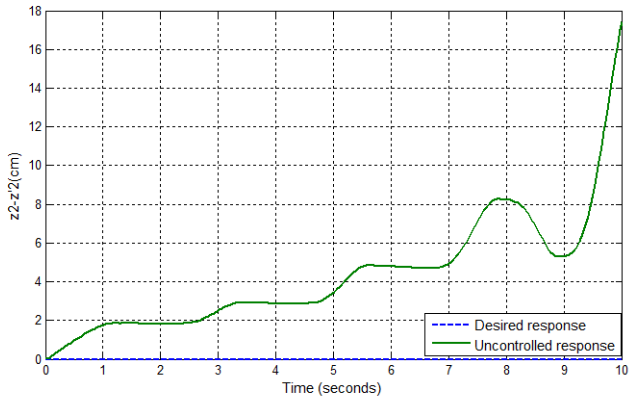

Figure 8. Uncontrolled response of the rear left active suspension system of the vehicle for random road roughness input signal

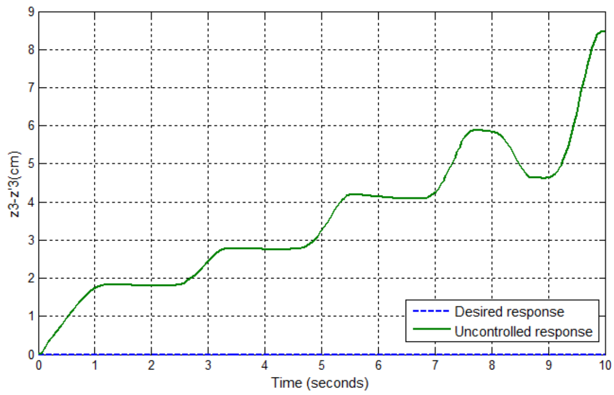

Figure 9. Uncontrolled response of the rear right active suspension system of the vehicle for random road roughness input signal

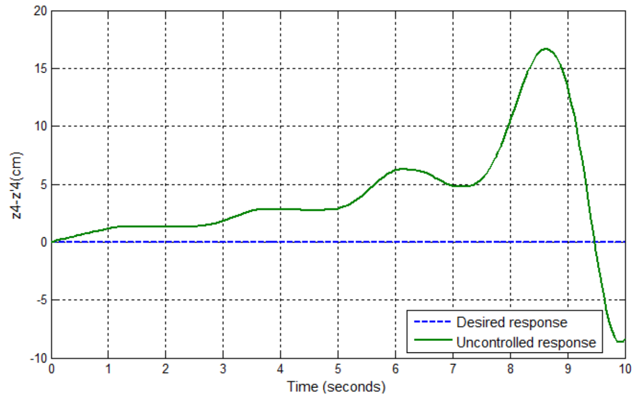

Figure 10. Uncontrolled response of the front right active suspension system of the vehicle for random road roughness input signal

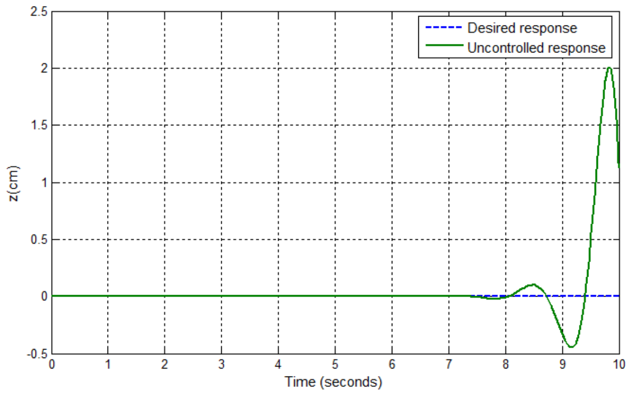

Figure 11. Uncontrolled response of the displacement of vertical motion of the vehicle body for random road roughness input signal

References:

[1] K. Sung, Y. Han, K. Lim and S. Choi. Discrete-time Fuzzy Sliding Mode Control for a Vehicle Suspension System Featuring an Electrorheological fluid damper. Smart Materials and Structures 16:pp. 798-808, 2007.

[2] Y. Kuo and T. Li. GA Based Fuzzy PI/PD Controller for Automotive Active Suspension System. IEEE Transactions on Industrial Electronics, Vol. 46, No. 6: pp.1051-1056, 1999.

[3] J. Feng and F. Yu. GA-Based PID and Fuzzy Logic Controller for Active Vehicle Suspension System. International Journal of Automotive Technology, Vol. 4, No. 4: pp. 181-191, 2003.

[4] M. Smith and G. Walker. Performance Limitations and Constraints for Active and Passive Suspensions: a Mechanical Multi-port Approach, Vehicle System Dynamics, Vol. 33, No. 3: pp.137-168, 2000.

[5] M. Biglarbegian, W. Melek and F. Golnaraghi. A Novel Neuro-fuzzy Controller to Enhance the Performance of Vehicle Semi-active Suspension Systems, Vehicle System Dynamics, Vol. 46, No.8: pp.691-711, 2008.

[6] M. Biglarbegian, W. Melek and F. Golnaraghi. Design of a Novel Fuzzy Controller to Enhance Stability of Vehicles, North American Fuzzy Information Processing Society: pp. 410-414, 2007.

[7] L. Yue, C. Tang and H. Li. Research on Vehicle Suspension System Based on Fuzzy Logic Control, International Conference on Automation and Logistics, Qingdao, China, 2008.

[8] M. Kumar. Genetic Algorithm-Based Proportional Derivative Controller for the Development of Active Suspension System, Information Technology and Control, Vol. 36, No. 1: pp.58-67,2007.

[9] Y. He and J. Mcphee. A design methodology for mechatronics vehicles: application of Multidisciplinary optimization, multimode dynamics and genetic algorithms, Vehicle System Dynamics, Vol. 43, No. 10:pp.697-733, 2005.

[10] P. Gaspar, I. Szaszi and J. Bokor. Design of Robust controller for Active vehicle Suspension Using the

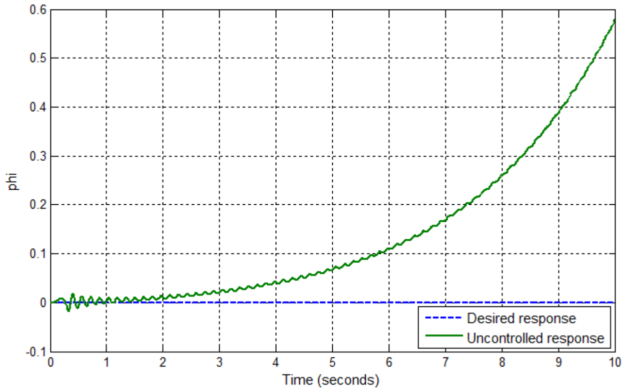

Figure 12. Uncontrolled response of the angular displacement of roll of the vehicle for random road roughness input signal

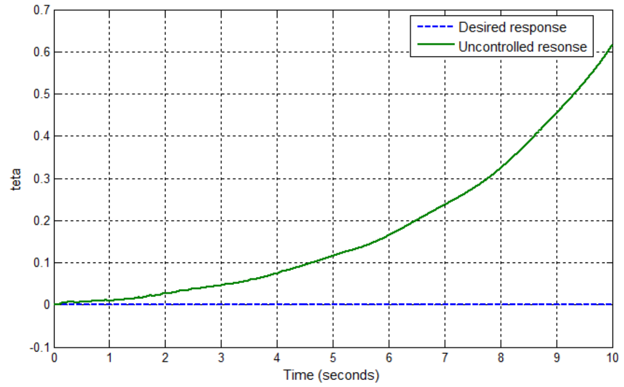

Figure 13. Uncontrolled response of the angular displacement of pitch of the vehicle for random road roughness input signal

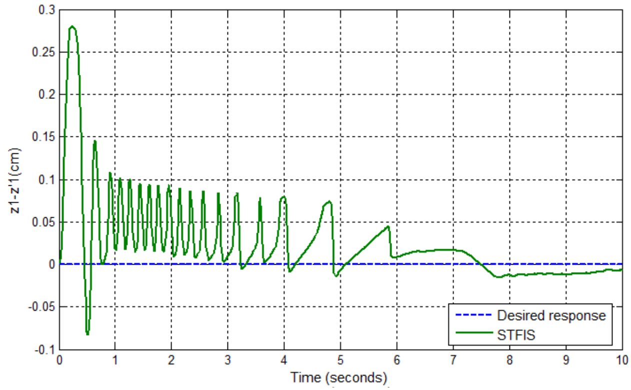

Figure 14. STFIS response of the front left active suspension system of the vehicle for random road roughness input signal

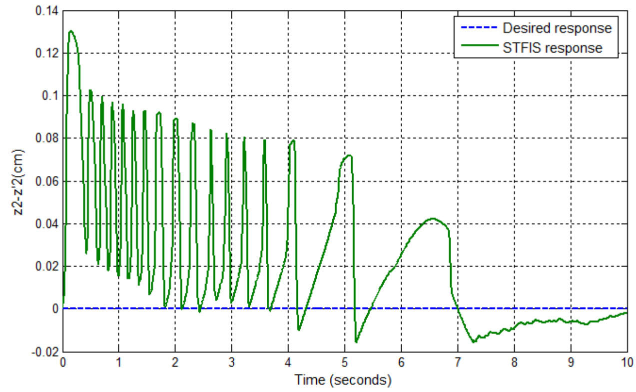

Figure 15. STFIS response of the rear left active suspension system of the vehicle for random road roughness input signal

Mixed Synthesis, vehicle Dynamic System, Vol. 40, no. 4: pp. 193-228, 2003.

[11] A. Chamseddine, H. Noura and T. Raharijana. Control of linear Full Vehicle Active Suspension System Using Sliding Mode Techniques, International Conference on Control Applications, Munich, Germany, 2006.

[12] C. March and T. Shim. Integrated Control of Suspension and front Steering to enhance Vehicle Handling, Processing IMechE, Vol. 221 Part D: pp. 377-391, 2006.

[13] S. Lee, G. Kim and T. Lim. Fuzzy logic Based Fast Gain Scheduling Control for Nonlinear Suspension System, IEEE Transaction on Industrial Electronics, Vol. 45, No.6, pp. 953-955, 1998.

[14] S. Li, S. Yang and W. Guo. Investigation on Chaotic Motion in Hysteretic Non-linear Suspension System with Multi-frequency Excitations, Mechanics Research Communication 31, pp. 229-236, 2004.

[15] J. Dixon. 'The Shock Absorber Handbook,'Society of Automotive Engineers, Inc., USA, p. chap, 1999.

[16] D. Joo, N. Al-Holou, J. Weaver, T. Lahdhir and F. Al-Abbas. Nonlinear Modelling of vehicle suspension System, 'Proceeding of the American Control Conference, Chicago, Illinois, 2000.

[17] C. Isik and M. Farrokhi. Recurrent neurofuzzy system, Annual meeting of the North American Fuzzy Information Processing Society Nafips, 1997.

[18] M. Brown and C. Harris. Neurofuzzy adaptive modeling and control, prentice hall international (UK) limited, 1994.

[19] Y. Zhang and A. Kandel. Compensatory neurofuzzy systems with fast learning algorithms, IEEE transactions on neural network, Vol. 9, No. 1: pp.80-105, 1998.

[20] T.Takagi and M. Sugeno,derivation of fuzzy control rules from human operator s control actions.In:Proc.IFAC Symp. On fuzzy information, knowledge representation and decision analysis,pp:55-60, July 1983.

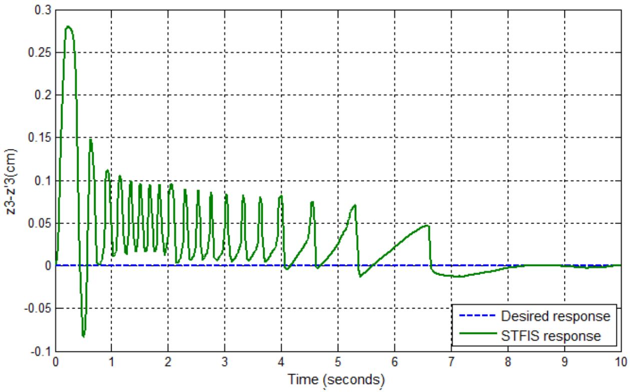

Figure 16. STFIS response of the rear right active suspension system of the vehicle for random road roughness input signal

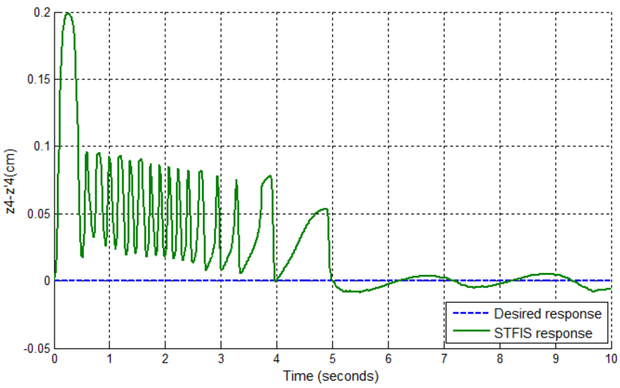

Figure 17. STFIS response of the front right active suspension system of the vehicle for random road roughness input signal

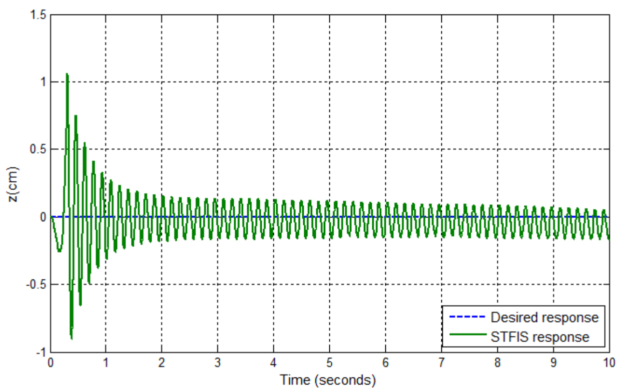

Figure 18. STFIS response of the displacement of vertical motion of the vehicle body for random road roughness input signal

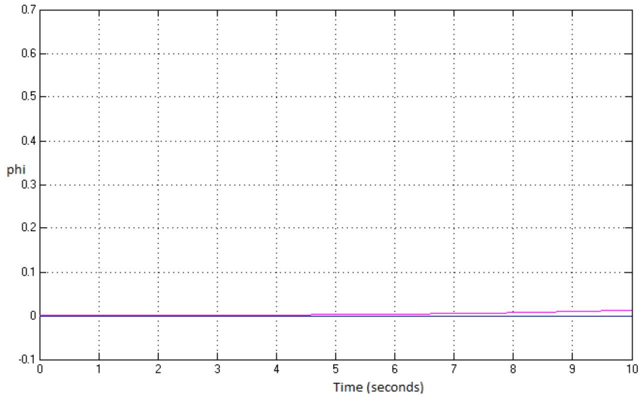

Figure 19. STFIS response of the angular displacement of roll of the vehicle for random road roughness input signal

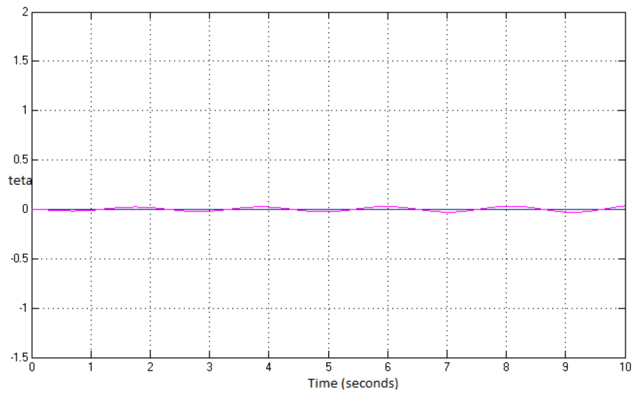

Figure 20. STFIS response of the angular displacement of pitch of the vehicle for random road roughness input signal

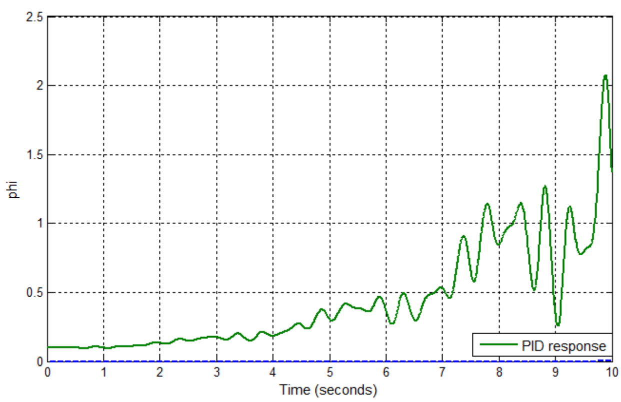

Figure 21. PID response of the angular displacement of roll of the vehicle for random road roughness input signal

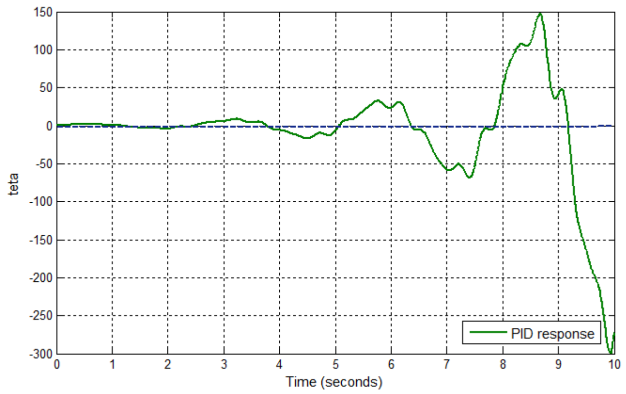

Figure 22. PID response of the angular displacement of pitch of the vehicle for random road roughness input signal

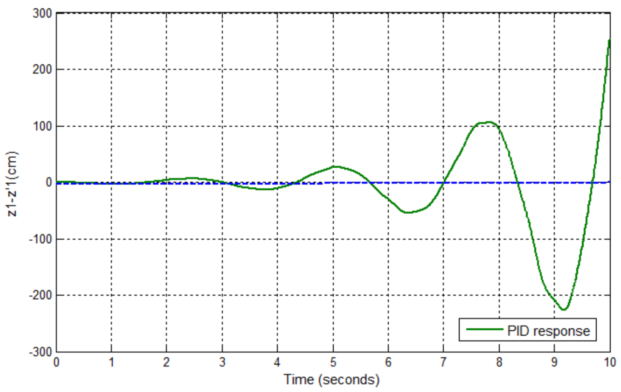

Figure 23. PID response of the front left active suspension system of the vehicle for random road roughness input signal

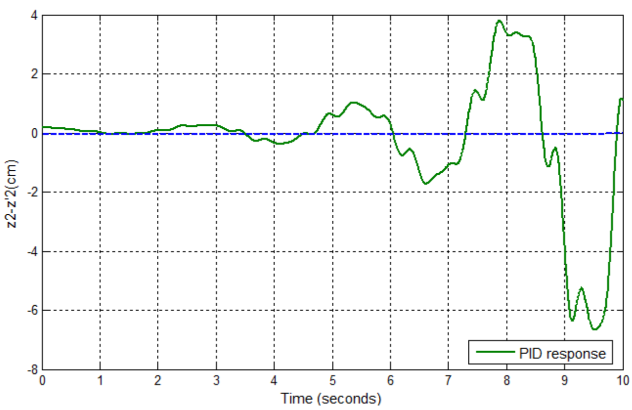

Figure 24. PID response of the rear left active suspension system of the vehicle for random road roughness input signal