International Journal of Technical Research and Applications e-ISSN: 2320-8163,
www.ijtra.com Volume 2, Issue 6 (Nov-Dec 2014), PP. 81-85

A MODEL BASED APPROACH FOR DESIGN OF SEMIACTIVE SUSPENSION USING VARIABLE STRUCTURE CONTROL

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Abstract— This paper presents the modeling and simulation of a magnetorheological damper based semiactive suspension using variable structure controllers. Passive suspension systems tend to limit the trade-off between passenger comfort and road handling. But Semiactive suspensions can reduce this trade-off margin and dynamically respond to the damping requirements. Active suspensions provide the best response since they can add damping force in any direction, but are prone to higher power consumption. Semiactive suspensions just change the damping coefficient by simply applying a control voltage as and when required. The performance of three controllers- sigma 1, sigma 2 and sigma 3, are measured and analyzed using nine parameters using peak, root mean square and normalized approaches. The road excitations considered are a single road hump and random road disturbance. The control system is applied to a 2-degree of freedom quarter car model of a passenger car. A modified Bouc-Wen model of MR damper is used to cater to the system responses at near zero velocities. The performance of these controllers is superior to the uncontrolled case, which is similar to passive suspension system. Sigma 3 controller is superior to the uncontrolled system by 63% while sigma 1 and sigma 2 are superior by 53% when it comes to peak suspension deflection for a random road disturbance. Both sigma 2 and sigma 3 controllers are better in terms of performance. The validation of the semiactive suspension leads to selection of sigma 2 controller over sigma 3 controller because of its simplicity in implementation in real-time systems.

Key Words— Semiactive suspension, magnetorheological damper, variable structure controller, quarter car model.

I. INTRODUCTION

Model Driven Development (MDD) is increasingly becoming relevant both in industry and research. MDD stresses the use of models in real time system development life cycle and facilitates automation through model execution and model transformation techniques.

In the first phase the system model for the application is developed and simulated for performance. In the next phase the computation models are implemented on an embedded platform using real world road excitations. The proposed model can be used for the embedded platform development of a semiactive suspension for passenger car using magnetorheological (MR) damper.

Automotive suspension research is one of the current trends in industry due to the boom in automotive sales all over the world. Ride comfort and road handling play a key role in safety and luxury, especially for the car segment. The conventional suspensions employed passive damping, where a trade-off was made between road comfort and road handling at different frequencies of road excitations. In passive suspensions, if we chose a low damping ratio we gain superior high-frequency isolation, but poor resonant frequency control. If we increase the damping ratio we begin to trade-off the high-frequency isolation for resonance control.

In [1] mathematical modeling of passive suspension system and control strategies of semiactive suspension is presented. Passive damping involves tuning of the system at the time of manufacture through field trials and simulations, for a predetermined comfort vs. road handling trade-off.

The parameters of the suspension system tend to drift due to ageing and other extreme atmospheric conditions, thereby affecting the performance. Active damping provided a solution to this problem by actively controlling the damping force required through controllers in real time. This consumes more energy and hence less efficient than the latest trend of semiactive damping in suspension systems. Through an imaginary concept of skyhook principle it was demonstrated that by providing a damping force through an already moving damper, we need not depend entirely on external power for generating the required damping force. The semiactive control policies tend to control the suspension such that a more favourable compromise between resonant control and high-frequency harshness is achieved.

Semiactive suspension systems modulate the viscous damping coefficient of the shock absorber, based on road excitations through a controller. They do not add energy to the suspension system, thus consuming less energy and are less expensive to design. In fully active suspension systems, an external force is exerted in an appropriate direction based on control inputs. This requires additional energy to lift or pull the car body. In this regard semiactive suspensions have a constraint of not being able to exert a force opposite to the movement of body mass. In recent times though, research in semi-active suspensions has narrowed the performance gap between semiactive and fully active suspension systems.

The benchmarking of the main semiactive suspension control strategies is done in [2]. The research article in [3] enlists various control strategies that compare them for comfort and road handling. In [4] Mauricio and et. al. design a variable structure controller for semiactive suspension using the Bouc-Wen model for the MR damper. The relative performances for three control approaches are depicted. $\sigma 1$, $\sigma 2$ and $\sigma 3$ are the three approaches compared with the uncontrolled 2 degree of freedom (DOF) quarter car suspension system model. It is demonstrated that $\sigma 3$ is superior by 22%, taking normalized peak suspension deflection as the performance parameter. However, there is a trade-off with normalized peak sprung mass velocity being inferior by 45%. It is also demonstrated that even if the peak sprung mass velocities for all the three controllers is larger than the uncontrolled case for a single road hump input, there is a faster roll-off to zero.

In [5] M. Zapateiro and et. al. discuss semiactive suspension control with MR dampers using quantitative feedback theory. This control approach demonstrates the improvement over backstepping approach in [2]. In both cases the results are illustrated for a single road hump and random road disturbance. In [6] a controller is designed for semiactive damping using sliding mode control. The parameters of the suspension system are computed as rms values and in terms of suspension working space. The semiactive damping is compared with passive and active damping. A small reduction of 6% is observed in comparison with passive suspension system using this approach. The responses of the designed system are analyzed for vehicle speed of 20 m/s.

In [7] Talatahari and et. al. investigate the parameters involved in a Bouc-Wen model of MR damper. Various parameters are identified using fitting model response. They propose two additional parameters, with modifications in few of the simple Bouc-Wen model parameters to ensure maintainability for near zero velocities. This is the optimized approach which can be used in a real world scenario more appropriately. In [8] on-off skyhook control approach for semiactive damping is demonstrated for different frequencies of road excitations. The damping is parameterized using normalized rms values for vehicle body, suspension working space and road handling with respect to the body weight.

From the above research work it is observed that the variable structure control approach for semiactive suspension can be validated better by using a modified Bouc-Wen model[7] for MR damper. The parameters for comparing the three sigma controllers can be done using rms and suspension working space measures [6]. The parameters for the quarter car model in [4] are for aircraft landing gear. We propose to use the quarter car model parameters of [6]. The variable structure controllers are further explored for real time implementation in multiple phases. In the first phase we use a modified Bouc-Wen model and study the behaviour of these controllers for passenger car comfort and road handling. The system model for semiactive suspension using variable structure controller is analyzed for a single road hump and random road disturbance.

II. SYSTEM LEVEL MODEL

The system level model of the semiactive suspension system is shown in Fig. 1. It comprises of the 2DOF quarter car model within the tyre and suspension subsystem as shown in Fig. 2, the controller, MR damper and road excitation blocks.

Fig. 1 also depicts the changes we have proposed in modeling of the semiactive suspension system. The three changes depicted in comment blocks are; including the model of a quarter car for passenger car, using the modified Bouc-Wen model for the MR damper, and formulation of performance indices for validation of relative performance among the controllers. The performance indices used are computations of parameters of suspension system using rms peak and suspension working space. To improve accuracy of the computations for near zero velocities modified Bouc-Wen model is suitable.

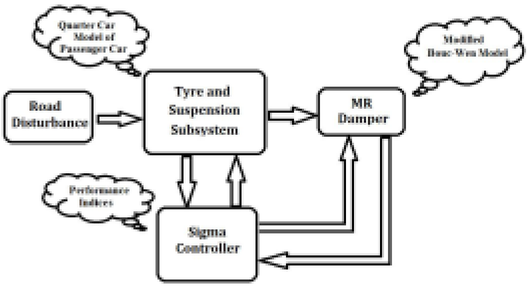

Fig.1 System model of a semiactive suspension system

The various parameters of the quarter car model are depicted in Fig. 2, and their correlation depicted thereafter. The tyre subsystem comprises of unsprung (wheel) mass mu and tyre stiffness kt. The suspension subsystem comprises of sprung (body) mass ms, stiffness of uncontrolled system ks and the damping force generated by the semiactive MR damper fmr. Applying the force balance equations to the model we get the following relations between these parameters.

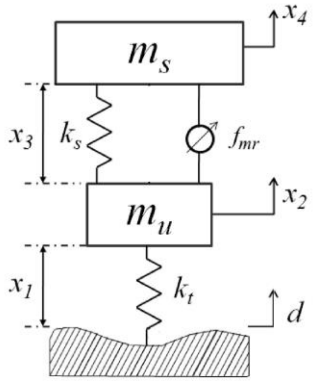

Fig.2 2DOF quarter car model of a semiactive suspension system

$$m_s x_4 + k_s \dot{x}_3 + f_{mr} = 0 \tag{1}$$

$$m_u \dot{x}_2 - k_s x_3 + k_1 x_1 - f_{mr} = 0 \tag{2}$$

Where, x1 is the tyre deflection, x2 is the unsprung mass velocity, x3 is the suspension deflection and x4 is the sprung mass velocity. Taking these as the state variables we can write the state-space representation as below.

$$\dot{x}_1 = x_2 - d \tag{3}$$

$$\dot{x}_2 = -\frac{k_t}{x_1} + \rho u \tag{4}$$

$$\dot{x}_3 = -x_2 + x_4 \tag{5}$$

$$\dot{x}_4 = -u \tag{6}$$

where, ρ = ms/mu , d is the road disturbance in m/s, and u is the acceleration input due to the damping subsystem and is given by,

$$u = \frac{1}{m_s}(k_s x_3 + f_{mr}) \tag{7}$$

The semiactive damping is achieved through MR damper as shown in Fig. 1, and is modeled here using the modified Bouc-Wen model [6] as shown in Fig. 3.

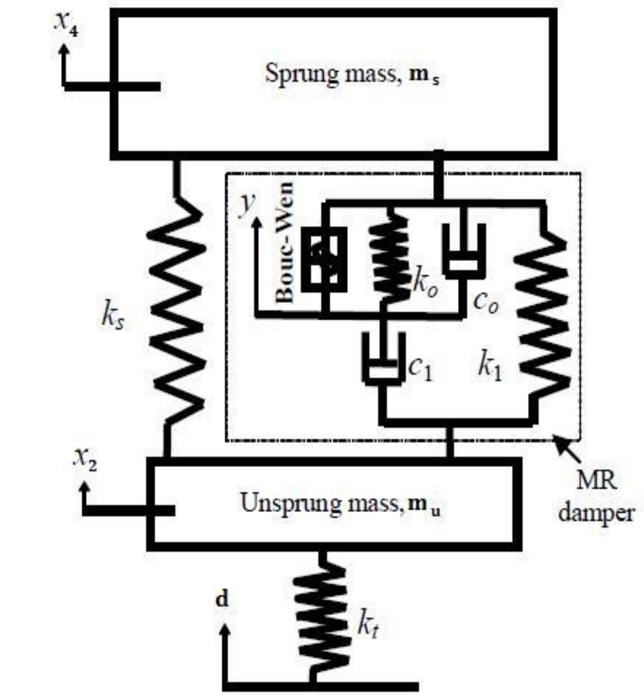

Fig.3 Modified Bouc-Wen model of semiactive suspension system

The model parameters are related as

$$f_{mr} = C_1(\dot{y} - x_2) + k_1(x_3 - x_0) \tag{8}$$

where C1 is the viscous damping coefficient which produces the system roll-off at low velocities and x0 is the initial deflection for the damper accumulator which is represented by the stiffness k1.

$$\dot{y} = \frac{1}{C_1 + C_0}\{\alpha Z + C_0 x_4 + C_1 x_2 + k_0(x_B - y)\} \tag{9}$$

where xB is the sprung mass (body) displacement, and Z is the Bouc-Wen variable given by

$$\dot{Z} = -\gamma|(x_4 - \dot{y})|Z|Z|^{n-1} - \beta(x_4 - \dot{y})|Z|^n + \delta(x_4 - \dot{y}) \tag{10}$$

α and δ are functions of the applied magnetic field and related to the height, width and slope of the pre-yield hysteresis loop. β, γ and n give the basic configuration of the hysteresis loop. The damper force fmr is related to the control voltage V through the following equations

$$\alpha = \alpha_a + \alpha_b V \tag{11}$$

$$C_1 = C_{1a} + C_{1b} V \tag{12}$$

$$C_0 = C_{0a} + C_{0b} V \tag{13}$$

The modified Bouc-Wen parameters are taken from [9] and shown in Table I.

The quarter car model parameters are suitably chosen for a passenger car from [10] instead of a flight landing gear as in [4] and shown in Table II. The controller chosen to implement semiactive damping using MR damper is the variable structure controller [4] which is a robust controller. The control signal is changed from 0V to 5V (Vmax) based on different control laws. Accordingly we have three sets of controllers as shown below.

$$\sigma_1: \quad v(x_2, x_4) = \frac{V_{max}}{2}[sgn(x_4 - x_2) + 1] \tag{14}$$

Parameter	Value
C0a	784 Ns/m
C0b	1,803 Ns/Vm
C1a	14,649 Ns/m
C1b	34,622 Ns/Vm
k0	3,610 N/m
k1	840 N/m
αa	12,441 N/m
αb	38,430 N/m
β	29,59,020 m^{-2}
γ	1,36,320 m^{-2}
δ	58
n	2
x0	0.0908 m

Table.1 MODIFIED BOUC-WEN MR DAMPER MODEL PARAMETERS

Parameter	Value
Sprung mass, ms	240 kg
Unsprung mass, mu	36 kg
Passive spring stiffness, ks	16,000 N/m
Tyre spring constant, kt	1,60,000 N/m

Table.2 MODIFIED BOUC-WEN MR DAMPER MODEL PARAMETERS

$$\sigma_2: \quad v(x_2, x_3, x_4) = \frac{V_{max}}{2}[sgn(sgn(x_4 - x_2) + x_3) + 1] \tag{15}$$

$$\sigma_3: \quad v(x_2, x_4) = \frac{V_{max}}{2}[sgn(r) + 1] \tag{16}$$

$$where \quad \frac{dr}{dt} = -100r|x_4 - x_2| - 10(x_4 - x_2) \tag{17}$$

III. EXPERIMENTAL RESULTS

The simulations for the three controllers implemented as per the strategies in equations 14-16 are done in Simulink. The variable step solvers are used for the simulation, and the simulation is run for 10 seconds. Two types of road excitations are analyzed for the semiactive suspension control. The system responses for a road hump model shown in Fig. 4 are the road hump disturbance (m/s), unsprung mass velocity and sprung mass velocity respectively. The responses for the three sigma controllers and uncontrolled case are depicted using colors as shown. The same analysis is repeated for random road disturbance case, and is depicted in Fig. 6. The system

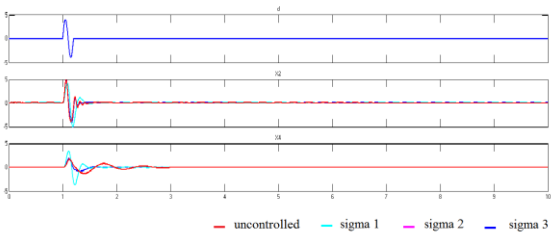

Fig.4 System responses for road hump disturbance

responses for a road hump model shown in Fig. 5 are the suspension deflection (m), tyre deflection (m) and sprung mass acceleration respectively. The same analysis for random road disturbance case is depicted in Fig. 7. The system responses

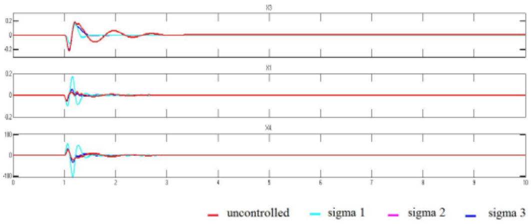

Fig.5 System responses for road hump disturbance

for a random road disturbance model are shown in Fig. 6 and Fig. 7. A random source with uniform noise and repeatable pattern is used to validate the results for multiple controllers, and the uncontrolled system. The relative performance of these controllers is done using the various measures shown in Table III. Here J1-J3, J7 are adapted from [4], J4-J6 from [6] and J8-J9 from [8]. The performance of the three controllers for road hump and random road disturbances are shown in Table IV and Table V with respect to the performance indices.

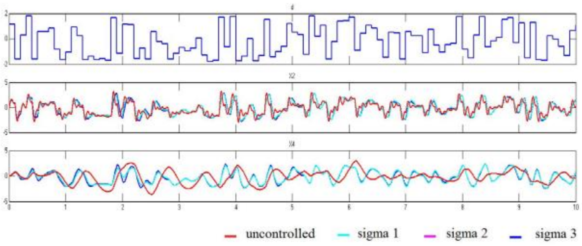

Fig.6 System responses for random road disturbance

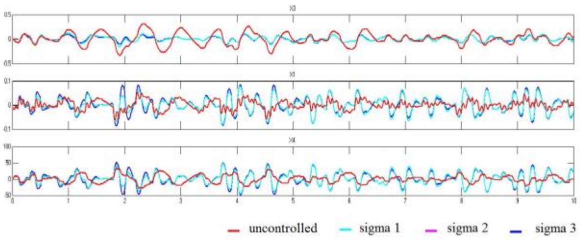

Fig.7 System responses for random road disturbance

Performance Index	*Definition*
$J_1 = \frac{max\|x_3(t)\|_{cont}}{max\|x_3(t)\|_{unct}}$	Norm. peak suspension deflection
$J_2 = \frac{max\|x_4(t)\|_{cont}}{max\|x_4(t)\|_{unct}}$	Norm. peak sprung mass velocity
$J_3 = \frac{max\|\dot{x}_4(t)\|_{cont}}{max\|\dot{x}_4(t)\|_{unct}}$	Norm. peak sprung mass acceleration
$J_4 = \frac{x_{3,rms,cont}}{x_{3,rms,unct}}$	Norm. rms suspension deflection
$J_5 = \frac{x_{4,rms,cont}}{x_{4,rms,unct}}$	Norm. rms sprung mass velocity
$J_6 = \frac{\dot{x}_{4,rms,cont}}{\dot{x}_{4,rms,unct}}$	Norm. rms sprung mass acceleration
$J_7 = \frac{max\|f_{mr}(t)\|}{w_s}$	Maximum control effort
$J_8 = \frac{1}{m_s g}\sqrt{\frac{1}{N}\sum_{i=1}^{n}[m_s\dot{x}_4(i)]^2}$	Norm. body acceleration
$J_9 = \frac{1}{m_s g}\sqrt{\frac{1}{N}\sum_{i=1}^{n}[k_s x_3(i)]^2}$	Norm. suspension working space

Table.3 PERFORMANCE INDICES FOR MEASUREMENT

IV. ANALYSIS

As can be seen from the system response to a single road hump excitation, all the three sigma controllers have lesser peak suspension deflection as compared to the uncontrolled case. The apparent drawback seems to be the peak sprung mass velocity and acceleration, but the system response tapers off quickly to zero for sigma 3 controller, which is not the case with uncontrolled system. The visual analysis is further strengthened by the numerical performance parameters which are listed from J1 to J9.

Index	σ_1	σ_2	σ_3
J_1	0.7367	0.7367	0.9265
J_1	0.7367	0.7367	0.9265
J_2	2.1938	2.1938	1.0751
J_3	3.8116	3.8116	1.2715
J_4	0.5855	0.5855	0.8639
J_5	1.7602	1.7602	0.8740
J_6	3.6753	3.6753	1.3022
J_7	10.2348	10.2348	3.2432
J_8	4.4232	4.4232	1.5672
J_9	0.3793	0.3793	0.5597

Table.4 PERFORMANCE INDICES FOR ROAD HUMP DISTURBANCE

Index	σ_1	σ_2	σ_3
J_1	0.4702	0.4702	0.3678
J_2	0.6548	0.6548	0.6794
J_3	1.4326	1.4326	1.5476
J_4	0.4093	0.4093	0.3752
J_5	0.7724	0.7724	0.7946
J_6	1.8410	1.8410	1.9964
J_7	4.6364	4.6364	5.0973
J_8	2.3306	2.3306	2.5273
J_9	0.3367	0.3367	0.3087

Table.5 PERFORMANCE INDICES FOR RANDOM ROAD DISTURBANCE

sigma 2 and 14% for sigma 3. The control effort required per unit sprung mass weight is lesser for sigma 3 as compared to sigma 1 and sigma 2. The controller design for sigma 3 is more complex as compared to sigma 1 & sigma 2 because of an additional feedback loop apart from the external MR damper feedback loop. Also the normalized body acceleration is slightly better for sigma 3 as compared to sigma 1 and sigma 2, measured by J8.

The analysis of the semiactive suspension system is also done for a random road disturbance to study the overall impact of the controllers for fast changing inputs. As can be seen from the system responses in Fig. 6 and Fig. 7 it is observed that all the three sigma controllers have lesser peak suspension deflection and peak sprung mass velocity as compared to the uncontrolled case. The peak sprung mass acceleration is inferior compared to uncontrolled case. The visual analysis is further strengthened by the numerical performance parameters which are listed from J1 to J9. J1 is 53% better for sigma 1 & sigma 2, while it is 63% better than the uncontrolled case for sigma 3. J2 is 35% better for sigma 1 & sigma 2, while it is 33% better than the uncontrolled case for sigma 3. The flip side is that J3 is inferior for all controllers. The rms value of suspension displacement is superior to the uncontrolled case by 60% for sigma 1 & sigma 2 and 63% for sigma 3. These results are in relative conformity with the system model simulated in [4] where the author has used the scaled values of the MR damper and tuned the system for the requirements of an aircraft landing gear. In both these cases the suspension working space was found to be minimal thus improving the ride comfort of the passenger.

Summarizing the experimental result analysis, peak suspension deflection is minimized using semiactive control. There is degradation within permissible range for peak sprung mass acceleration. The passenger comfort will be nevertheless better, if peak suspension deflection is less without compromising on road handling.

Using the system model illustrated above and the investigations of the results thereafter, reveals a feasible platform for rigorous validation for the study of relative performance of the semiactive suspension controllers. The performance indices reflect the behavior of the system for normalized parameters with respect to both uncontrolled system and body weight. The results obtained depict the feasibility of choosing the typical controller for real time embedded system implementation. The investigations also give a platform to replace the simulated road hump disturbance data with the actual road disturbance using an embedded processor platform.

The merit of the present system model is freezing the type of the suitable controller to implement in a real time scenario for a passenger car. This system model gives a way to identify the various computational needs with appropriate interfaces. Presently Mercedes car has the semiactive suspension system embedded in it. Our attempt is to bring this comfort to middle range car models with economic embedded solution or compact system on chip solution, which will be further studied.

V. CONCLUSIONS

The semiactive suspension system using MR dampers is modeled and measured for various performance indices for the three variable structure controllers. It has been observed that all the controllers are superior to the uncontrolled case, and can be explored for future implementations. The diversity in measuring the performance through various indices provides a deeper insight into the behaviour of these systems for road hump and random road disturbance. Sigma 3 controller is superior by 63% while sigma 1 and sigma 2 are superior by 53% when it comes to peak suspension deflection for a random road disturbance as compared to the uncontrolled case. It was observed that there is no trade-off with respect to peak suspension deflection and the time it takes to reach zero steady state value after a road disturbance. The performance indices are in general conformity to previous designs, but need to be tested for a real road disturbance case for a passenger car under different road conditions and multiple driving speeds. The variations in some indices can be attributed to the scaled versions of the MR damper parameters used in [4] for the aircraft landing gear, as compared to the unsealed parameters of MR damper here. The controllers are to be implemented through the model based design and development approach using an embedded processor platform.

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