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# Genetic Algorithm Aided Groundhook Control Strategy for Semi-Active Magneto Rheological Damper Suspension System

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**Abstract:** In this paper, a novel strategy called Genetic Algorithm aided Groundhook Control (GAGC) is proposed for vibration suppression of a 5 Degree Of Freedom (DOF) model of semi-active vehicle suspension system. This model considers bounce and pitch motions and utilizes magneto rheological damper (MRD). The model represents a pitch-plane vehicle suspension system with two independent spring-MRD sets in front section, two spring-MRD sets in rear section and a spring-MRD set as the driver's seat suspension system. The last spring-MRD set is controlled using groundhook control strategy. In this research we used Bingham plastic model to characterize the behavior of this MRD. The parameters of the proposed MRD are determined using Genetic Algorithm (GA). Simulation results prove that the proposed strategy yields a comfortable ride as well as effective vibration suppression.

**Key words:** Bingham plastic model, Semi-active vibration control, 5 DOF pitch-plane suspension model, Genetic algorithm, Groundhook controller.

## INTRODUCTION

Although the passive suspension systems provide design simplicity and cost-effectiveness, performance limitations are inevitable due to the uncontrollable damping force. At the other pole, active suspension systems provide comfortable ride (Yoshimura, T., Y. Isari, 1997). However, they requir high-power actuators and a large power supply (Yi, K. and B.S. Song, 1999). To enjoy the advantages of both passive and active suspension systems, semi-active suspension systems were proposed (Vassal, C.P., *et al*., 2008; Yao, G.Z., *et al*., 2002). In these systems, conventional springs are retained, but passive shock absorbers are replaced with controllable shock absorbers. Semi-active suspension systems are almost as effective as active suspension systems in vibration isolation and ride quality while being safe, simple and economic (Karnopp, D.C., M.J. Crosby, 1974; Karnopp, D.C., 1990; Krasnicki, E.J., 1980; Krasnicki, E.J., 1981).

Two types of control strategies can be used in a semi-active suspension system: Skyhook control and groundhook control. The latter one which is considered in this study, works as follows: When the relative velocity between driver's seat and sprung mass is in the same direction with the velocity of the sprung mass, an electric current is applied to the MRD, otherwise no damping force is required. Since a MRD is unable to provide a zero force, we should minimize the semi-active damping force without any electric current. This condition implies that the actuating of the controller only assures the increment of energy dissipation of the stable system.

The review of simple but credible models reveals that the vibration response of vehicles to different excitations can be investigated through the analysis of various in-plane models (Ahmed, A.K., 2002). In literatures, half-car model (Rao, L.V.V.G. and S. Narayanan, 2008) as well as the pitch-plane suspension model (Martynowicz, P. and B. Sapin, 2005; Sapin, B. and M. Roso, 2005) are considered for the purpose of studying bounce and pitch motions. Spinski and Rosol (Sapinski, B.and M. Rosol, 2008) concerned an autonomous control system (ACS) for a 3 DOF pitch-plane suspension model. They assumed negligible contributions due to the wheel-axle-brake assemblies' masses and tires stiffness. That means the road input is taken to be the same as the wheels input.

To develop a more precise model, we introduce a 5 DOF pitch-plane model of the system representing a vehicle suspension equipped with four spring-damper sets in front and rear sections and a driver's seat

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suspension with the fifth spring-damper set. In the proposed model, the effect of wheel-axle-brake assemblies' masses on the suspension system is considered by adding two more degrees of freedom. For further considerations, it is assumed that only the MRD of driver's seat is controlled and all other dampers are passive. Since the passenger's acceleration should be minimized due to the comfortable ride criterion, the aim of control is to minimize both vibration (tracking error) and acceleration of the driver's seat under road bumps. The parameters of the proposed MRD are selected and optimized using GA.

***Mathematical Model of Suspension System:***

The diagram of the 5 DOF pitch-plane suspension model is shown in Fig. 1. In this model, two subsystems can be distinguished: a 4 DOF system comprising a suspended beam that is model of body with two unsprung mass and a 1 DOF system comprising a mass which is the model of driver's seat. In this figure, $m_{ur}$ is the rear unsprung mass, $m_{uf}$ is the front unsprung mass, $m_s$ is the sprung mass and $m_c$ is the mass of driver's seat. In addition, $k_{tr}$ is the elasticity factor of the rear wheel-axle-brake assembly, $k_{tf}$ is the elasticity factor of the front wheel-axle-brake assembly, $k_{sr}$ is the elasticity factor of the rear spring, $k_{sf}$ is the elasticity factor of the front spring and $k_c$ is the elasticity factor of the seat spring. It should be noted that $C_{sr}$ is the damping coefficient of the rear damper, $C_{sf}$ is the damping coefficient of the front damper and $C_c$ is the damping coefficient of the seat damper. $Z_{ur}$, $Z_{uf}$, $Z_{sr}$, $Z_{sf}$ and $Z_c$ are absolute displacement of rear unsprung mass, front unsprung mass, rear edge of sprung mass, front edge of sprung mass and the seat respectively. *I* is the mass moment of inertia of the vehicle about a centroidal axis, *a* is the distance between the driver's seat and front edge of sprung mass and *b* is the distance between the driver's seat and rear edge of sprung mass. *u* which is the applied force in the driver's seat suspension will be controlled actively and all the dampers are passive.

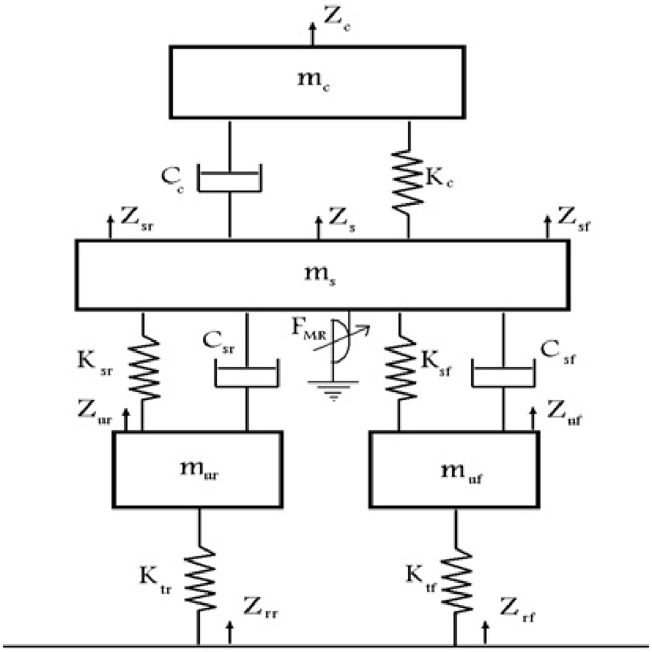


**Fig. 1:** Vehicle model

The state variables are adopted as follows:

$$x_1 = z_{uf} - z_{rf} \qquad x_2 = z_{sf} - z_{uf}$$

$$x_3 = \dot{z}_{uf} \qquad x_4 = \dot{z}_{sf}$$

$$x_5 = z_{ur} - z_{rr} \qquad x_6 = z_{sr} - z_{ur}$$

$$x_7 = \dot{z}_{ur} \qquad x_8 = \dot{z}_{sr}$$

$$x_9 = z_c - z_s \qquad x_{10} = \dot{z}_c \tag{1}$$

The state space equation is shown in relation 2:

$$\dot{X} = AX + Bu + L\dot{w} \tag{2}$$

$$Y = CX$$

Where $w$ is the road profile, $u$ is control force applied by groundhook force controller and matrixes A, B, L and C, are as follows:

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ A_{31} & A_{32} & A_{33} & A_{34} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & A_{42} & A_{43} & A_{44} & 0 & A_{46} & A_{47} & A_{48} & A_{49} & A_{4\,10} \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & -1 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & A_{75} & A_{76} & A_{77} & A_{78} & 0 & 0 \\ 0 & A_{82} & A_{83} & A_{84} & 0 & A_{86} & A_{87} & A_{88} & A_{89} & A_{8\,10} \\ 0 & 0 & 0 & A_{94} & 0 & 0 & 0 & A_{98} & 0 & 1 \\ 0 & 0 & 0 & A_{10\,4} & 0 & 0 & 0 & A_{10\,8} & A_{10\,9} & A_{10\,10} \end{bmatrix} \quad B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{1}{m_c} \end{bmatrix} \quad L = \begin{bmatrix} -1 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad C = \begin{bmatrix} 1 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}^T \tag{4}$$

and unknown elements are defined below in Equation (5):

$$A_{44} = -\frac{c_{sf}}{m_s} - \frac{b \times c_c}{m_s(a+b)} - \frac{a^2 c_{sf}}{I} \qquad A_{86} = -\frac{k_{sr}}{m_s} - \frac{b^2 k_{sr}}{I} \qquad A_{49} = \frac{k_c}{m_c}$$

$$A_{48} = \frac{-c_{sr}}{m_s} - \frac{a c_c}{m_s(a+b)} + \frac{abc_{sr}}{I} \qquad A_{87} = \frac{c_{sr}}{m_s} + \frac{b^2 c_{sr}}{I} \qquad A_{5.10} = \frac{c_c}{m_c}$$

$$A_{84} = -\frac{c_{sf}}{m_s} - \frac{b c_c}{m_s(a+b)} + \frac{abc_{sf}}{I} \qquad A_{94} = -\frac{b}{a+b} \qquad A_{75} = -\frac{k_{tr}}{m_{ur}}$$

$$A_{88} = -\frac{c_{sr}}{m_s} + \frac{ac_c}{m_s(a+b)} - \frac{b^2 c_{sr}}{I} \qquad A_{10.4} = \frac{bc_c}{m_c(a+b)} \qquad A_{76} = -\frac{k_{sr}}{m_{ur}}$$

$$A_{42} = -\frac{k_{sf}}{m_s} - \frac{a^2 k_{sf}}{I} \qquad A_{98} = -\frac{a}{a+b} \qquad A_{77} = -\frac{c_{sr}}{m_{ur}}$$

$$A_{43} = \frac{c_{sf}}{m_s} + \frac{a^2 c_{sf}}{I} \qquad A_{10.8} = \frac{ac_c}{m_c(a+b)} \qquad A_{78} = \frac{c_{sr}}{m_{ur}}$$

$$A_{46} = \frac{-k_{sr}}{m_s} + \frac{abk_{sr}}{I} \qquad A_{31} = -\frac{k_{tf}}{m_{uf}} \qquad A_{89} = \frac{k_c}{m_c}$$

$$A_{47} = \frac{c_{sr}}{m_s} - \frac{abc_{sr}}{I} \qquad A_{32} = \frac{k_{sf}}{m_{uf}} \qquad A_{8.10} = \frac{C_c}{m_c}$$

$$A_{82} = -\frac{k_{sf}}{m_s} + \frac{abk_{sf}}{I} \qquad A_{33} = -\frac{c_{sf}}{m_{uf}} \qquad A_{10.9} = -\frac{k_c}{m_c}$$

$$A_{83} = \frac{c_{sf}}{m_s} - \frac{abc_{sf}}{I} \qquad A_{34} = \frac{c_{sf}}{m_{uf}} \qquad A_{10.10} = -\frac{C_c}{m_c} \tag{5}$$

***Mathematical Model of MRD:***

MR devices can be divided into three groups of operational modes, valve mode, direct shear mode, and squeeze mode. In the valve mode, one of two surfaces that are in contact with the MR fluid moves relative to the fluid. This relative motion creates a shear stress in the fluid. The shear strength of the fluid may be varied by applying different levels of magnetic field. In the direct shear mode, the fluid is pressurized to flow between two surfaces which are stationary. The flow rate and the pressure of the fluid may be adjusted by varying the magnetic field. In the squeeze film mode, two parallel surfaces squeeze the fluid and the motion of the fluid is perpendicular to that of the surfaces. The applied magnetic field determines the force needed to squeeze the fluid and also the speed of the parallel surfaces during the squeezing motion (Yalcintas, M., 1999).

Several different designs of MRDs have been issued. One of them is the bypass damper where the bypass flow occurs outside the cylinder and an electromagnet applies a magnetic field to the bypass duct (Sodeyama, H., *et al*., 2003; Sodeyama, H., *et al*., 2000). In another design the electromagnet is inside the cylinder and the MR fluid passes through an annular gap between the electromagnet and the inner cylinder. This method uses an accumulator to make up for the volume of fluid displaced by the piston rod which is going into the damper (Snyder, R.A., G.M. Kamath, 2000; Snyder, R.A. and N.M. Wereley, 1999) Other MRD fluid models in the literatures include the Herschel-Bulkley model which takes into account the postyield shear thinning and thickening behavior and the Bouc-Wen model where the parameters of the model can be adjusted to control the linearity in the unloading and the smoothness of the transition from the pre-yield to the post-yield region (Snyder, R.A. and N.M. Wereley, 1999).

Several models have been suggested to characterize the behavior of MRDs. The most common one is the Bingham plastic model. This model is an extension of the Newtonian flow and it is obtained by considering the yield stress of the fluid. It assumes that flow will occur when the dynamic yield stress is reached. The total stress is given by

$$\tau = \tau_y \operatorname{sgn}(\dot{\gamma}) + \eta\dot{\gamma} \tag{6}$$

Where $\tau_y$ is the yield stress induced by the magnetic field, $\dot{\gamma}$ is the shear rate and $\eta$ is the viscosity of the fluid. In this model, the relationship between the damper force and the shear velocity may be given as

$$F = \begin{Bmatrix} F_y \operatorname{sgn}(\dot{x}) + C_0\dot{x}, \dot{x} \neq 0 \\ -F_y < F < F_y, \dot{x} = 0 \end{Bmatrix} \quad (7)$$

Where $C_0$ is the post-yield damping coefficient and $F_y$ is the yield force. In the post-yield part, the slope of the force-velocity curve is equal to the damping coefficient which is essentially the viscosity of the fluid, $\eta$. Both $C_0$ and $F_y$ are functions of the control current input, i and can be modeled as second order polynomial functions:

$$\begin{aligned} F_y(i) &= F_{yc}i^2 + F_{yb}i + F_{ya} \\ C_0(i) &= C_c i^2 + C_b i + C_a \end{aligned} \quad (8)$$

The proposed MRD model has 6 parameters: $F_{yc}, F_{yb}, F_{ya}, C_c, C_b, C_a$ These parameters are not the same for different systems and should be tuned. For the proposed model, GA method is used to optimize these parameters.

***Semi-Active Control Strategy:***

The semi-active damper can only dissipate energy (Ahmed, A.K., 2002). In the semi-active implementation of the groundhook damper control an input current to the damper is only applied when the relative velocity between the two bodies and the absolute velocity of the sprung mass are in the same direction. Otherwise, the damping coefficient is set to its minimum value which is when the input current is equal to zero. It means that it is called groundhook damper control because the active control force is proportional to the sprung mass absolute velocity as if the sprung mass were connected to an inertial frame. The control policy therefore is

$$F = \begin{Bmatrix} F_{\max} when(\dot{z}_c - \dot{z}_s)\dot{z}_s \geq 0 \\ F_{\min} when(\dot{z}_c - \dot{z}_s)\dot{z}_s < 0 \end{Bmatrix} \quad (9)$$

Or equivalently:

$$F = \begin{Bmatrix} F_{\max} when(\dot{x}_9)(x_{10} - \dot{x}_9) \geq 0 \\ F_{\min} when(\dot{x}_9)(x_{10} - \dot{x}_9) < 0 \end{Bmatrix} \quad (10)$$

The system has the Bingham plastic model MR damper and an on/off control algorithm. In on-off groundhook control, the damper is controlled by two damping values illustrated in Fig. 2, these are referred to as high-state and low-state damping.

Determination of whether the damper is to be adjusted to either its high state or its low state depends on the product of the relative velocity across the suspension damper and the absolute velocity of the sprung mass attached to that damper, as illustrated in Fig. 3.

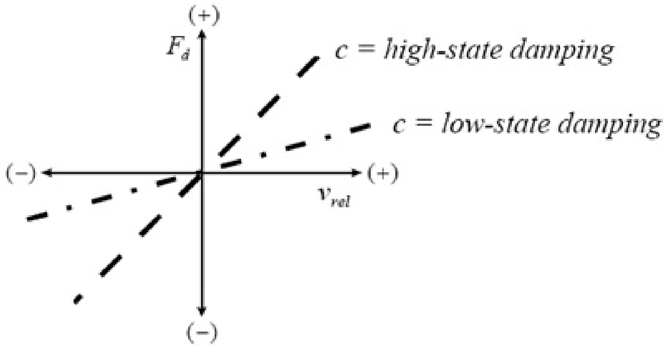


**Fig. 2:** Semi-active damping: on-off control

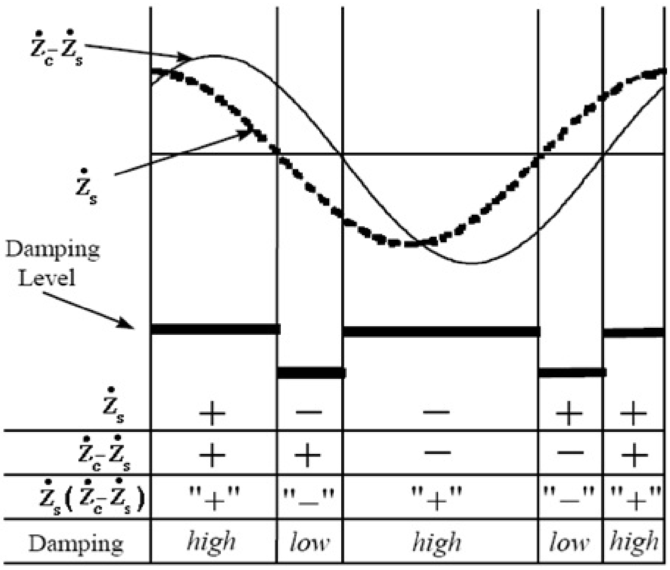


**Fig. 3:** On-off groundhook control illustration

If the product is positive or zero, the damper is adjusted to its high state; otherwise, the damper is set to the low state. The logic of the on-off groundhook control policy is as follows: When the relative velocity of the damper is positive, the force of the damper acts to pull up on the sprung mass; when the relative velocity is negative, the force of the damper pushes down on the sprung mass. However, when the absolute velocity of the sprung mass is negative, it is traveling downwards and the maximum (high state) value of damping is desired to pull the mass, while the minimum (low state) value of damping is desired to continue pushing down on the mass. But, if the absolute velocity of the sprung mass is positive and the mass is traveling upwards, the maximum (high state) value of damping is desired to push down the mass, while the minimum (low state) value of damping is desired to further pull the mass upwards. The on-off groundhook semi-active policy emulates the ideal sprung mass displacement control configuration of a passive damper "hooked" between the sprung mass and the "ground," as shown in Fig. 1. In this algorithm, the MR damper is either turned on by applying a 1.5A input current or it is turned off by setting the input current to 0A. No intermediate current is applied. In the simulations, the MR damper is assumed to respond very fast and the time delay associated with the damper's response time is ignored.

***Genetic Algorithm:***

As mentioned, GA is used to determine unknown parameters of the proposed MRD model- $F_{yc}, F_{yb}, F_{ya}, C_c, C_b, C_a$ . These parameters are not the same for different systems and should be tuned. The desired performance which should be satisfied by GA is minimizing the tracking error of the driver's seat and its acceleration. That means the fitness function is defined as:

$$Fitness = norm(\frac{d^2(e(t))}{dt^2}) + 50norm(e(t)) \tag{13}$$

Where e(t) (error) is the difference between the actual position and the desired position (zero) of the seat. GA was run in the following steps.

1. Set the initial parameters for GA: population size, crossover type and probability, and mutation probability.
2. Generate the initial population randomly.
3. Reckoning of a fitness value for each subject.
4. Selection of the subjects that will mate according to their share in the population global fitness.
5. Apply the genetic operators (crossover, mutation...).
6. Repeat Steps 3 to 6 until the generation number is reached.

***Simulation Results:***

To evaluate the effectivness of the GAGC, a simulation has been performed and two important parameters in the system were compared. These parameters are the absolute displacement (Fig. 4) and absolute acceleration (Fig. 5) of the driver's seat. In addition, the applied force by the groundhook damper is presented in Fig. 6. The step function is applied as road profile. As seen in these figures, groundhook control strategy can improve the system response. Results revealed adequacy and good performance of MRD in the driver's seat suspension and groundhook control strategy as it limits the amplitude of vibration acceleration transmitted onto the seat.

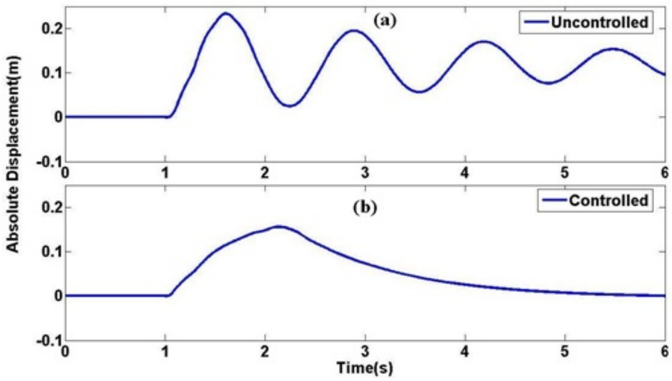


**Fig. 4:** Absolute Displacement; (a): Uncontrolled, (b): Controlled

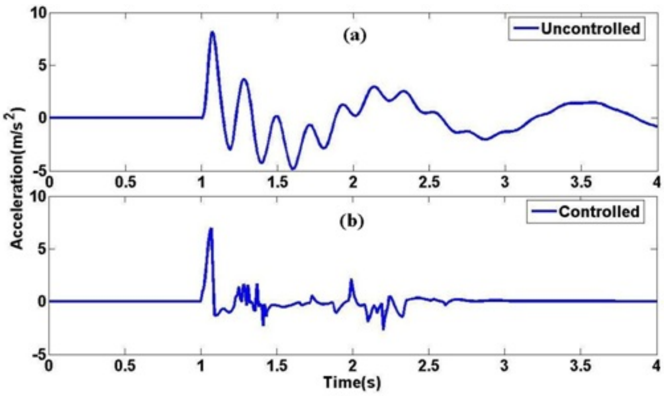


**Fig. 5:** Absolute Acceleration; (a): Uncontrolled, (b): Controlled

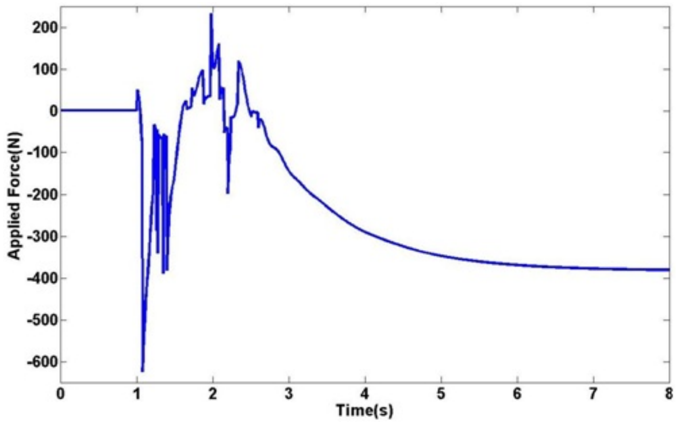


**Fig. 5:** Applied Force

***Conclusion:***

In this study, a 5 DOF pitch-plane suspension model of car was introduced and the effects of axle and tire assembly were investigated. We applied a MRD in driver's seat suspension in order to suppress its vibrations. The proposed MRD model has 6 unknown parameters. These parameters are not the same for different systems and should be tuned. For the proposed model, GA was used to determine the optimized value of these parameters. The desired performance which should be satisfied utilizing GA is minimizing the tracking error of controller and its acceleration. The simulation results demonstrate the efficacy of proposed method in satisfying confortable ride criteria.

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