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Analysis of MR Damper for Quarter and Half Car Suspension Systems of a Roadway Vehicle

D.V.A. Rama Sastry[a,b], K.V. Ramana[a], N. Mohan Rao[c], P. Pruthvi[a] and D.U.V. Santhosh[a]

[a]*Dept.of Mech. Engg., K L University, Vaddeswaram, Guntur, India*
[b]*Corresponding Author Email: dvarsastry@kluniversity.in*
[c]*Dept.of Mech. Engg., University College of Engg.,JNTUK, Kakinada, Andhra Pradesh, India*

ABSTRACT:

Magnetorheological (MR) dampers are evolving as one of the most promising devices for semi-active vibration control of various dynamic systems. In this paper, the suspension system of a car using MR damper is analysed for 2DOF quarter car and 4DOF half car models and then compared with corresponding suspension system using passive damper for ride comfort and handling. Magnetorheological damper is fabricated using a MR fluid of Carbonyl iron powder and Silicone oil added with additive. Experiments are conducted to establish the behaviour of the MR damper and are used to validate Spencer model for MR damper. Further, using the validated Spencer model of MR damper, the quarter car and half car models of Vehicle Suspension system are simulated by implementing a semi-active suspension system for analysing the resulting displacement and acceleration in the car body. The ride comfort and vehicle handling performance of each specific vehicle model with passive suspension system are compared with corresponding semi-active suspension system. The simulation and analysis are carried out using MATLAB/SIMULINK.

KEYWORDS:

Magnetorheological dampers, Suspension systems; Spencer model; Simulation and fabrication of MR damper

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1. Introduction

Achieving a right combination of ride comfort and vehicle handling is posing challenge to all automobile manufacturers today. With passive suspension systems, either of these needs to be comprised [1] and with active suspension systems, complexity, cost and stability issues are on raise [2]. The emergence of semi-active suspension systems, especially Magnetorheological (MR) damper, has lead to many interesting studies in vibration control of vehicles [3]. MR damper is very much like hydraulic damper in construction with an electrical coil wounded around the piston head and basically consists of MR fluid which consists of suspensions of non-colloidal, multi-domain (0.05-10μm) and magnetically soft particles in organic or aqueous liquids [4]. MR fluid can change reversibly from free-flowing, linear viscous liquids to semi-solids having controllable yield strength under a magnetic field, which can be controlled using electric coil around piston head. The apparent viscosity of MR fluid changes significantly (105−106 times) within a few milliseconds, when the magnetic field is applied [5]. There are various parametric models proposed to represent the behaviour of MR damper. The model proposed by Spencer [6] modifying Bouc Wen model, is supposed to be most appropriate model among all the available models. In this paper, the suspension system of a car using MR damper is analysed for 2DOF quarter car and 4DOF half car models and then compared with corresponding suspension system using passive damper for ride comfort and handling. The quality of ride gets influenced by acceleration and vehicle handling by displacement of disturbances [7].

2. Experimental setup and procedure

Fig. 1 shows the experimental setup used for testing the MR damper. In order to determine the amount of damping force that is needed for the dampers, the damper has been tested using VIB- LAB apparatus with speed regulator, NI-COMPACT DAQ with NI-lab view software installed in laptop, VARIAC (variable voltmeter), load cell and LVDT. A MR damper was developed with dimensions as given in Table 1, which are similar to the existing dampers. The coil consisting of double wired parallel winding with a copper wire of 25gauge and 200 turns, which has a resistance of 3.5Ohms, is housed in the piston head assembly. The developed experimental setup is used for testing of MR damper fabricated, at 0.4 amps current on VIB-Lab. Using data acquisition system (NI Compact DAQ) connected to the system, the damping force data in variation to displacement caused by roadway irregularities is acquired. For the present investigation, the vehicle considered is Hyundai i20 with curb weight of 1180kg and gross total weight of 1580kg, of which 180kg is the weight of total unsprung mass leading to 45kg at each wheel, and therefore 1500kgs is considered as total sprung mass, of which 60% acts on rear side leading to 450kg on each rear wheel.

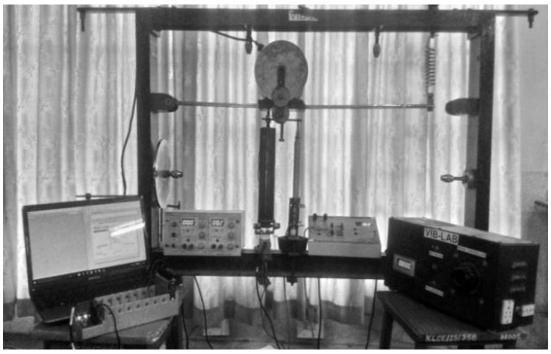

Fig. 1: Experimental setup for testing MR damper

Table 1: Geometry of MR Damper

Damper Parameter	Dimension (mm)
Extended height	380
Compressed height	360
Stroke length	20
Damper tube length	300
Damper tube outer diameter	60
Damper tube inner diameter	50
Piston head diameter	48
Piston rod diameter	12

3. Modelling of MR damper using Spencer model

The Spencer model for MR damper is shown in Fig. 2. According to Spencer model, the damping force produced due to the MR damper is given by,

$$f_{MR} = \alpha Z + C_o(\dot{x} - \dot{y}) + K_O(x - y) + K_1(x - x_o)$$

$$\dot{Z} = -\gamma|\dot{x} - \dot{y}||Z||Z|^{n-1} - \beta(\dot{x} - \dot{y})|Z|^n + A(\dot{x} - \dot{y})$$

$$\dot{y} = \frac{1}{C_O + C_1}\{\alpha Z + C_O\dot{x} + K_O(x - y)\}$$

The values of various parameters mentioned in the equation at 0.4amps of current supply [8] are shown in Table 2. These equations are modelled in MATLAB/SIMULINK and shown in the Fig. 3. The comparison of Spencer model analysed in MATLAB/SIMULINK with experimental values is shown in the Figs. 4 to 6. It can be observed that the results obtained from Spencer model using MATLAB/SIMULINK are comparable with experimental values and thus validates the Spencer model. So, the Spencer model is adapted to represent behaviour of MR damper for further analysing quarter car and half car models in SIMULINK.

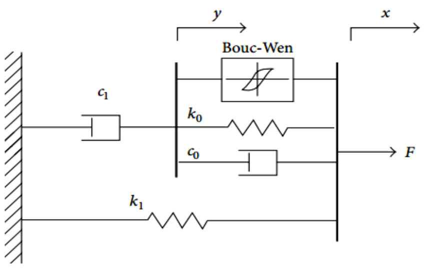

Fig. 2: Spencer model of MR damper

Table 2: Spencer damper model parameters [8]:

System Parameter	Values at 0.4 amp
α	866.65 N/cm
β, γ	119842 cm^{-1}, 5226 cm^{-1}
N, A	2, 29.37
K_1, K_0	6.7530 N/cm, 2807.8 N/cm
C_1, C_0	20626.5 Ns/cm, 1965.10 Ns/cm

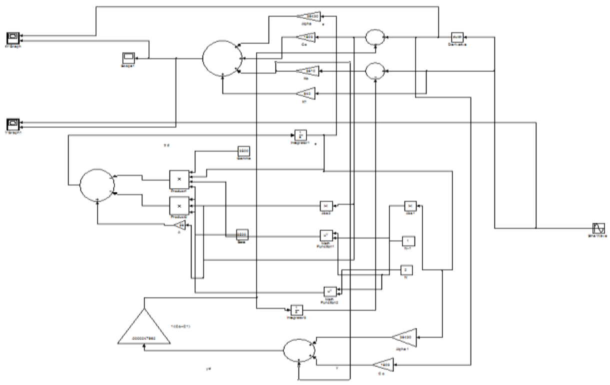

Fig. 3: Simulink modelling of Spencer model of MR damper

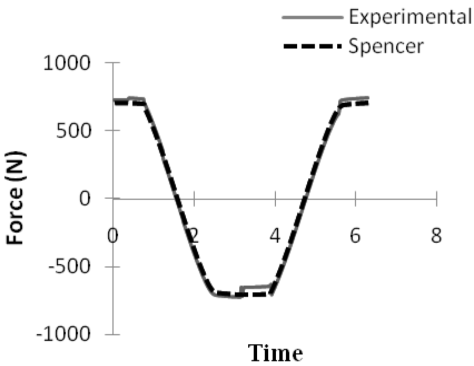

Fig. 4: Comparison of Force vs. Time of Spencer model for MATLAB/SIMULINK and Experimental analysis

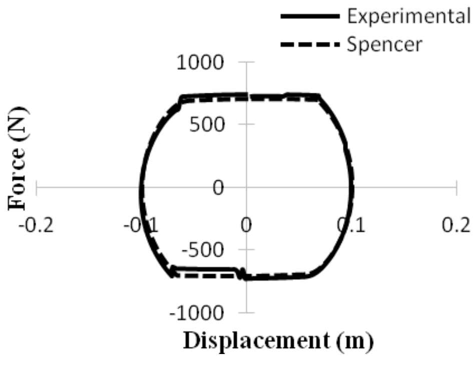

Fig. 5: Comparison of Force vs. Displacement of Spencer model for MATLAB/SIMULINK and Experimental analysis

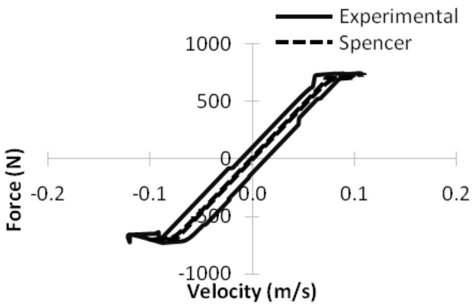

Fig. 6: Comparison of Force vs. Velocity of Spencer model for MATLAB/SIMULINK and Experimental analysis

4. Modelling & analysis of quarter car

Fig. 7 represents the quarter car model for passive suspension system. The equations of motion for this model are given as,

$$m_s\ddot{x}_s = -[K_S(x_s - x_u) + C_S(\dot{x}_S - \dot{x}_U)]$$

$$m_u\ddot{x}_u = -\{-[K_S(x_S - x_u) + C_S(\dot{x}_s - \dot{x}_u)] + [K_S(x_u - q) + C_t(\dot{x}_u - \dot{q})]\}$$

Using the parameters of suspension systems as mentioned in Table 3, the passive suspension system of quarter car is analysed to identify the displacement and acceleration of the sprung mass. Modelling is carried out in MATLAB/SIMULINK and shown in Fig. 8.

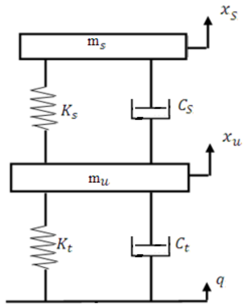

Fig. 7: Quarter car model for passive suspension system

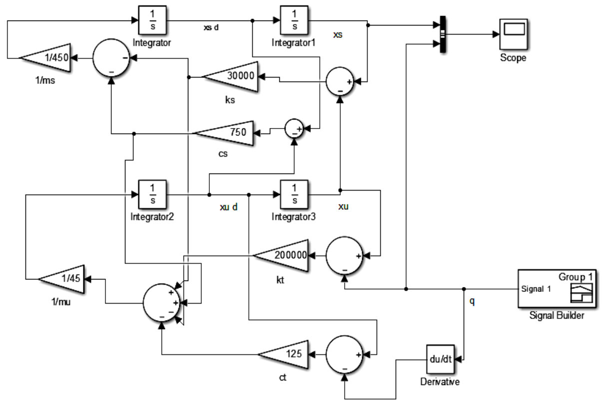

Fig. 8: MATLAB/SIMULINK model of quarter passive suspension system

Table 3: Parameters of quarter passive suspension system

System parameter	Value
Sprung mass (m_x)	450kg
Unsprung mass (m_u)	45kg
Suspension stiffness (k_s)	300N/cm
Damping coefficient (c_s)	7.50Ns/cm
Tire Suspension stiffness (k_t)	2000N/cm
Tire Damping coefficient (c_t)	1.25Ns/cm

Fig. 9 represents the quarter car model for Semi-active suspension system. The equations of motion for this model are given by,

$$m_s\ddot{x}_s = -[K_S(x_s - x_u) + f_{MR})]$$

$$m_s\ddot{x}_u = -[K_S(x_s - x_u) + f_{MR})] + [K_t(x_u - q) + C_t(\dot{x}_u - \dot{q})]$$

Using the parameters of suspension systems as mentioned in Tables 2 and 3, the semi-active suspension system of quarter car is analysed to identify the displacement and acceleration of the sprung mass. Modelling is carried out in MATLAB/SIMULINK and shown in Fig. 10.

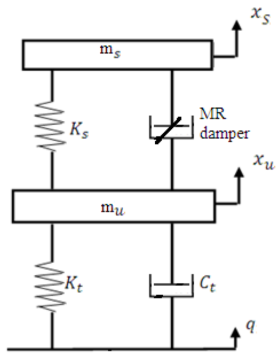

Fig. 9: Quarter car model for semi-active suspension system

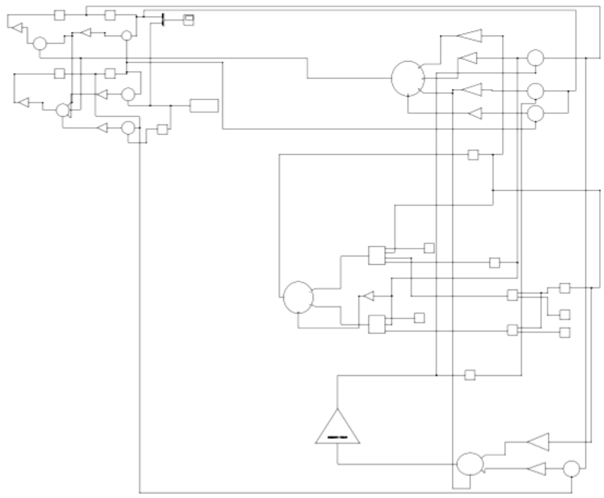

Fig. 10: Matlab/Simulink quarter car model of semi-active suspension system

The comparison of quarter car passive and semi-active suspension systems in terms of variation of displacement and acceleration with respect to time are shown in Figs. 11 and 12 respectively. The actual disturbance applied is a bump as shown in Fig. 11. Comparisons of different parameters analysed for passive and semi-active systems are shown in Table 5. In case of quarter car analysis, the maximum displacement for passive system is 1.792 and semi active system is 1.226, which is reduced by 31%. The maximum acceleration for passive system is 138.26 and semi active system is 102.98, which is reduced by 25%.

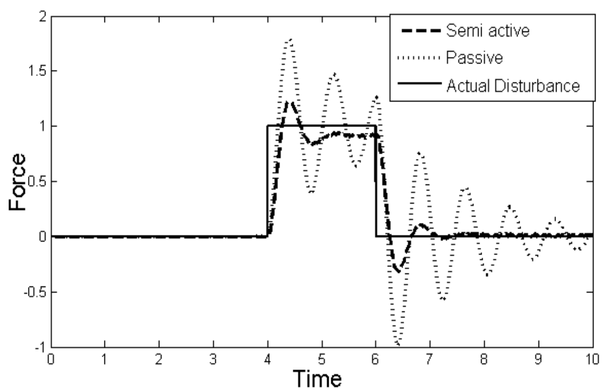

Fig. 11: Displacement for passive & semi-active quarter car model

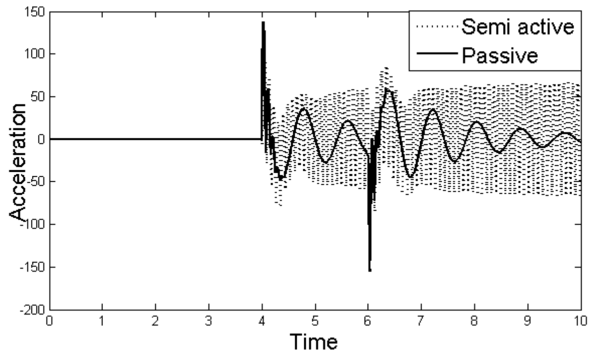

Fig. 12: Acceleration vs. Time for passive and semi-active quarter car model

5. Modelling & analysis of half car

Fig. 13 represents the half car model for passive suspension system. The equations of motion for this model are given as,

$$m_1\ddot{x}_{u1} = \left[K_{s1}(X_3 + b\theta - X_{u1}) + C_{31}\left(\dot{X}_3 + b\dot{\theta} - \dot{X}_{u1}\right) - \left[K_t(X_{u1} - q_1) + C_t(\dot{X}_{u1} - \dot{q}_1)\right]\right]$$

$$m_2\ddot{x}_{u2} = \left[K_{s2}(X_3 + a\theta - X_{u2}) + C_{32}\left(\dot{X}_3 - a\dot{\theta} - \dot{X}_{u2}\right) - \left[K_t(X_{u2} - q_2) + C_t(\dot{X}_{u2} - \dot{q}_2)\right]\right]$$

$$m_3\ddot{x}_{u3} = -\left[K_{s1}(X_3 + b\theta - X_{u1}) + C_{31}\left(\dot{X}_3 + b\dot{\theta} - \dot{X}_{u1}\right) + \left[K_{32}(X_3 - a\theta - X_{u2}) + C_{32}(\dot{X}_3 - a\dot{\theta} - \dot{X}U_2)\right]\right]$$

$$I_\theta\dot{\theta} = -\left[\left[K_{s2}a(X_3 - a\theta - X_{u2}) + C_{32}a\left(\dot{X}_3 - a\dot{\theta} - \dot{X}_{u2}\right)\right] - \left[K_{31}b(X_3 + b\theta - X_{u1}) + C_{31}b(\dot{X}_3 + b\dot{\theta} - \dot{X}U_1)\right]\right]$$

Using the parameters of suspension systems as mentioned in Table 4, the passive suspension system of quarter car is analysed to identify the displacement and acceleration of the sprung mass. The modelling is carried out in MATLAB/SIMULINK as shown in Fig. 14.

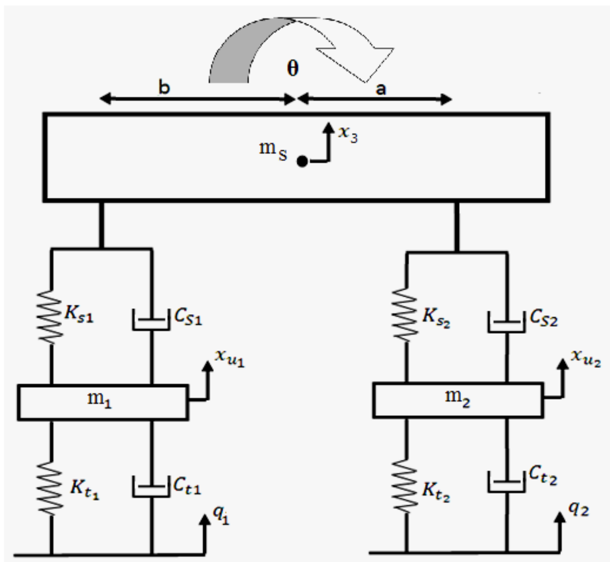

Fig. 13: Half car model for passive suspension system

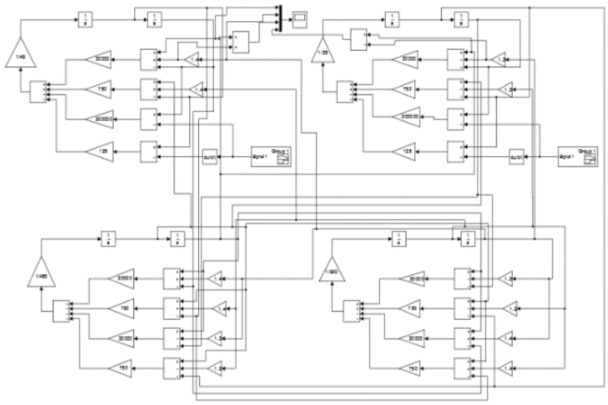

Fig. 14: Matlab/Simulink half car model for passive suspension system

Table 4: Parameters of half car model for passive suspension system

System parameter	Value
Sprung mass (m_s)	450kg
Unsprung mass (m_u)	45kg
Front suspension stiffness(k_{s1})	300N/cm
Front damping coefficient (c_{s1})	7.50Ns/cm
Tire suspension stiffness(k_t)	2000N/cm
Tire damping coefficient (c_t)	1.25Ns/cm
Rear suspension stiffness(k_{s2})	250N/cm
Rear damping coefficient (c_{s2})	7.50Ns/cm

Fig. 15 represents the half car model for semi-active suspension system. The equations of motion for this model are given as,

$$m_1\ddot{x}_{u1} = -\left[\left[K_{s1}(X_3 - b\theta - X_{u1}) + F_{MR1}\right] - \left[K_t(X_{u1} - q_1) + C_t(\dot{X}_{u1} - \dot{q}_1)\right]\right]$$

$$m_2\ddot{x}_{u2} = -\left[\left[K_{s2}(X_3 + a\theta - X_{u2}) + F_{MR2}\right] - \left[K_t(X_{u2} - q_2) + C_t(\dot{X}_{u2} - \dot{q}_2)\right]\right]$$

$$m_3\ddot{x}_{u3} = -\left[K_{s1}(X_3 - b\theta - X_{u1}) + F_{MR1} + \left[K_{s2}(X_3 + a\theta - X_{u2}) + F_{MR2}\right]\right]$$

$$I_\theta\dot{\theta} = -\left[\left[K_{s2}a(X_3 + b\theta - X_{u2}) + aF_{MR2}\right] - \left[K_{s1}b(X_3 - b\theta - X_{u1}) + bF_{MR1}\right]\right]$$

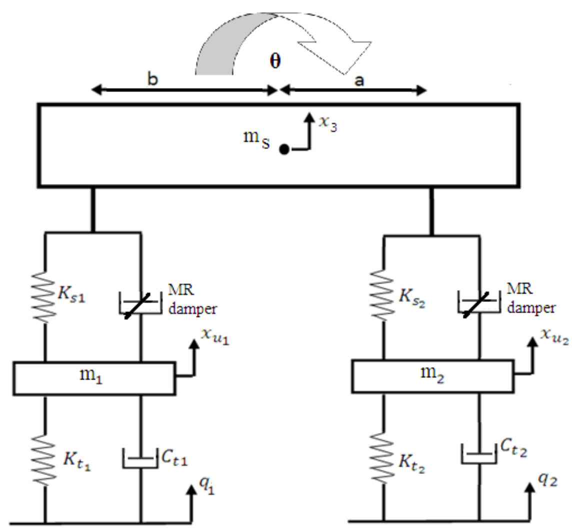

Fig. 15: Half car model for semi-active suspension system

Using the parameters of suspension systems as mentioned in Tables 2 and 4, the semi-active suspension system of quarter car is analysed to identify the displacement and acceleration of the sprung mass. Mathematical modelling is carried out in MATLAB/SIMULINK and shown in Fig. 16. The comparison of half car passive and semi-active suspension systems in terms of variation of displacement at centre of Gravity, pitch and acceleration with respect to time are shown in Figs.18 to 20 respectively. The front wheels are excited with a bump that starts at 2 seconds and ends at 4 seconds and rear wheels are excited with a bump that starts at 5 seconds and ends at 7 seconds as shown in Fig. 17. In case of half car analysis, the maximum displacement for passive system is 0.973 and semi active system is 0.82 which is reduced by 15%. The maximum pitch for passive system is 0.69 and semi active system is 0.68 which is reduced by 1.44%. The maximum acceleration for passive system is 143.49 and semi active system is 79.36, which is reduced by 44%. This analysis clearly indicates the semi active suspension is more effective than passive in dealing with ride comfort and handling of vehicles.

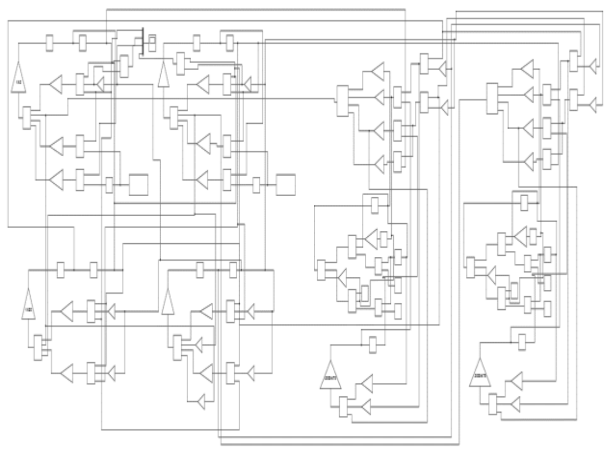

Fig. 16: MATLAB/SIMULINK model of half car semi-active suspension system

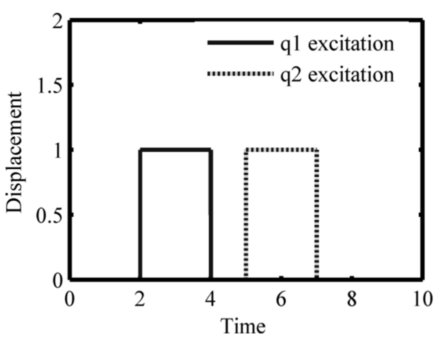

Fig. 17: Excitation applied for half car

Table 5: Comparison of different parameters analysed for passive and semi-active systems

Parameter analysed	Passive	Semi-active	Damping %
Displacement for quarter car	1.792	1.226	31
Acceleration for quarter car	138.26	102.98	25
Disp. of C.G for half car	0.973	0.82	15
Pitch for half car	0.69	0.68	1.44
Acceleration for half car	143.49	79.36	44

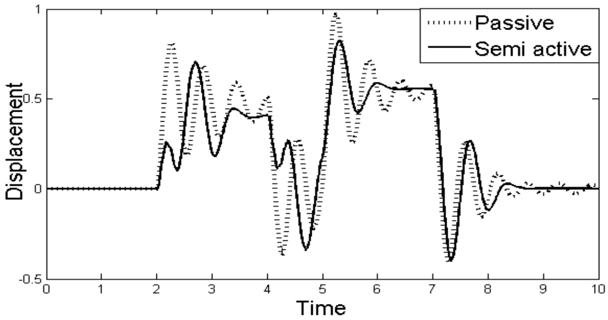

Fig. 18: Comparison of displacement at centre of gravity for passive and semi-active half car model.

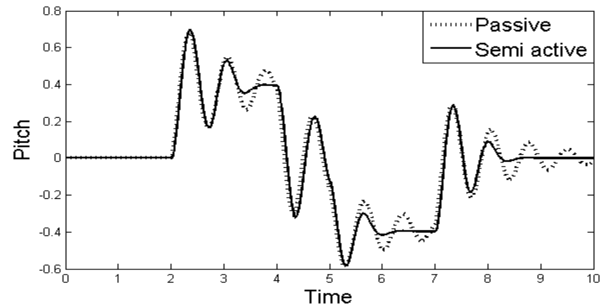

Fig. 19: Comparison of pitch for passive and semi-active half car model.

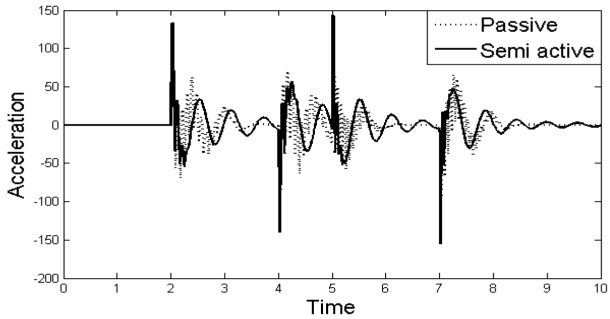

Fig. 20: Acceleration vs. Time for semi-active half car model

6. Conclusion

In this paper, quarter and half car suspension systems are analysed separately for the passive suspension system and Semi-active suspension system using MR dampers. Further, the passive suspension system and Semi-active suspension systems using MR dampers are compared for each of these cases. An attempt has been made to investigate both theoretically and experimentally to establish the advantage of replacing the existing passive dampers of 4 wheelers by Magnetorheological fluid dampers. The results presented in this paper for bump excitation have shown that by replacing Passive damper with MR damper, displacement that influences road handling of car body has reduced by 31% and 15% for quarter car and half car respectively. Similarly, by replacing Passive damper with MR damper, acceleration that influences ride comfort of car body has reduced by 25% and 44% for quarter car and half car respectively. The comparison of results is a clear indication that replacement of Passive dampers with MR dampers in suspension system of automobiles provides better ride comfort and vehicle handling. The concept can be extended to a full car for ensuring comfort of the passengers.

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