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PID Based Simulation of Semiactive Suspension System Using MR Damper

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Abstract: *In this paper, a brief introduction to vehicle primary suspension system is presented along with a semiactive suspension system with Bingham model for MR damper. The primary function of the suspension system is to isolate the forces transmitted by external excitation. The heart of a semiactive suspension system is the controllable damper. In this paper, the ride and handling performance of a vehicle with passive suspension system is compared to semiactive suspension system. The vehicle suspension system is modeled as a two degree of freedom quarter car model. Simulation is carried out using MATLAB/Simulink. The behaviour of suspension system under continuous definite operating condition is summerised, without compromising on ride comfort. The simulation model of PID Controller has been developed for semi active suspension. Results are being quantified using vertical sprung mass acceleration and comparing with standards of ISO 2631-1:1997 Ride Comfort Chart.*
Keywords: *MR damper, Passive, Ride comfort, Bingham model, PID controller*

I. Introduction

Suspension is the system of springs, shock absorbers and linkages that connects a vehicle to its wheels and allows relative motion between the two. Isolation from the forces transmitted by external excitation is the fundamental task of any suspension system. Basic tasks of any automotive suspension on a vehicle are typically to isolate a car body from road disturbances, to keep good road holding, and to support the vehicle static weight [1]. Springs and dampers are two basic types of elements in conventional suspension systems. The role of the spring in a vehicle's suspension system is to support the static weight of the vehicle [2]. The role of the damper is to dissipate vibrational energy and control the input from the road that is transmitted to the vehicle. The basic function and form of a suspension is the same regardless of the type of vehicle or suspension. Vehicle Primary Suspensions are divided into passive, active and semi-active systems [3]. Primary suspension designates suspension components connecting the axle and wheel assemblies of a vehicle to the frame of the vehicle. This is in contrast to the suspension components connecting the frame and body of the vehicle, or those components located directly at the vehicle's seat, commonly called the secondary suspension.

In Semiactive suspension system, the conventional spring element is retained, but the damper is replaced with a controllable damper. Magnetorheological (MR) damper is a kind of semiactive device. A wide range of Magneto-rheological (MR) fluid based dampers are currently being explored for their potential implementation in various systems, such as vibration control devices and suspension system. The main function of vehicle suspension systems is to minimize the vertical acceleration transmitted to passengers to provide ride comfort and to maintain the tire road contact to provide holding characteristics and to keep suspension travel small [4]. In this paper, performance of semiactive suspension model (2DOF) based on the Bingham model subjected to random road excitation is compared with passive suspension system. This semiactive vehicle suspension shows improvement over passive vehicle suspension. Advantages of MR Damper is having variable damping coefficient and needs no energy source as such needed for active suspension [5].

II. Mathematical Modelling Of System

2.1 Quarter Car Model with Passive Suspension System

To simulate the performance of vehicle subjected to road excitation the passive quarter car model as shown in Fig.1 is taken for study [6]. The equation 1 and 2 represents mathematical representation of quarter car model with passive suspension. Where m_s= Sprung mass, m_u= unsprung mass, z_s = Displacement of sprung mass, z_u= Displacement of unsprung mass, k_s= Spring stiffness, k_u= tire stiffness, C_s= Damping coefficient of dashpot, c_u= Tire damping coefficient. The equations of motion for this linear model is-

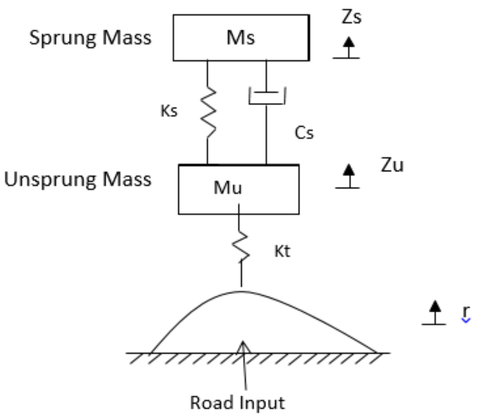

Fig.1: Quarter car model with passive suspension [6].

$$m_s\ddot{z}_s + c_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) = 0 \quad \ldots (1)$$

$$m_u\ddot{z}_u + c_s(\dot{z}_u - \dot{z}_s) + k_s(z_u - z_s) + c_u\dot{z}_u + k_u z_u = k_u r + c_u\dot{r} \quad \ldots (2)$$

1.2 Quarter Car Model with Semiactive Suspension System

The proposed system is 2DOF quarter car vehicle with a MR damper. The behaviour of the damper are modeled with the Bingham model. The Bingham model contains the nonlinear behavior of a viscous fluid going through an orifice [7].

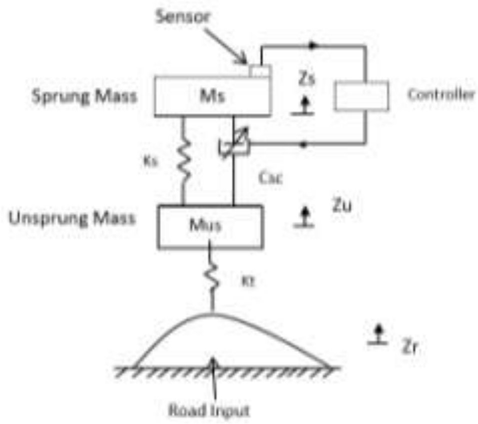

Fig.2. Quarter Car model with semi-active suspension [7].

1.3 Bingham model

The idealization of the visco-plastic MR damper model presented in uses similarities in the rheological behavior of Electro-rheological and Magneto-rheological fluids and the similar techniques in the modelling of Electro-rheological dampers.The model in Fig.3 shows Bingham mechanical model, the Coulomb friction element fc and dashpot C_0 are placed parallel. The damping force F can be expressed with Accordance to Bingham's MR damper model, for non-zero piston velocities y [8].

$$F = f_c\, sgn(\dot{y}) + c_o\dot{y} + f_o \quad \ldots (3)$$

The equation 3, represents mathematical representation of quarter car model with semi-active suspension. Where, fc- frictional force, Co - viscous damping parameter; f_0 - force due to the presence of the accumulator. Fig.4 shows the semi active quarter car model with MR damper and controller. Ms - one quarter of sprung mass; mu - unsprung mass (wheel, damper and spring etc.) , xs and xt - mass displacement; q-road disturbances, kt-tire stiffness; ks - spring between wheel and chassis [9].

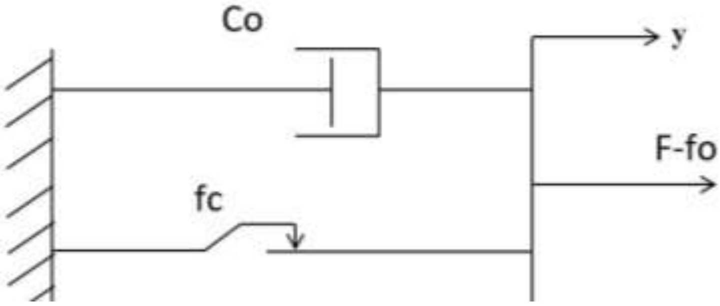

Fig.3- Bingham model of MR Damper [8].

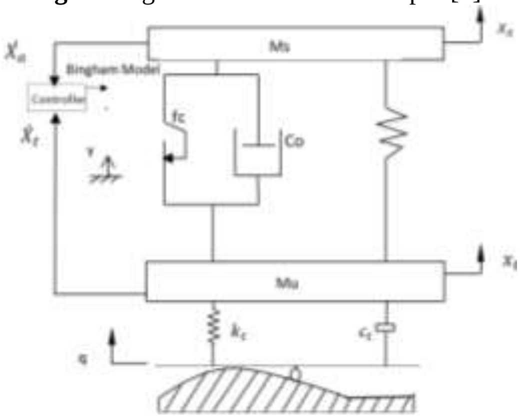

Fig.4 Semi active quarter car model with MR damper, PID Controller [9]

For the controller, (zs) and (zt) represents absolute displacement of sprung mass and unsprung mass respectively. Controller generates the current zs in the MR damper and changes the force F of semi active suspension system [10]. The motion equations of the car body and wheel of this model areas follows

$$m_s\ddot{z}_u + c_s(\dot{z}_u - \dot{z}_s) + k_s(z_u - z_s) + c_u\dot{z}_u + k_u z_u = -U_c + k_u r + c_u\dot{r}$$

…………………………………… (4)

Table 1. Quarter Car Model Parameters [10]

Parameter name	Parameter notation	Parameter value
Sprung mass	m_s	2500 kg
Un-sprung mass	m_s	320 kg
Stiffness of suspension	k_s	80000 N/m
Stiffness of Un-sprung mass	k_u	500000 N/m
Damping co efficient of Sprung mass	c_s	320 Ns/m
Damping Co efficient of unsprung mass	c_u	15020 Ns/m

Table 2. Bingham Model Parameters[10]

Parameter name	Parameter notation	Parameter value
Damping coefficient in Bingham Model	c_o	320 Ns/m
Offset Force	F_o	10 N
Frictional Force	F_c	100 N
Stiffness of an elastic components	K_o	300 N/m
Form factor	d	10

III. Simulation Analysis

MATLAB/Simulink tool is used to simulate for passive and semiactive with PID controller suspension system. The Simulink model is prepared from the equation no.4 and for simulation the input parameter is from Table 1 and Table 2. Sprung mass acceleration is measure of the ride comfort for road excitations with bump frequency of 50 rpm.

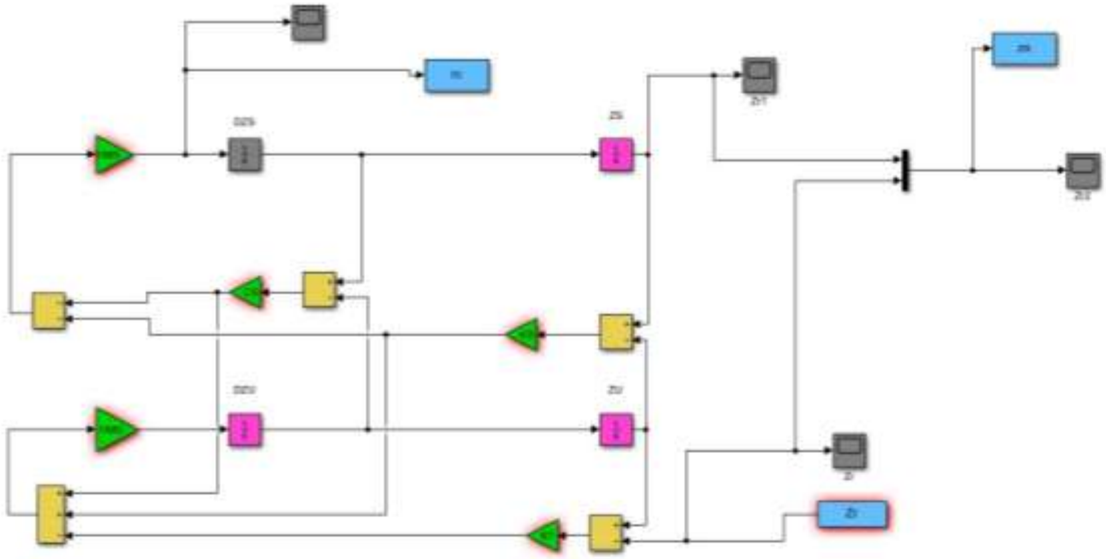

Fig.5- Simulink Model of Quarter Car Test Rig.

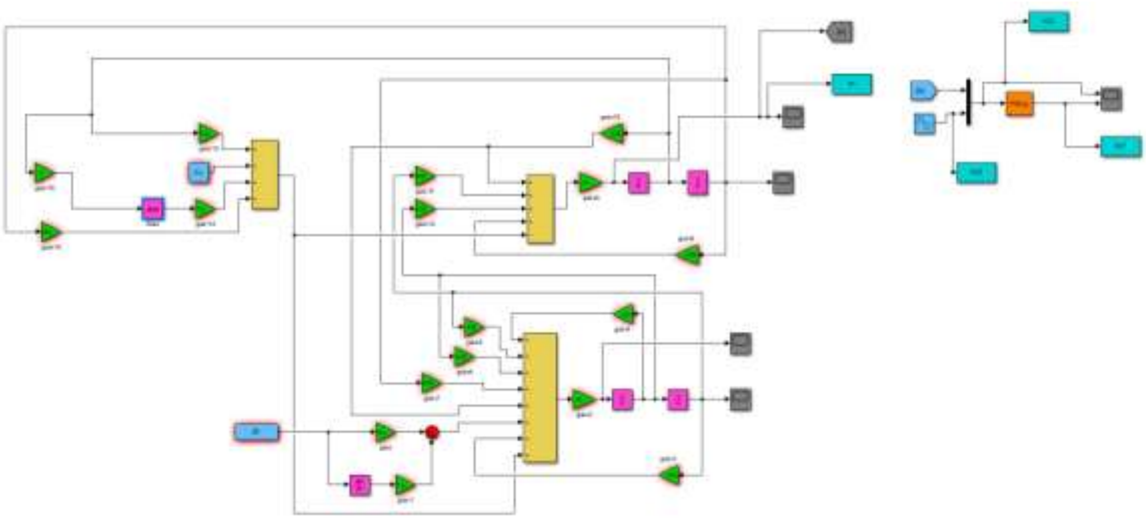

Fig.6- Bingham Model of Semi-Active Suspension of Quarter Car With Using PID Controller

Random road excitation

For implementing more accurate input to the Simulink program the displacement diagram is drawn in CATIA software, so that the details of the bump profile e= 21mm is enhanced into the input. A suitable Matlab syntax program is created and stored in the workbench of the Simulink so that it can be called back as an input whenever required.

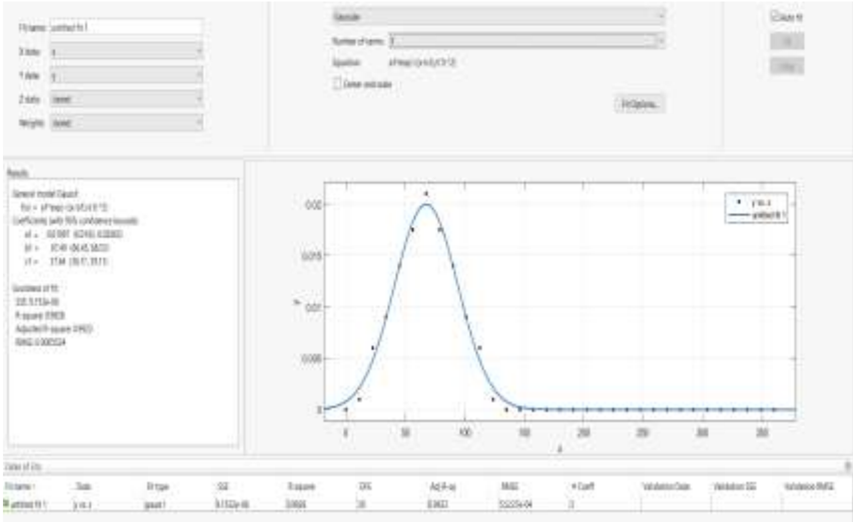

Fig.7- Displacement Curve

Simulation Results

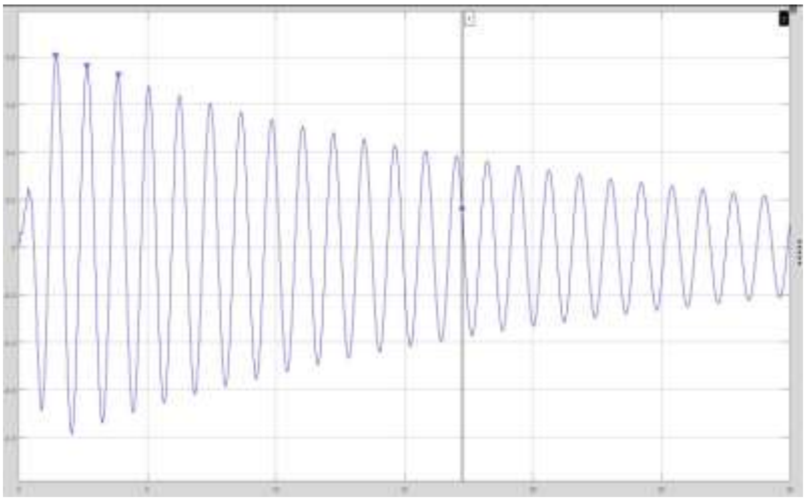

Fig. 8 Sprung mass Acceleration vs time (sec) of Passive Suspension

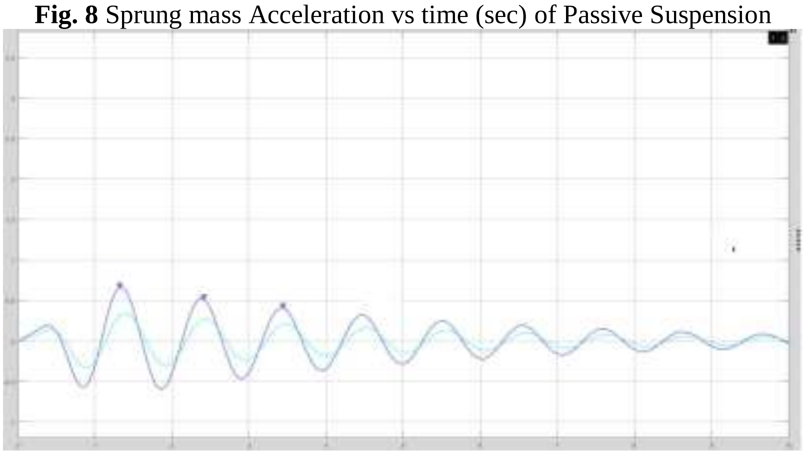

Fig. 9. Sprung mass Acceleration of Semi-active suspension with PID and without PID Vs Time (sec)

Table 3. Result

Suspension system	Maximum sprung mass acceleration (m/s^2)
Passive suspension system	0.8
Semi-active suspension system	0.65
Semi-active suspension with PID controller	0.3

IV. Discussion Of Results

The percentage variation in maximum sprung mass acceleration of semi-active suspension system based on Bingham model is 18.75% from passive suspension system. Also the percentage variation in maximum sprung mass acceleration of semi-active suspension with PID controller is 53.85% from semi-active suspension system and 62.5% from passive suspension system

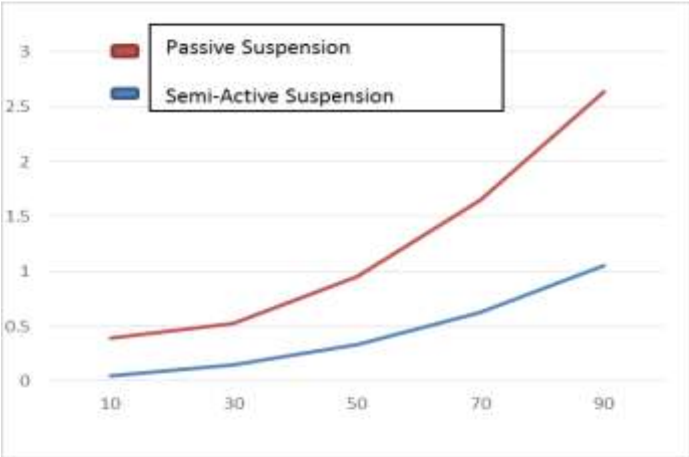

Fig. 10. Comparison of Semi-Active with PID controller and Passive suspension system for Ride Comfort Characteristics, graph represents vertical sprung mass acceleration vs excitation bump frequency in RPM.The sprung mass acceleration is categorized in accordance to ISO 2631 standard as shown in given table 4.

Table 4**. ISO 2631-1:1997 Ride Comfort Chart [12]**

Sprung mass acceleration	Ride Condition
Less than 0.315 m/s^2	Not uncomfortable
0.315 m/s^2 to 0.63 m/s^2	A little uncomfortable
0.5 m/s^2 to 1 m/s^2	Fairly uncomfortable
0.8 m/s^2 to 1.6 m/s^2	Uncomfortable
1.25 m/s^2 to 2.5 m/s^2	Very uncomfortable
Greater than 2 m/s^2	Extremely uncomfortable

V. Conclusion

The passive suspension system, semiactive suspension system with PID and MR damper (Bingham Model) is simulated. From the simulation results it is observed that the sprung mass peak acceleration for Passive is $0.8m/s^2$, and semi-active suspension is $0.65m/s^2$ and semiactive suspension with PID controller is $0.3m/s^2$. The simulation results shows that semiactive suspension system with PID Bingham model gives lower value of maximum sprung mass acceleration for road excitation. Hence suspension model with semi-active suspension with PID controller provides good passenger comfort and vehicle stability than passive suspension system.Hence, the semiactive suspension system gives the ride comfort as non- uncomfortable i.e. under the comfort level as per IS 2631 standard.

Conflict of interest the authors declare that there is no conflict of interests regarding the publication of this paper.

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