

## Implementation of fuzzy logic control on a new low cost semi-active vehicle shock absorber

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**ABSTRACT** – This work implements a fuzzy logic control (FLC) on a proposed new low-cost semi-active shock absorber to improve vehicle ride comfort. The ordinary passive shock absorber is replaced with a new apparatus consisting of a conventional hydraulic cylinder with a proportional throttle valve placed outside the cylinder between its ports. FLC is used to adjust the damping coefficient by regulating the valve opening region. The fuzzy logic controller is configured using acceleration driven damping (ADD) methodology. Inputs are the accelerations of the sprung and the un-sprung masses, while the opening region of the valve is the output of the controller. Simscape/Matlab is used to build the model of the suggested semi-active shock absorber. The magneto-rheological (MR) suspension system and the suggested semi-active shock absorber with artificial neural network (ANN) controller are provided for comparison purposes. The findings demonstrated superior performance for the suggested shock absorber controlled by FLC relative to other controllers and to the ordinary damper as well, with one fourth of the cost of the magneto-rheological (MR) damper.

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## INTRODUCTION

Ride comfort and good vehicle handling are the main functions of the vehicle's suspension system [1, 2]. Suspension system designers usually strive to achieve these objectives while holding costs and energy consumption to a minimum [3]. Several suspension system types are commercially available and are being used by vehicle manufacturers depending on the type of vehicle and the purpose of vehicle usage. Each type of suspension system is characterized by the type of the shock absorber (damper) and each has its own pros and cons. The passive suspension, which is fitted with a passive damper, is mostly installed in almost all types of vehicles. Its low cost and simple construction make it the most economical option for all car manufacturers. Moreover, it avails a reasonable level of vehicle's suspension performance. The problem with such a system is that it has a constant damping coefficient. Passive suspension systems cannot therefore respond to the change in state of the vehicle that could be triggered by a change in the type of road or even a change in the distribution of weight within the vehicle. For these reasons, the performance of the vehicle with an ordinary system of suspension can be altered only if the damper is replaced by another one with a different damping coefficient.

In contrast, active suspension utilizes a hydraulic/pneumatic circuit that includes a linear actuator. The system is also equipped with road predicting sensors that can read the road profile in front of the vehicle. This combination works together in such a way that the actuator keeps the vehicle at a steady vertical level and completely isolates the vehicle from road vibrations. However, there are many problems with active suspension systems, such as an intensive power consumption that requires a considerable portion of the engine's power to drive the hydraulic components. In addition, it has a relatively high weight that increases the total weight of the vehicle. Moreover, the active suspension system is an exclusive choice for luxury vehicle manufacturers due to its high cost [4].

On the other side, suspension with a semi-active damper is a moderate system that closes the gap between the two previously mentioned suspension systems. The semi-active damper can vary its damping coefficient without the need to change the entire damper and doesn't consume much power. There are many types of semi-active dampers on the market, the most common of which is magneto-rheological (MR) dampers. It has the same outline construction as the ordinary passive shock absorber, except it is packed with magnetic oil. The viscosity of the oil can be changed via an electro-magnetic coil suited around the damper which then applies a certain value of magnetic flux that changes the viscosity of the oil. The suspension system may therefore adjust its damping characteristics to achieve riding comfort [5]. The drawback of such a damper is its high construction cost due to its complexity. Furthermore, MR dampers are irreparable. They are run-to-die equipment with no possibility of performing regular maintenance.

Semi-active damping systems are designed to provide adequate damping for constantly changing riding parameters. Therefore, an appropriate control strategy must be applied to the system. Efforts have been made to investigate the most suitable strategy for the application. In 1974, Karnopp et al. [6] first theoretically introduced the skyhook control technique. The skyhook principle attaches the vehicle to an imaginary reference and switches the damper between two extreme cases (very soft and very stiff) to keep the vehicle at a constant vertical level. However, the physical achievement of such a principle was not feasible in the meantime. Later in 2000, the skyhook, ground hook and hybrid methods were

experimentally tested using a quarter car rig with an MR damper [7]. Further, classic control methods (linear quadratic, linear parameter varying and sliding mode controllers) have been investigated throughout successive years [8-11]. However, due to the high nonlinearity of the oil's state transformation from liquid to solid to semi solid [12], these conventional methods of control were still inadequate for dealing with the MR damper.

In contrast to conventional computing, soft computing works with approximate models and provides solutions to complex real-world problems. Soft computation, unlike hard computing, accepts imprecision, ambiguity, partial truth, and approximations. Fuzzy logic, genetic algorithms, artificial neural networks, deep learning, and expert systems are examples of soft computing techniques. Despite the fact that soft computing theory and techniques were first applied in the 1980s, it has since grown into a significant research and analysis field in automatic control engineering [13]. Early trials to implement ANN on MR dampers were performed by Guo et al. [14] in 2004 using a single hidden layer forward neural network. The controller was trained using a quarter car rig model to simulate road excitation. At low frequency excitation, the proposed design outperformed passive systems. Lately, more complex models of ANNs have been presented with non-passive suspension systems. Hamza and Ben Yehia [15, 16] presented neural network controller for active suspension in heavy vehicles using Matlab. Their findings revealed that the model performed excellently when evaluated using the parameters specified by ISO 8608 and 2631-5.

Further to control methodologies that don't require a precise mathematical model for the system, FLC has been implemented broadly in vehicle suspension applications. Cai and Konik [17] first introduced a fuzzy logic-based controlled active suspension in 1993. The rule based was based on the sprung mass velocity and un-sprung mass velocity as the two inputs while the damping coefficient was the output of the controller. Although the membership functions used were very simple, the controlled suspension system showed better performance compared to the ordinary suspension and skyhook system as well. Later, Long [18] used more linguistic values for the membership functions and more rules in the rule based to enhance the performance of the controller. His trials enhanced damping the vibration and reduced the acceleration RMS value. Tudón-Martínez et al. [19] used FLC to set the gain schedules of the linear parameter varying (LPV) controller that controlled an MR damper. It was found that the performance was superior to that of the regulated system based solely on LPV. Ghandi et al. [20] compared the performance of four types of controllers implemented on a half car active suspension model. In terms of RMS and settling time, the FLC suspension system outperformed the passive, LQR and PID suspension systems. Nicolas et al. [21] investigated the possibility of performing the control action based on linguistic rules that are not related to the system dynamics but related to the driver's behavior. They compared the performance of two fuzzy logic controllers. The first controller was designed to act based on the behavior of the driver and the other based on the dynamics of the vehicle's acceleration. The controller that acted based on the driver's behavior showed better performance over the other controller.

In addition, attempts have been made to integrate FLC with other control methodologies for use on MR semi-active suspension. In 2000, Campos et al. [22] merged the backstepping control with FLC to control active vehicle suspension. They started with the PID controller in a stable condition and then used backstepped with FLC to control the rest of the vehicle's nonlinear unstable conditions. As compared to passive suspension systems, their approach can yield better results. Similarly, in 2004, Yang et al [23] used a composite PID and FLC controller to enhance the performance of the maglev train. It was found that vehicle displacement could be reduced by 40% using the composite controller. In a similar study, FLC was combined with a PID controller to improve the controller's performance and broaden its operating range [24]. The FLC had three inputs while outputs were the PID controller's three gains. The proposed apparatus showed a consistent performance over a wider range of road excitations. X. Jiang et al. [25] proposed a new system design to boost semi-active suspension performance to that of active suspension while avoiding the disadvantages of the latter system. FLC was used in conjunction with a magnetic actuator mounted in parallel with the MR damper and spring coil. The magnetic actuator was used to generate a force that aids in the adjustment of the system's stiffness. The configuration achieved excellent results in reducing the RMS of vibration in various road profiles, including a random profile.

Based on the foregoing, there hasn't been much focus in the literature on cost savings for semi-active suspension systems. Therefore, as a first phase of this research, the authors [26] suggested a new low-cost design of semi-active dampers by replacing the high cost and irreparable MR damper with a basic hydraulic cylinder and proportional throttle valve mounted outside the cylinder. The new design has a low cost compared with the cost of MR dampers. Since it can be one-quarter the price of an MR damper and can be scaled to be installed in a wider variety of vehicles, including mass transportation vehicles. In the proposed design, the damping coefficient can be modified by controlling the opening of the valve based on the ANN controller. The proposed design is simple to build relative to the currently available MR shock absorber. It also showed good potential compared with other commercially available shock absorbers. However, the ANN controller had limited ability since it was trained using a PID controller which has limited capabilities for such non-linear systems with many uncertainties. Therefore, the objective of current research is to explore the effects of implementing FLC on the suggested new low cost proposed suspension design. FLC doesn't require a precise mathematical model and can be adjusted regardless of the continuous changes in the system's parameters. The acceleration-driven damping (ADD) control strategy is considered where accelerations of sprung and un-sprung masses are the inputs and the opening region of the valve is the output of the FLC. The physical model of the system has been constructed using SIMSCAPE/MATLAB. The findings are compared to those of the same configuration based on the ANN controller and the semi-active suspension system with the MR damper. The findings showed improved performance of the FLC system compared to the passive suspension system and other controllers as well.

## MATHEMATICAL MODELS FOR PASSIVE AND NEW LOW-COST SUSPENSION SYSTEMS

### The Ordinary Passive Suspension Mathematical Model

The model of the quarter car illustrated in Figure 1 is widely used in literature to define the dynamics of the ordinary passive suspension system since it offers reasonable precision. Equations (1) and (2) describe the governing equations of the model.

$$m_w \ddot{Z}_w = -C_o (\dot{Z}_w - \dot{Z}_s) - k_2 (Z_w - Z_s) - k_1 (Z_w - q) \tag{1}$$

$$m_s \ddot{Z}_s = -C_o (\dot{Z}_s - \dot{Z}_w) - k_2 (Z_s - Z_w) \tag{2}$$

Where  $Z_w$  denotes the un-sprung mass displacement,  $Z_s$  denotes the sprung mass displacement and  $q$  denotes the road excitation. The values of the system parameters used in this model are shown in Table 1.

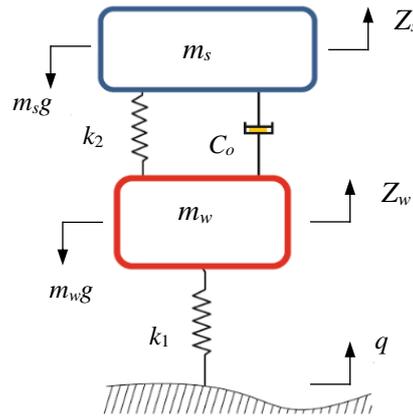


Figure 1. Quarter car model

Table 1. Parameter values for traditional passive suspension configuration

Parameter	Symbol	Value
Sprung mass	$m_s$	350 kg
Mass of the wheel	$m_w$	15 kg
Stiffness of the tire	$k_1$	200,000 N/m
Stiffness of the spring	$k_2$	16,000 N/m
Damping coefficient	$C_o$	800 N.s/m

### The Suggested New Low-Cost Suspension System Mathematical Model

The new concept proposed in this research is to position the throttle valve directly outside the conventional hydraulic cylinder and between its two input ports, as seen in Figure 2. The throttle valve is used to choke the oil flow and thus to adjust the damping coefficient of the intended shock absorber. It is noteworthy that the cost of the new proposed shock absorber is about 450 USD [27, 28] which is perceived to have a low cost relative to the MR damper available on the market. It can be said that the system proposed can be easily built and is easy to maintain.

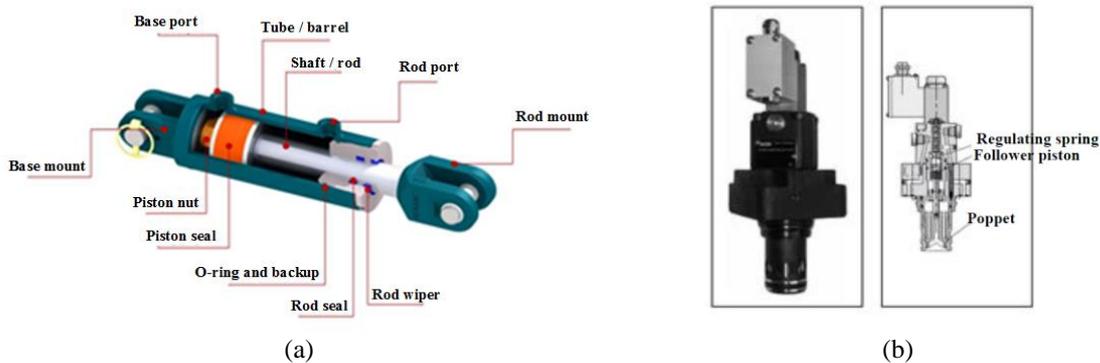
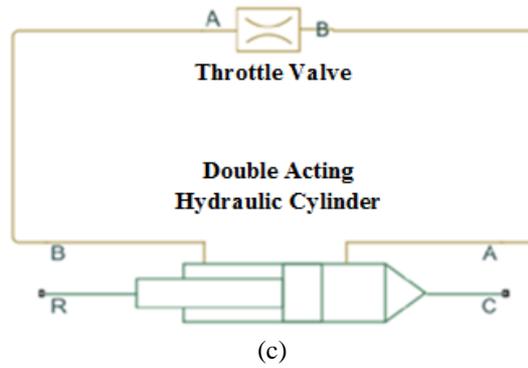


Figure 2. Proposed shock absorber configuration: (a) ordinary hydraulic cylinder, (b) proportional throttle valve



**Figure 2.** Proposed shock absorber configuration: (c) proposed circuit (cont.)

Referring to the first phase of this research by the authors [26], the mathematical equations of the proposed shock absorber device are as follows:

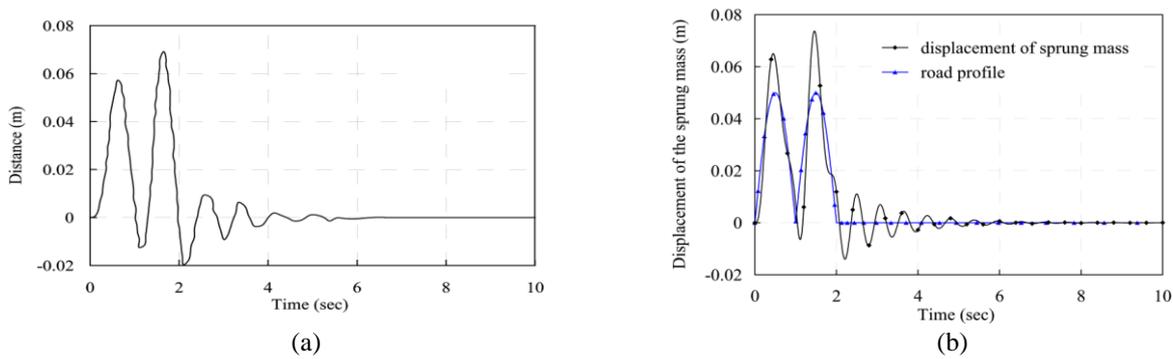
$$m_w \ddot{Z}_w = -\frac{A_p^3}{A_1^2} \times (\dot{Z}_w - \dot{Z}_s)^2 \times \left( \frac{1}{nC_c} - 1 \right) \times \rho - k_2(Z_w - Z_s) - k_1(Z_w - q) \tag{3}$$

$$m_s \ddot{Z}_s = -\frac{A_p^3}{A_1^2} \times (\dot{Z}_w - \dot{Z}_s)^2 \times \left( \frac{1}{nC_c} - 1 \right) \times \rho - k_2(Z_s - Z_w) \tag{4}$$

Where  $A_1$  is the valve nominal area,  $A_p$  is the piston area of the hydraulic cylinder,  $C_c$  is the Vena-Contracta area ratio,  $n = (A_1/A_z)$  and  $A_z$  is the orifice area. The full description of all symbols used in this paper is given in [26]. In addition, the data sheets for the components of the proposed damper can be found in [27, 28].

**Validation of the Proposed New Model**

To validate the proposed system, Eqs. (3) and (4) were modeled in Matlab/Simulink against the same road profile and with the same vehicle parameters as described in [29]. It is worth noting that the response of the proposed system is obtained when the throttle valve is completely opened (i.e. without control). The displacement of the sprung mass obtained from the suggested configuration is compared to the displacement obtained from the full car model with eight degrees of freedom ordinary suspension system described in [29] as shown in Figure 3. The figure clearly shows that the proposed model's response conforms to the verified model in [29].



**Figure 3.** Verification of the proposed passive system response: (a) ordinary passive system response [29] and (b) response of the suggested configuration

The simulation results of the mathematical model of the proposed shock absorber in the different opening areas of the throttle valve compared to the results of the traditional passive suspension are described in [26]. It is evident from the results that adjusting the opening region of the throttle valve has a significant impact on the performance of the suspension system.

The following section will therefore discuss the implementation of a fuzzy logic technique to control the opening region of the throttle valve in order to boost the efficiency of the proposed suspension system.

## FUZZY LOGIC CONTROLLER DESIGN

As can be seen from the results presented in [26], the adjustment in the valve opening would change the damping coefficient and thereby increase the efficiency of the suspension system. Therefore, in this paper, a fuzzy logic controller will be configured to adjust the opening of the valve in order to attain satisfactory ride comfort. FLC has a range of benefits. For example, it is inexpensive, simple to implement, and applicable to non-linear dynamic systems. Therefore, it has been used in many applications [30].

There are two types of suspension control strategies; one is position-driven damping (PDD) and the other is acceleration-driven damping (ADD). A PDD based controller processes the position signals of the two masses to determine the output signal. On the other hand, the ADD-based controller processes the acceleration of sprung and unsprung masses to determine the output signal. In the automotive industry, it is much easier to implement and utilize acceleration sensors than to use position sensors. Moreover, position sensing is a poor describing tool to be used in control processes. The reason for such poor performance of PDD is that position neither indicates the direction of motion nor the speed of sensed objects. Besides, it needs further computational and comparative processes on the history of the sensors' readings. Furthermore, because semi-active suspension systems need a response time of milliseconds, the PDD-based controller will often be a step behind the real time values of the device parameters. On the other hand, ADD uses a simplified measuring device with indications for the direction of motion and the predicted speed of the objects without further signal processing [31].

Based on the foregoing, a multiple input single output (MISO) type FLC based on ADD is presented in this research.

The two inputs are the acceleration signals  $\ddot{Z}_s$  and  $\ddot{Z}_w$  and the output signal shall be the valve opening ratio ( $n$ ) as described in the following:

Input 1:  $\ddot{Z}_s$   
 Input 2:  $\ddot{Z}_w$   
 Output:  $n$

Figure 4 displays the block diagram of the FLC presented. Simscape/Matlab model was developed for the passive suspension system described by Eqs. (1) and (2) to figure out the limit values of the  $\ddot{Z}_s$  and  $\ddot{Z}_w$  required to build up the membership functions of the input signals as illustrated in Figure 5.

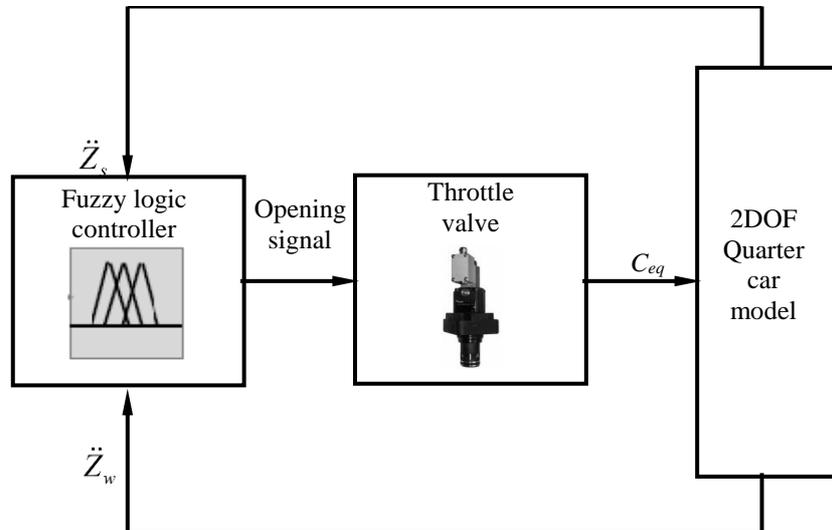


Figure 4. Block diagram of the FLC

The FLC architecture consists of three stages. In the first stage, namely, the fuzzification, the controller makes the measured input values compatible with a set of linguistic values characterized by membership functions. The linguistic terms are shown in Table 2. In order to provide more rule coverage, five fuzzy sets are used for the two inputs and four sets for the output in this work. The proposed input and output fuzzy sets are defined as follows:

$$\{\ddot{Z}_s, \ddot{Z}_w\} = [NB, NS, Z, PS, PB] \quad (5)$$

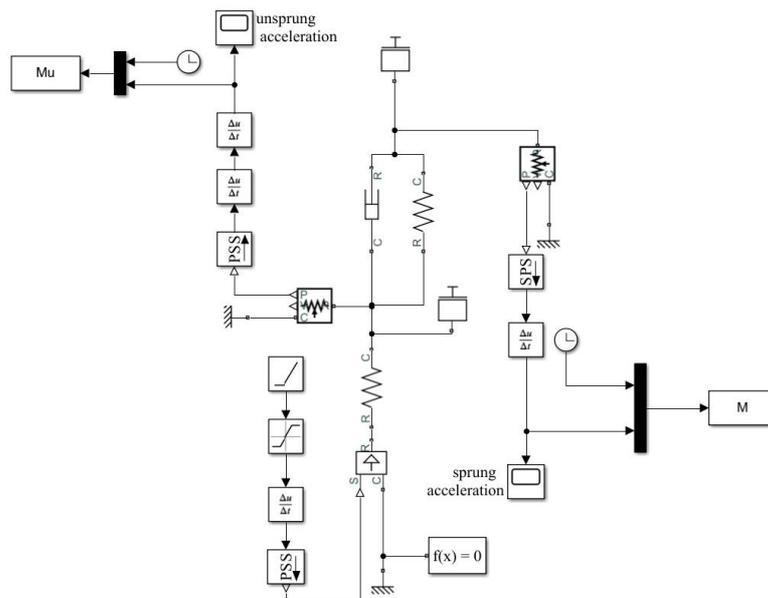
$$\{n\} = [Z, S, M, B] \quad (6)$$

The second stage is the fuzzy decision process in which the controller processes a defined list of rule based on the inputs to generate the output. Table 3 shows the rule based for the suggested FLC presented in this study. The rule based mentioned in Table 3 is derived from the skyhook principle [6] shown in Figure 6 as well as the extensive simulations performed in this work. The rule based follows the criteria below:

- If  $(\ddot{Z}_w > 0 \text{ and } \ddot{Z}_s > 0)$  or  $(\ddot{Z}_w < 0 \text{ and } \ddot{Z}_s < 0)$ , in this case, the two masses are moving in the same direction, then the damping coefficient should be maximum to dissipate all the energy that is generated by the road excitation before affecting the sprung mass.
- If  $(\ddot{Z}_w > 0 \text{ and } \ddot{Z}_s < 0)$  or  $(\ddot{Z}_w < 0 \text{ and } \ddot{Z}_s > 0)$ , in this case, the two masses are moving in opposite directions, then the damping coefficient should be near zero to allow the un-sprung mass to return to its place without affecting the sprung mass.

**Table 2.** FLC linguistic terms

Abberviation	Linguistic term
<i>NB</i>	Negative big
<i>NS</i>	Negative small
<i>Z</i>	Zero
<i>PS</i>	Positive small
<i>PB</i>	Positive big
<i>S</i>	Small
<i>M</i>	Medium
<i>B</i>	Big



**Figure 5.** Simscape/Matlab passive model for acceleration tracking

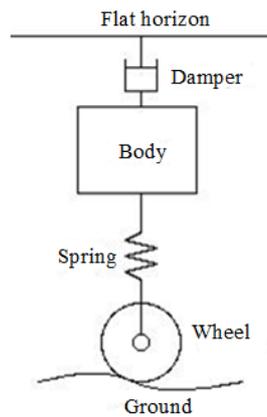
In the third stage, defuzzification and output scaling, the controller maps and scales the output from the fuzzy decision process to get the value of the output, which is the opening of the valve. It is worth noting that in the physical model, the system actuator is the valve stem which is lifted by the solenoid coil to set the valve opening. The amount of voltage applied to the valve's solenoid coil determines the stem lift. Consequently, the controller's output is the voltage applied to the valve's solenoid coil. However, for convenience, the output will be referred to as the position of the valve stem.

Limits of the membership functions representing input variables are obtained through simulations of the passive suspension system against step input road profile at different speeds. The findings are shown in Figures 7 and 8. Figure 7 displays acceleration profiles of the sprung mass at speeds of 10, 20, 30 and 40 km/hr while Figure 8 displays acceleration of un-sprung mass at the same mentioned speeds. The results showed that the acceleration of the sprung and un-sprung masses alternates between the average limits of  $-5$  and  $5 \text{ m/sec}^2$ . According to the data sheet of the throttle valve used in this work, the maximum displacement of the valve is  $1/8$  inch. As a result, the membership functions reflecting the output value should have limits of 0 inch at 0% valve opening and 0.125 inch at 100% valve opening. However, setting the minimum opening limit of the valve at 0 inch means that the valve is fully closed, resulting in a full

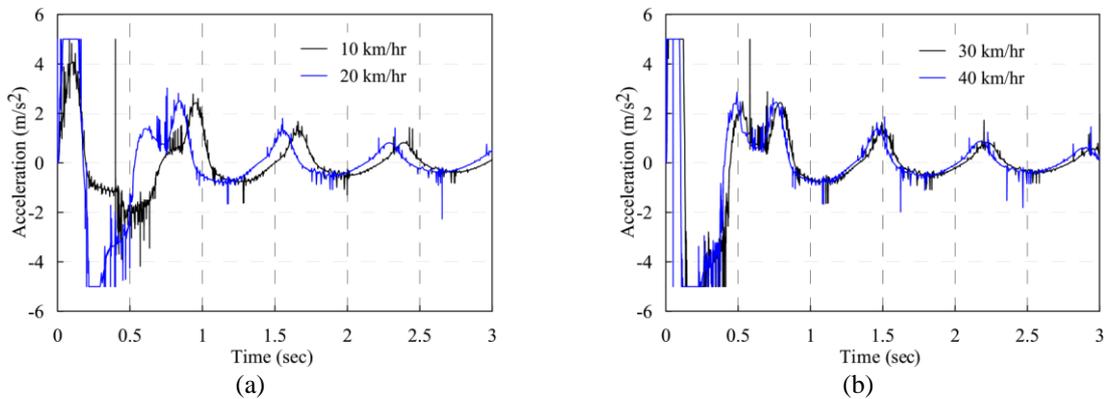
stop of the piston and limitless stiffness for the suspension. For this reason, the minimum limit has been adopted to be 0.05 inch. The membership functions of the inputs and outputs are seen in Figure 9.

**Table 3.** FLC rule base

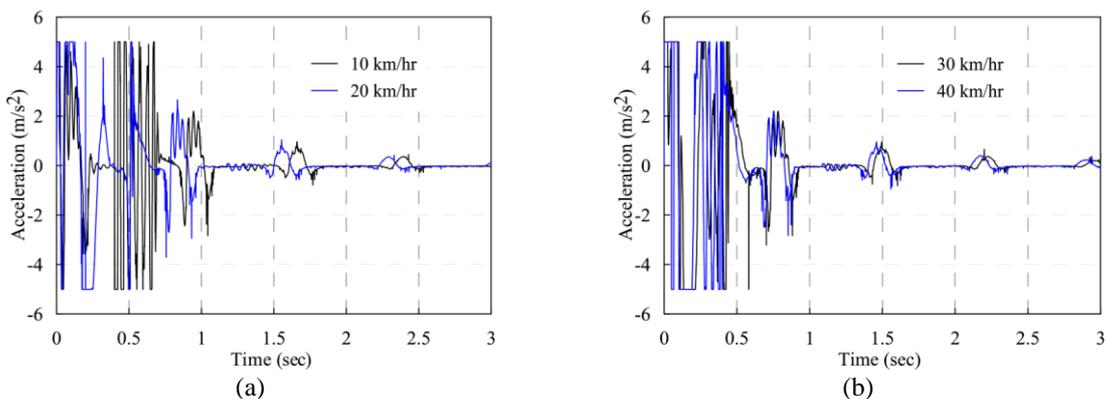
$\ddot{Z}_w \backslash \ddot{Z}_s$	<i>NB</i>	<i>NS</i>	<i>Z</i>	<i>PS</i>	<i>PB</i>
<i>NB</i>	<i>B</i>	<i>B</i>	<i>S</i>	<i>S</i>	<i>Z</i>
<i>NS</i>	<i>M</i>	<i>S</i>	<i>S</i>	<i>Z</i>	<i>Z</i>
<i>Z</i>	<i>L</i>	<i>Z</i>	<i>Z</i>	<i>Z</i>	<i>S</i>
<i>PS</i>	<i>Z</i>	<i>Z</i>	<i>Z</i>	<i>S</i>	<i>M</i>
<i>PB</i>	<i>Z</i>	<i>S</i>	<i>S</i>	<i>B</i>	<i>B</i>



**Figure 6.** Skyhook principle



**Figure 7.** Acceleration time history of sprung mass: (a) speeds 10 and 20 km/hr and (b) speeds of 30 and 40 km/hr



**Figure 8.** Acceleration time history of un-sprung mass: (a) speeds 10 and 20 km/hr and (b) speeds 30 and 40 km/hr

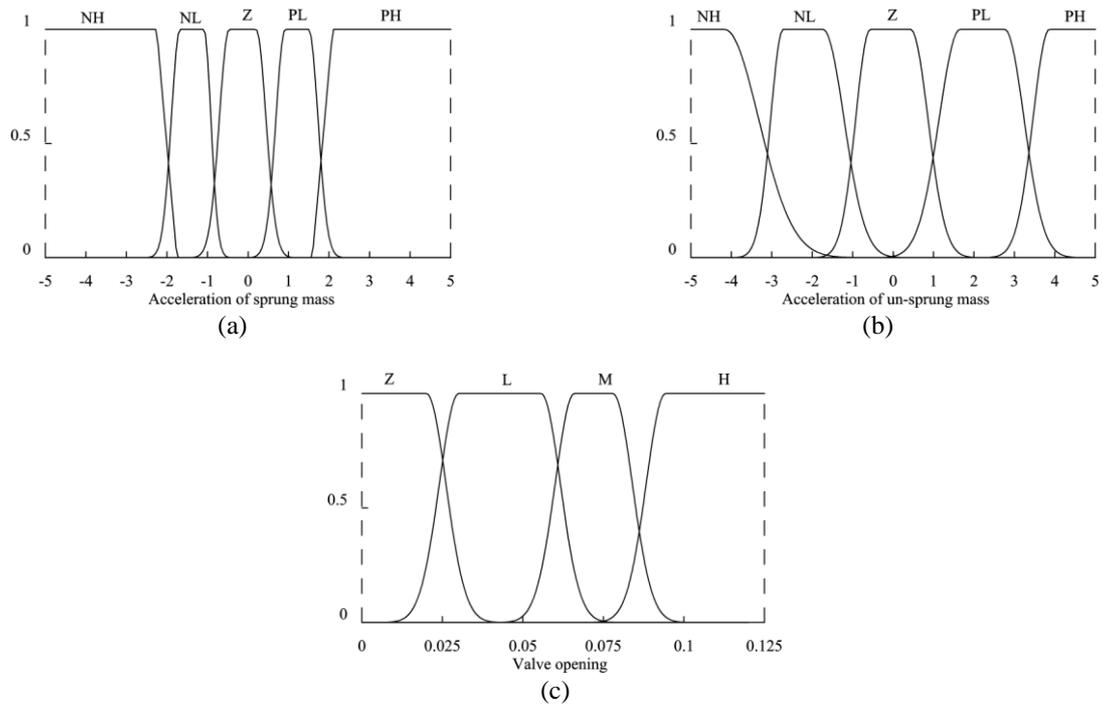


Figure 9. Membership functions: (a) the input 1, (b) the input 2 and (c) the output (valve opening)

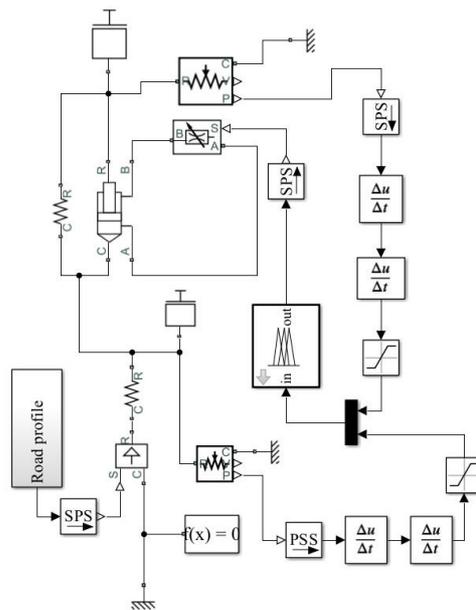
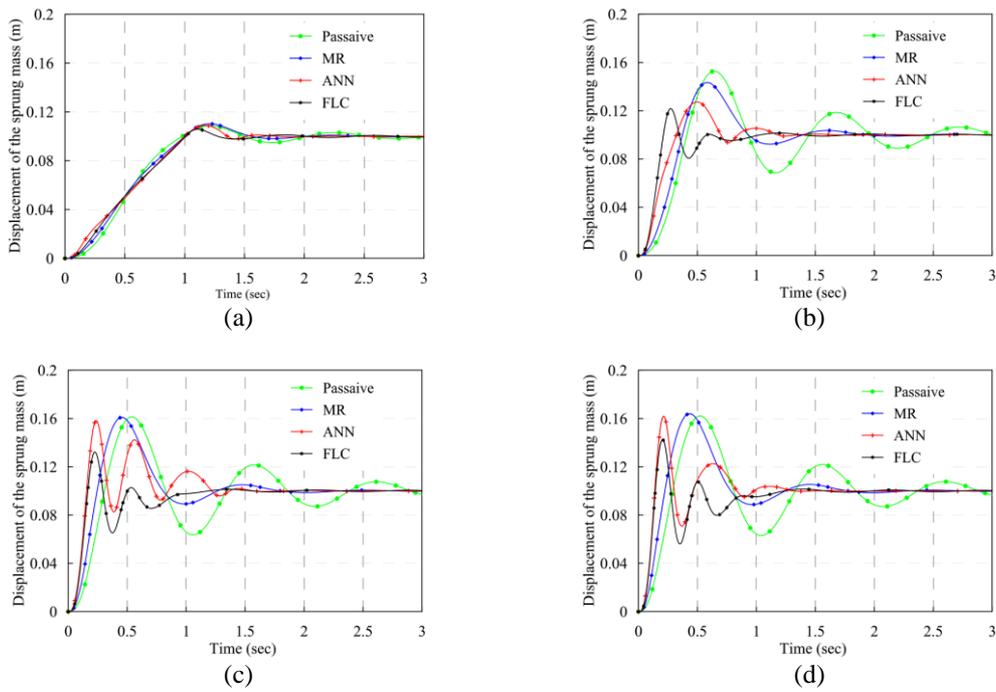


Figure 10. Simscap/Matlab simulation model of the suggested new shock absorber with FLC

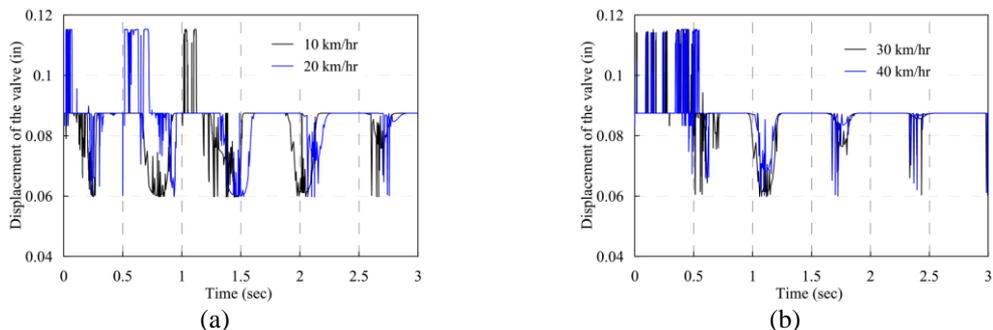
## RESULTS AND DISCUSSION

A Simscape/Matlab model was developed to determine the effect of the application of the FLC on the efficiency of the proposed shock absorber as seen in Figure 10. MATLAB/SIMSCAPE was used instead of SIMULINK. SIMSCAPE utilizes pre-programmed blocks that simulate physical components of a system. Blocks that represent the proportional throttle valve and the hydraulic cylinder were prefaced in the graphical user interface (GUI). The connection between the valve and cylinder blocks is performed to achieve the proposed system's configuration. The response of the sprung mass is plotted to assess the enhancement of ride comfort of a vehicle equipped with the proposed shock absorber. Furthermore, the responses of the MR suspension system as well as the proposed suspension system controlled by ANN are provided for the sake of comparison. On the other hand, the response of the controlled throttle valve is also monitored in order to verify the compatibility of the commercially available throttle valves on the market.



**Figure 11.** Suspension systems response with different controllers against up-step input: (a) speed of 10 km/hr, (b) speed of 20 km/hr, (c) speed of 30 km/hr and (d) speed of 40 km/hr

The findings appear in Figures 11 and 12. Figure 11 displays the time history of vertical sprung mass shifts at varying vehicle speeds with a 10 cm step excitation input. The figure is divided into four sub-figures: sub-figure (a) is the time history at a speed of 10 km/hr, sub-figure (b) at a speed of 20 km/hr, sub-figure (c) at a speed of 30 km/hr, and sub-figure (d) at a speed of 40 km/hr. Each sub-figure has four plots; the green plot is the time history of the vertical displacement with the ordinary suspension system, the blue plot is in the case of MR damper suspension configuration, the red plot is with the proposed damper system managed by ANN control and finally, the black plot is the displacement with the suggested system regulated by FLC. The figure shows that the four suspension configurations at low speeds (10 km/hr) have almost the same performance characteristics in relation to frequency of vibration and settling time. However, relative to the three other suspension systems, the FLC has a 50 percent lower overshoot. As the speed increases to higher values (20, 30 and 40) km/hr, the superiority of the FLC prevails, as the performance of the suspension system could achieve remarkable enhancements regarding the ride comfort. In terms of overshoot, at speed of 20 km/hr, the proposed damper with FLC reduces the overshoot by 60% in comparison with the ordinary damper, 50% in comparison with the MR damper and 20% in comparison with the suggested damper regulated by the ANN controller. Compared to the other three systems, the FLC decreases the overshoot by 50% at 30 km/hr. On the other hand, at 40 km/hr, the reduction is 30%. In terms of settling time, the suggested suspension system with FLC exhibits the lowest value among all the presented systems. In order to achieve the reliable outcomes of a control strategy, the action devices should be compatible with the controller in terms of action time. Figure 12 shows the valve displacement time history of at four different vehicle speeds. As illustrated in the figure, the frequency and amplitude of the stem increase with the increase of vehicle speed. At low speeds, the system does not require high damping. As a result, the displacement of the valve stem is less as compared to greater speeds. In addition, due to the low speed, the oil is pumped through the valve at a lower flow rate. Hence, the valve stem moves towards the closing direction so that it can provide proper damping. On the other hand, as speed increases, the valve tends to move toward the opening direction to allow the higher flow rates of the oil to pass through the valve. Furthermore, it can be shown from the figure that the displacement of the valve indicates a mean frequency of 40 Hz (25 millisecond response time) which is compatible with the commercially available throttle valves on the market [28].



**Figure 12.** The valve displacement time history against step input at vehicle speed of (a) 10 and 20 km/hr and (b) 30 and 40 km/hr

## CONCLUSION

This paper proposes the implementation of FLC on a new low cost shock absorber to enhance vehicle ride comfort. The proposed shock absorber has proved to have good potential with vibration damping. The controller modifies the damping coefficient of the vehicle shock absorber by regulating the opening region of the throttle valve. The FLC is designed on the basis of an acceleration-driven damping strategy, where accelerations of sprung and un-sprung masses are the inputs and the valve opening area is the output of the controller.

Simulations are conducted at various vehicle speeds to assess the feasibility of the proposed new shock absorber regulated by FLC. The simulation model is produced in a Simscape/Matlab environment on the basis of a quarter vehicle model. The performance of the suggested shock absorber using the ANN controller and the performance of the suspension system with the MR damper were presented for comparative purposes.

It can be concluded that the suggested suspension system is simple to maintain in comparison to current commercially available MR dampers and can be scaled to be installed in a wide range of vehicles, including mass transportation vehicles. Furthermore, the proposed suspension system is simply built in comparison to commercially available MR shock absorbers on the market. In addition, the proposed shock absorber is one-fourth the cost of the MR shock absorber. In terms of performance, the suggested shock absorber controlled by FLC has a better performance when it is compared with the ordinary damper, the suggested shock absorber controlled by ANN controller and the MR damper. Moreover, the specifications required for the throttle valve in the proposed shock absorber are compatible with commercially available ones on the market.

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