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Vehicle ride comfort optimization based on Magneto-rheological damper

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Abstract

Vehicle suspension design many time's leads in a compromise between the flagging demands of riding comfort and road holding. To overcome this challenge new technologies and smart materials need to be installed in vehicle dynamic systems to improve the suspension performance. This paper describes firstly the smart material (magneto rheological fluid) which can change the viscosity due to the magnetic field applied on it, this fluid is suitable for making shock absorbers which can react on different vibrations in few time less than ten milliseconds. To control the damping force of MR damper, two controllers have been designed and built in Simulink: a fuzzy logic controller and damper controller to track the desired damping force. To perform the simulation a 7 degree of freedom for vehicle dynamic model and road model was designed in CarSim software; a co-simulation model of vehicle and controller was constructed in Simulink. Finally, the comparative simulation experiments of passive suspension and semi-active suspension with magneto rheological damper was performed. The results demonstrate that, the ride comfort has been improved more than 35 % compared to passive suspension and the road handling evaluation indicators were improved 22 %. In summary, magneto rheological damper can effectively improve vehicle ride comfort and road handling.

Keywords: Vehicle ride comfort, Bouc-Wen model, Fuzzy logic controller, Semi active suspension

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1. Introduction

In design of automotive dynamic system, there are two important systems need to be emphasized on; the comfort of vehicle occupants and vehicle stability. Suspension systems are ones of the most important dynamic subsystems in passenger and transportation vehicles. Suspension systems provide comfort to occupants and avoid the damage of chassis and freight. They are normally constituted by springs and dampers which reduce the shock and absorb the frequencies vibrations during the vehicle ride. Road cracks, clays, debris, bad pavement, bumps are the most disturbances of vehicle drive where the driver is subjected to drive over them at a reduced speed in order to mitigate vibrations of vehicle, in order to give a comfort to occupants as well as preventing the wear and damage of the car components especially the suspension system (shock absorbers), freight and chassis. The big challenges in vehicle suspension design many time's leads in a compromise between the flagging demands of riding comfort and road holding. Automobiles owning soft suspension separate the vehicle body from the higher vibrations in suspension but decrease the capacity of the dampers to monitor the wheel movements which result the poor road holding. Inversely, hard suspension generates the best road holding but transfers more suspension vibration to the body[1]. The design of a better quality suspension system remains an important development objective for the automotive industry. Effective vehicle suspension system must have the ability to minimize as possible the displacement, jounce of the vehicle body, and maximizing ride comfort. It must also aim to reduce the dynamic deflection of the tire to maintain road tire contact. The stability control of automobile is based on different subsystems which work together to achieve the same goal those subsystems are steering system, anti-lock braking system (ABS) and vehicle suspension control system. Presently intelligent vehicle comfort and stability subsystems are being raised to enhance driver to monitor the car, vehicle operators become more indirectly in control of their vehicles. To reach the intelligent control of car subsystem, fused sensors and vehicle control strategies must be put in consideration. To reduce the vibration caused by the unven road, many suspension researches are emphasizing on finding methods to optimize both issues at the same time, but problems arise because comfort and stability are opposite goals, thereby a fixed type of suspension has to be chosen, based on certain application. The smart materials like, magneto-rheological damper which change the viscosity in little time has showed the ability to mitigate the tradeoff between comfort and stability. To improve damping force and reduce response time of magneto rheological (MR) damper, Some researches like CHEN Bing[2] made the control of vehicle semi-active suspension with the use of double fuzzy controllers, and the results of simulation demonstrate that the fuzzy control mentioned above can not only improve the ride comfort, however decrease the probability of suspension broke up. To solve the problem of complex nonlinear vehicle semi-active suspension, YAN Wen-Jun[3]established a fuzzy control strategy to determine the current of MR damper. The simulation results show that the strategy can improve the riding comfortless, driving safety and operating stability remarkably. In summary the intelligent suspension is being expanded in automotive era. In this paper an independent suspension system of full vehicle model composed by magneto-rheological dampers, and IF-THEN rules of fuzzy logic controller has been developed through the joint simulation of Matlab/Simulink and CarSim has been used. The output results are suspension deflection, body velocity, vertical body displacement which are the index of vehicle stability and comfort.

2. Magneto-rheological damper

Magneto-rheological (MR) materials are smart materials which are being applied in many field like, automotive, building and industrial era. They are materials where physical properties can be modified or changed by external physical properties like as, pH, magnetic field, moisture, stress, temperature or electricity. Magneto-rheological material has various forms of fluids; it can be a gel or even a solid material like elastomers. It is composed by micron-sized iron part that is plunged in carrier oil. Magneto rheological fluids shows the ability of changing from free-moving liquid state into a semi-solid state with resistance on fluid movement in fast reaction within some milliseconds when applied to magnetic field, it means that their viscosity changes in few time which is less than ten milliseconds. Since its invention the yield stress and viscosity varies when magnetic field is subjected on it; the application of MR fluid have been studied, investigated and applied in different field like automotive clutches, engine mounts, dampers (vehicle damper or building damper) haptic devices, etc. When the magnetic field is applied to the MR fluid, the characteristics changes from the chain shape structure between paramagnetic MR particles in the low permeability solvent [5]. At the nominal condition, MR fluid shows the isotropic Newtonian characteristics since the MR iron particles move freely as shown in Fig. 1 (a). When the magnetic field is applied to the MR fluid, MR fluid makes a chain shape as it is demonstrated in Fig. 1(b), where the viscosity of the oil is increased[4]. MR fluids demonstrate the anisotropic Bingham characteristic which provoke the resistance on flow and external shear stress.

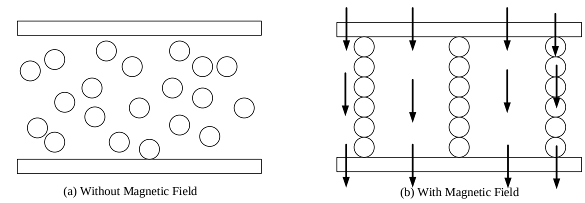

Fig. 1 Magneto-rheological fluid

2.1 Modelling Magneto-Rheological damper

In modeling and simulation of MR damper the hysteretic Bouc-Wen equation is among different models and the most used to study and simulate various method of expanding the damping system by varying the hysteresis. The first model of magneto-rheological damper has been designed by Bouc in 1971 [8] and later developed by Wen in 1976 [9] who has demonstrated various parameters of this model to show a large type of hysteretic patterns. Due to this wonderful behavior, the model was utilized to investigate various nonlinear hysteretic systems such as MR damper and hysteretic isolators. Even if, MR dampers demonstrated a good typical characteristic, the final hysteretic setting depends on some specific features related to absorber, damper geometry etc. Thus, the model must be satisfactory to have the real MR damper behavior. The Bouc-Wen model is established by three main parts: Bouc-Wen block, dashpot and spring mounted in parallel setting as it indicated on Fig. 3. As it has shown in previous paragraphs, MR damper is a type of semi-active damper where the viscosity of the MR fluid is monitored by changing the amount of the input current and hence changes the output torque. To control output torque a complex program of computation is needed to generate the required current. To present the hysteretic characteristic of MR dampers, various models like, Lurge model, Dahl model and Bingham model can be applied but the modified Bouc-Wen model has been chosen for this paper due to its the flexibility of control.

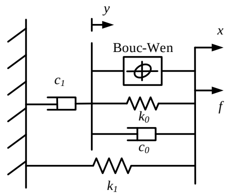

Fig. 3 Bouc-Wen model

$$f = c_1\dot{y} + k_1(x - x_0) \tag{1}$$

$$\dot{z} = -\gamma \mid \dot{x} - \dot{y} \mid z \mid z \mid^{n-1} - \beta(\dot{x} - \dot{y}) \mid z \mid^{n} + A(\dot{x} - \dot{y}) \tag{2}$$

$$\dot{y} = [\alpha z + c_0\dot{x} + k_0(x - y)]/(c_0 + c_1) \tag{3}$$

As shown in the Fig. 3, f defines the output damping force of magneto-rheological damper, c_1 denotes viscous damping coefficient at low velocity, k_0 is spring stiffness coefficient at high velocity, c_0 is viscous damping at high speed, k_1 is stiffness of shock absorber, x is relative displacement of spring (i.e, relative displacement of unsprung and sprung mass), x_0 is initial relative displacement. Where z *is* defined as evolution variable, is scale factor of Bouc-Wen hysteresis. Operator, γ, β, A, are correlation coefficients of hysteresis parameters, n is index coefficient, typically is 2. y is denoted as internal displacement, parameters c_0, k_0, k_1, x_0, γ, β, A and n are constant coefficient, parameters α and c_1 are functions of input current. Through curve fitting with linear function according to experimental data (table 1)[5].

$$\begin{cases} c_1 = c_{1a} + c_{1b}i \\ \alpha = \alpha_b + \alpha_b i \end{cases} \tag{4}$$

Table 1 Bouc-Wen model characteristics

Parameters/Unit	Values
C_0 (N.S.mm^{-1})	1333
C_{1a} (N.S.mm^{-1})	8.168
C_{1b} (N.S.mm^{-1}.A^{-1})	2.725
α_a (N. mm^{-1})	0
α_b (N.mm^{-1})	1.723
k_0 (N.mm^{-1})	0.01072
k_1 (N.mm^{-1})	0.134
x_0 (mm)	114.9
β (mm^{-2})	0.07
A	300
γ/mm^2	0.07
n	2

MR damper is a hysteretic and non-linear component due to big non-linear parameters of the magneto-rheological fluid. This non-linearity explains the relationship between the input and output. Meanwhile the output (force) Fig. 4 is the non-linear function of the electrical current supplied to the damper coil and the mechanical input which is displacement of damper piston one end of damper cylinder to the other end. The simulation process was carried out with a sinusoidal frequency of 5Hz and amplitude of 10 mm and displacement for a specific current apply and we repeated this process for every parameter combination. The responses of the MR damper for the variable input current simulation are indicated in Fig. 5. In this case: the MR damper response was obtained by changing the input current while the amplitude and frequency are kept constant. The damping force augmented along with the input current value applied on MR damper and the hysteretic characteristic is increased as shown in Fig. 6. If the damper is operating without input current, the damper response shows a reduced hysteretic loop while operating with a non-zero constant input current level the damper represents a huge important hysteretic characteristic.

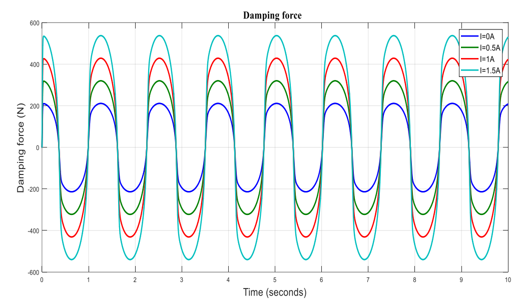

Fig. 4 Damping force (N)

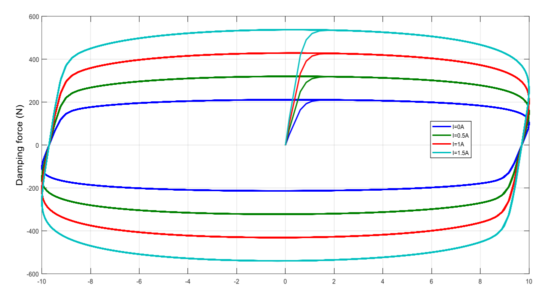

Fig. 5 Damping force vs Displacement

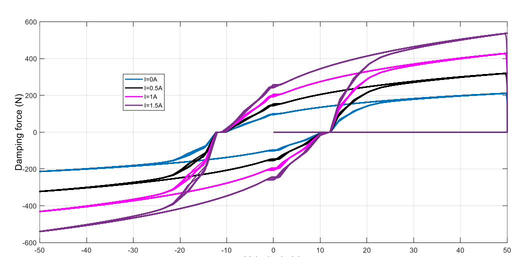

Fig. 6 Damping force versus Velocity

3. Full vehicle model

Semi-active suspension model for quarter car model can be extended to full car, as it is indicated on the Fig. 8 below, to model a full car dynamic model; CarSim environment software has been used. CarSim is software which models automotive dynamic systems via various mathematic models embedded in it. The CarSim has many feature such as validation and testing of different parameters of model[7]. it is incorporated by different models such as road input, driver aerodynamics and so on. It uses a high calculation speed of many mathematical models[8].

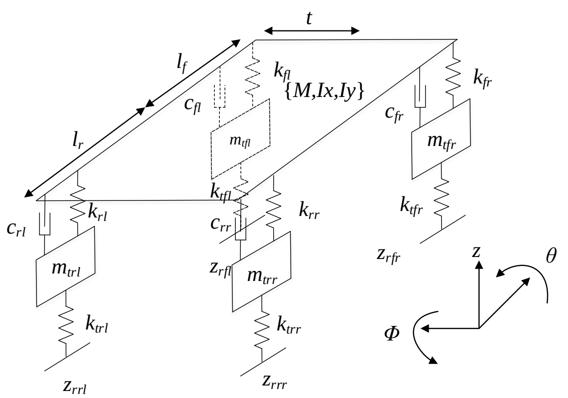

Fig.8 vehicle model

$$\begin{cases} Z_{fl} = Z + l_f \sin(\emptyset) - t\sin(\theta) \\ Z_{fl} = Z + l_f \sin(\emptyset) + t\sin(\theta) \\ Z_{rl} = Z - l_r \sin(\emptyset) - \sin(\theta) \\ Z_{rr} = Z - l_r \sin\emptyset + t\sin\theta \\ \dot{Z}_{fl} = \dot{Z} + \dot{\emptyset} l_f \cos(\emptyset) - \dot{\theta} t\cos(\theta) \\ \dot{Z}_{fr} = \dot{Z} + \dot{\emptyset} l_f \cos(\emptyset) + \dot{\theta} t\cos(\theta) \\ \dot{Z}_{rl} = \dot{Z} - \dot{\emptyset} l_r \cos(\emptyset) - \dot{\theta} t\cos(\theta) \\ \dot{Z}_{rr} = \dot{Z} - \dot{\emptyset} l_r \cos(\emptyset) + \dot{\theta} t\cos(\theta) \end{cases} \quad (6)$$

Where z defines chassis mass gravity center, φ (resp. θ) denoted as the pitch angle of the sprung mass at the gravity center. t, l_f , and l_r denote the vehicle geometrical characteristics. On this model, a notice is given to the full vertical vehicle model description which is the concatenation of the two previously introduced half models, including vertical, pitch and roll dynamics. Primo, assumptions under which the model is analyzed are introduced, then, according to the car geometry properties the kinematic equations are generated, and lastly, hereinafter are the nonlinear vertical dynamical equations

$$\begin{cases} M\ddot{z} = -F_{sz_{fl}} - F_{sz_{fr}} - F_{sz_{rl}} - F_{sz_{rr}} \\ m_{t_{ij}}\ddot{z}_{t_{ij}} = F_{sz_{ij}} - F_{tz_{ij}} \\ I_x\ddot{\theta} = \left(F_{sz_{rl}} - F_{sz_{rr}}\right)t + \left(F_{sz_{fl}} - F_{sz_{fr}}\right)t + M_{dx} \\ I_y\ddot{\emptyset} = \left(F_{sz_{rr}} + F_{sz_{rl}}\right)l_r - \left(F_{sz_{fr}} + F_{sz_{fl}}\right)l_f + M_{dy} \end{cases} \quad (7)$$

where the vertical tire and suspension forces are defined as

$$\begin{cases} F_{t_{ij}} = K_t\left(K_{t_{ij}} - Z_{r_{ij}}\right) \\ F_{s_{ij}} = K\left(Z_{ij} - Z_{t_{ij}}\right) + C_{ij}\left(\dot{Z}_{ij} - \dot{Z}_{t_{ij}}\right) \end{cases} \quad (8)$$

Where $m_{t_{ij}}$ and M hold for the unsprung masses and body mass (sprung mass). The inertia of vehicle in the x-axis (resp. y-axis) is denoted as Ix (resp. Iy). FL (resp. M_{dx} and M_{dy}) are external forces (resp. moments) disturbances on the x (resp. y and z} axes[9]. Fig.9 shows the vehicle dynamic parameters used in this model where every MR damper is modeled and simulated by a nonlinear model.

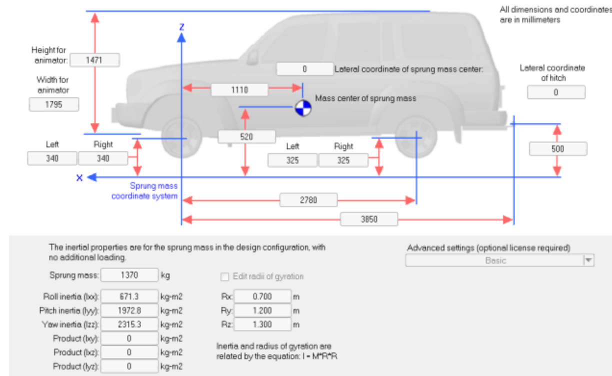

Fig. 9 CarSim full vehicle suspension parameters

4. Simulation procedure

The model is composed by vehicle model (CarSim model), Bouc-wen model, current driver model and 2 controllers: a system controller and a damper controller. The main purpose of system controller is to generate the required damping force. The second controller serves on controlling the voltage to be applied to the current driver to track the desired damping force. The damping force provided by MR damper is fed to the vehicle dynamics to attenuate the vibration caused by uneven road [10] Fig. 10.

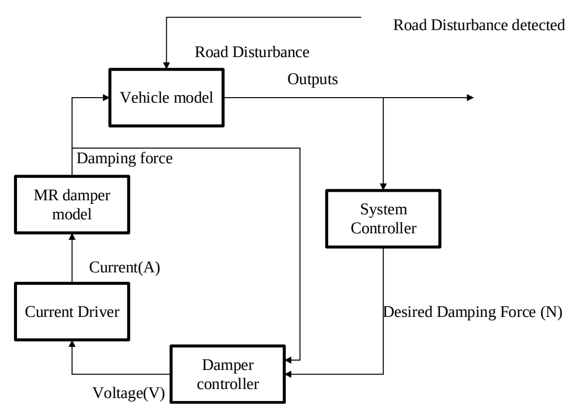

Fig. 10 Block diagram of semi-active suspension system

4.1 System controller

To achieve the desired damping force, fuzzy logic controller has been utilized. Here it is used to control damping force of suspension system. Fuzzy logic algorithm controls many loops variation in various non-linear systems. On this model (Fig. 11) it has two inputs, suspension deflection and suspension velocity the controller computes those inputs and provides the desired damping force based on inference rules subjected on it, the figures below explain the process of fuzzy logic, these control systems embed the human as thinking through the application of fuzzy configurations and linguistic variables related by a configuration of IF-THEN fuzzy rules as indicated in table 2 and Fig 15. In this model, triangular shaped membership functions (MFs), as it indicated in Fig. 12, Fig. 13 and Fig. 14 are utilized to represent the input variables, which are suspension deflection and suspension velocity.

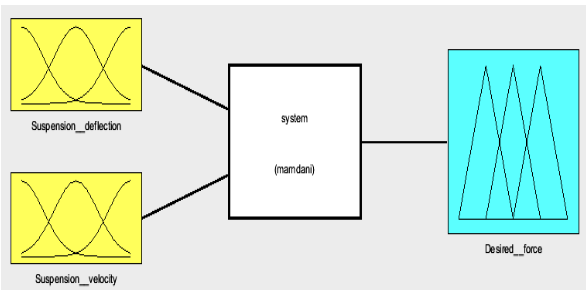

Fig. 11 Fuzzy controller

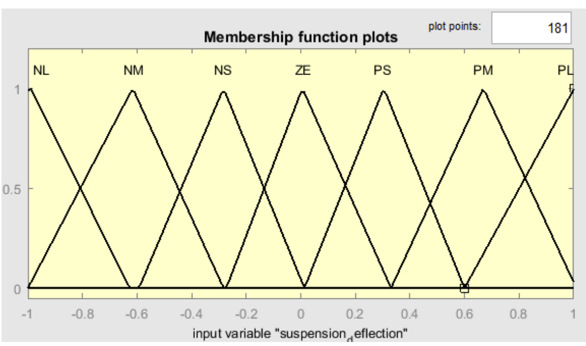

Fig. 12 Suspension deflection (MFs)

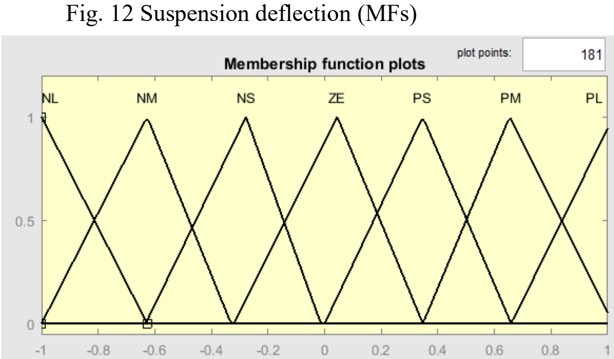

Fig. 13 suspension velocity

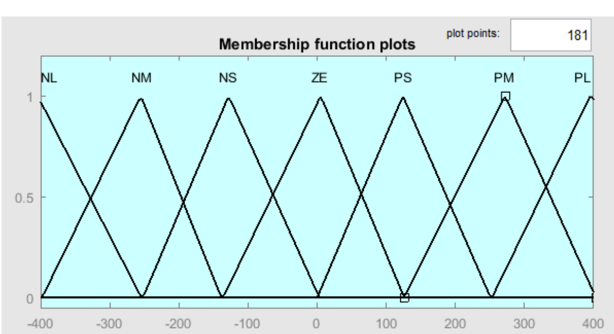

Fig. 14 Desired damping force (N)

The fuzzy controller has seven linguistic variables for each input: NL, negative large, NM negative medium, NS negative small, ZE zero, PS positive Small, PM, positive medium and PL positive large.

Table 2 Fuzzy inference rules

velocity	Suspension deflection						
	PL	PM	PL	ZE	NS	NM	NL
NL	PM	PS	ZE	NS	NS	NS	NM
NM	PM	PM	PS	NS	NM	NM	NM
NS	PL	PL	PM	NS	NM	NM	NB
ZE	PL	PL	PM	ZE	NM	NL	NB
PS	PL	PM	PM	PS	NM	NL	NB
PM	PM	PM	PM	PS	NS	NM	NM
PL	PM	PS	PS	PS	ZE	NS	NM

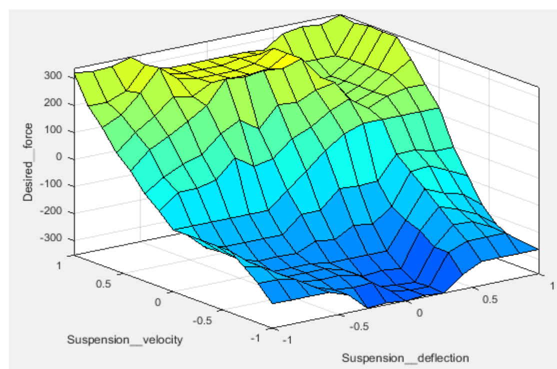

Fig. 15 Fuzzy map

4.2 Damper controller

The role of damper controller is to provide a proper command of voltage to the current driver by tracking the desired damping force and actual damping. The current driver has a role of supplying the current the damper coil.

$$v = V_{max}H[(f_c - f_d)f_cf_d] \qquad (9)$$

Where Vmax is defined as the maximum voltage applied to the current driver related with saturation of the magnetic field in the MR damper, H (·) is the Heaviside step function, fc the desired damping force, fd the actual damping force[4].

4. Simulation results

The simulation was performed by extending quarter car model to full vehicle model which is based on four wheels of vehicle model (7 degree of freedom); the vehicle model was performed in CarSim and control strategy in Matlab/ Simulink. On this model we used the independent suspensions where every MR damper is controlled separately Fig. 16.

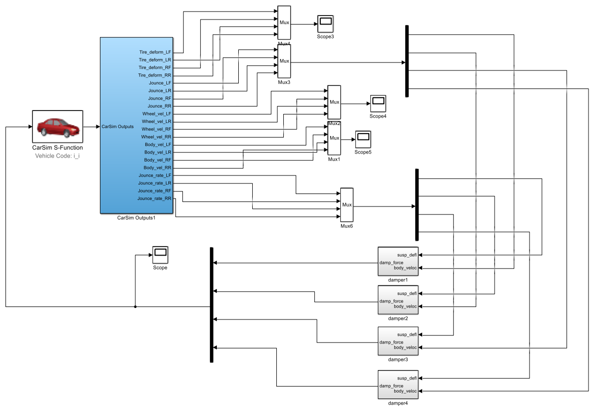

Fig. 16 CarSim full vehicle suspension model

The simulation tests made are based on the comparison of four MR dampers and passive damper. MATLAB/Simulink software has been utilized to model the control strategy and full vehicle dynamic model was modeled in CarSim software shown in Fig. 16. The rough road model and E-class Sedan vehicle has been selected to model and test the semi-active suspension system of full car model. The real-time simulation has been performed. The vehicle measurements were acquired from CarSim and Simulink implements the control algorithm and generates semi-active damping forces for every damper. The damping forces of each damper transmitted to the vehicle model in CarSim. The whole simulation has been analyzed in Simulink. To test the model; the road mode with a shaped bump of 3.4 cm of height has been selected to perform the simulation as it is indicated on the Fig. 17.

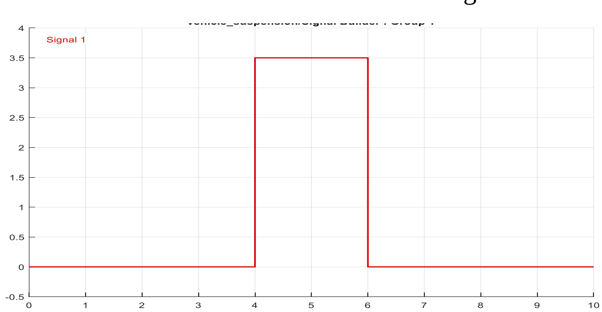

Fig. 17 Road model for full car model

As result of simulation, we observed that whenever the vehicle passes on road irregularities the damper controller generates more current pulses to increase the damping force in order to produce the stability and comfort of vehicle occupants as it shown on fig. 18.

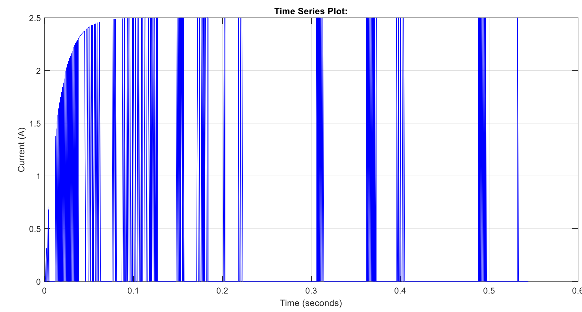

Fig. 18 Current pulse subjected to the damper in Ampere

The Fig. 19, shows four dampers forces, the variation of damping force depends on road disturbance subjected on it. More damping forces determine high road handling; inversely low damping force provides a good comfort for vehicle occupants.

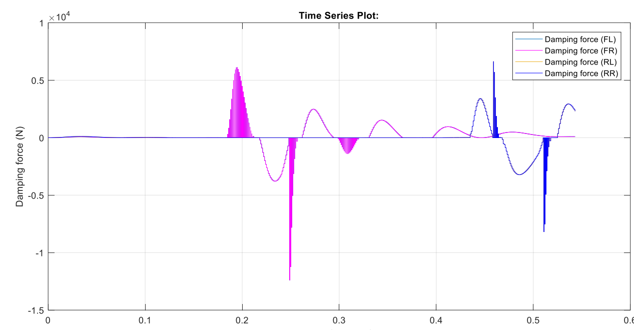

Fig. 19 MR damper damping forces

The Fig. 20, fig. 21, fig. 22 and fig. 23 indicate body velocity which are the index of ride comfort, the low variation of velocity defines the ride comfort; in this figure the semi active suspension amplitude is lower than passive one.

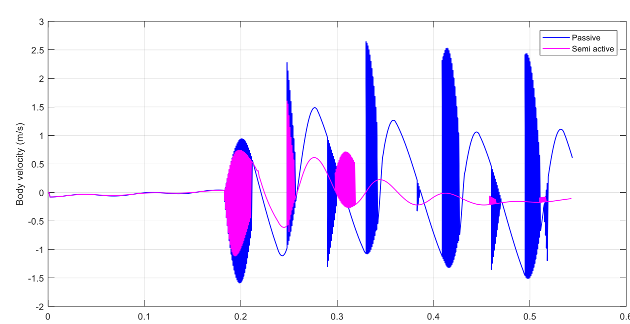

Fig. 20 Body velocity front left (m/s)

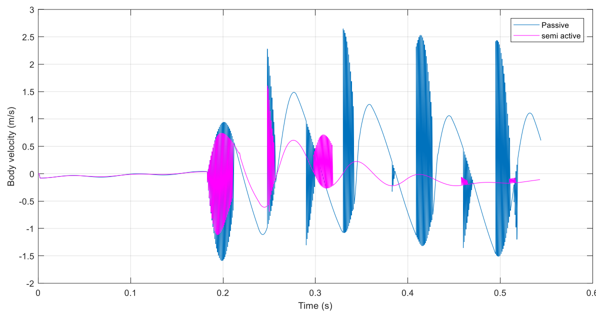

Fig. 21 Body velocity on front right (m/s)

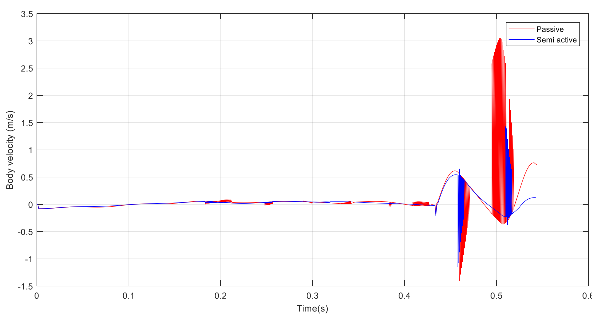

Fig. 22 Body velocity on rear left (m/s)

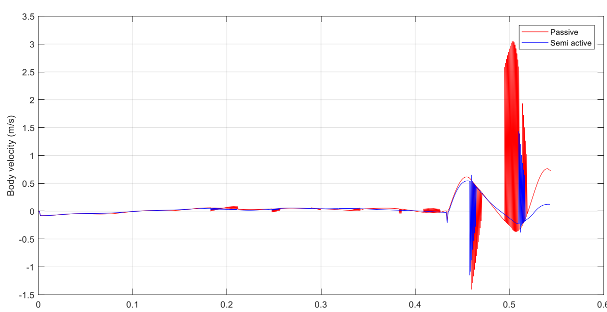

Fig. 23 Body velocity on rear right (m/s)

The Figures below indicate the comparison of the road handling, between passive and semi active, which is determined by the deflection of Front wheels and rear wheels.

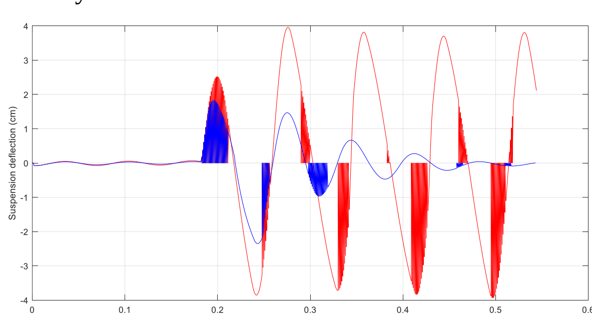

Fig. 24 suspension deflection on front left (cm)

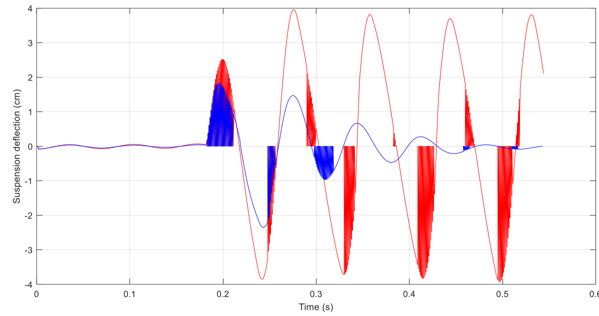

Fig. 25 suspension deflection on front right (cm)

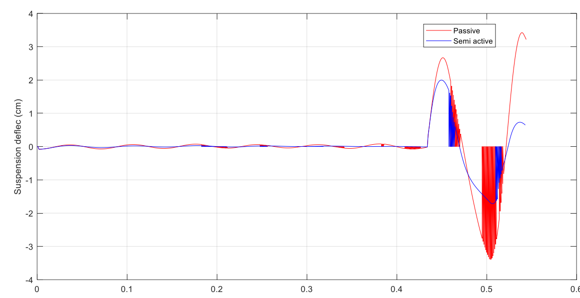

Fig. 26 suspension deflection on rear left (cm)

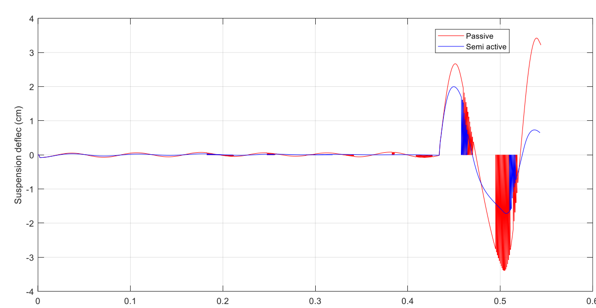

Fig. 27 suspension deflection on rear right (cm)

The RMS comparison between passive suspension and semi-active suspension graphs; semi-active shows a significant improvement on ride comfort and road tire handling as it is shown on the table 3 below.

Table 3. Comparison between passive and semi active suspension

RMS	Passive suspension	Semi active suspension	Improvement %
Body velocity m/s left front	0.6725	0.4142	38.43%
Body velocity m/s left rear	0.6725	0.4142	38.43%
Body velocity m/s front	0.6814	0.4371	35.85%
Body velocity m/s right rear	0.6814	0.4371	35.85%
suspension deflection on front left (m)	0.5208	0.4027	22.67%
suspension deflection on fromt right (m)	0.5208	0.4027	22.67%
suspension deflection on rear left (m)	0.5098	0.4162	18.36%
suspension deflection on rear right (m)	0.5098	0.4162	18.36%

By using a combination of fuzzy logic and damper controller system there is a big contribution on improvement of suspension trade off. The comfort has been increased 38.43% compare to passive suspension and the road handling has been increased 22.67 %.

5. Conclusion

This paper entitled vehicle comfort optimization based on MR damper is composed by three many parties; road model, full car dynamic model and controllers. A road model and vehicle model has been modeled in Carsim and control strategies in Matlab/Simulink. The control strategies based on fuzzy logic and damper controller for tracking desired force have been used to control and monitor the damping force of MR damper. By considering vertical motion of full car model, mathematical equations of second law of Newton has been performed using a 7 degree-of-freedom model of full car model. Fuzzy controller design approach has been investigated for the semi-active system where suspension deflection as well as body displacement has greatly minimized compare to passive suspension. The output of simulation of full vertical model showed a significant improvement of more than 35% of ride comfort and 22% stability handling. The simulation has been performed on straight road with irregular surface at a constant speed of 40 km/h. By concluding in the future research more efficient between ride comfort the road handling in different road situation will be optimized, by combining various control strategies like neuro-network and fuzzy logic controller.

References

[1] X.-m. Dong, M. Yu, C.-r. Liao, and W.-m. Chen, 2010. *Comparative research on semi-active control strategies for magneto-rheological suspension*, Nonlinear dynamics, vol. 59, pp. 433-453.

[2] Chen Bing, Zeng Ming, and Yin Zhongjun, 2008. *Design and Simulation of Fuzzy Control Strategy for Vehicle Semi-active Suspension,* Journal of System Simulation, vol. 20, pp. 420-424.

[3] Yan Wenjun, Dong Dan, Wang Weirui, Wu Shen, and Liu Zhe,2011 "*Fuzzy Control Strategy for Nonlinear Semi-Active Suspension Systems,*" Control Engineering, vol. 18, pp. 941-946.

[4] K. Sosthene, M. Josee, and X. Hui, 2018. "*Fuzzy logic controller for semi active suspension based on magneto-rheological damper.*" *International Journal of Automotive Engineering and Technologies,* vol. 7, pp. 38-47, 2018.

[5] Y. Shen, M. Golnaraghi, and G. Heppler, 2005. *Analytical and experimental study of the response of a suspension system with a magnetorheological damper*, Journal of intelligent material systems and structures, vol. 16, pp. 135-147.

[6] C. C. De Wit and P. Tsiotras, 1999. *Dynamic tire friction models for vehicle traction control.* in Decision and Control, 1999. Proceedings of the 38th IEEE Conference on, 1999, pp. 3746-3751.

[7] J. D. Setiawan, M. Safarudin, and A. Singh, 2009. *Modeling, simulation and validation of 14 DOF full vehicle model*, in Instrumentation, Communications, Information Technology, and Biomedical Engineering (ICICI-BME), International Conference on, 2009, pp. 1-6.

[8] T. Kinjawadekar, N. Dixit, G. J. Heydinger, D. A. Guenther, and M. K. Salaani, 2009. V*ehicle dynamics modeling and validation of the 2003 Ford Expedition with ESC using CarSim*, SAE Technical Paper 0148-7191.

[9] A. Kruczek and A. Stribrsky, 2004. *A full-car model for active suspension-some practical aspects," in Mechatronics,* 2ICM'04. Proceedings of the IEEE International Conference on, 2004, pp. 41-45.

[10] D. Wang and W. H. Liao, 2011. *Magnetorheological fluid dampers: a review of parametric modelling*, Smart materials and structures, vol. 20, p. 023001.