Model development and simulation of vehicle suspension system with magneto-rheological damper

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Abstract. A vehicle suspension system is designed to maintain directional control (road holding) during manoeuvring or braking while supporting the vehicle’s weight and provide stability (handling). The structure of a suspension system consists of parts connecting the axle to wheel assembly and the chassis of an automobile, thus supporting engine, transmission system and vehicle load. Suspension system components consist of dampening devices, springs, steering knuckles, ball joints and spindles or axles. It could be designed according to a passive, semi-active or active mode of working. For evaluation, this assembly could be modelled as a spring-mass-damper system. The semi-active suspension system has been modelled with a magneto-rheological damper following the Bingham plastic theory. In this paper, the performance of a passive and a semi-active suspension of a quarter car model are compared by MATLAB simulation. Thus, a better suspension system is found out by simulating with different road conditions.

1. Introduction

The purpose of a car suspension system is to keep the vehicle in touch with the ground while allowing the driver to guide the tyres firmly. The suspension system governs three dynamic motions of a vehicle, namely roll, pitch and yaw. The working principle of a suspension consists of force absorption and dissipation using springs, dampers and struts. As a vehicle takes shock loading at the wheels, the spring stores the energy during its compression. The quantity of energy that a spring can store is determined by a variety of parameters, including the spring’s material and the spring coefficient. The spring transfers the stored energy to the damper or shock absorber which dissipates this energy. The damper is made of a cylinder and piston assembly with small orifices in the piston and is filled with pressurized oil. When energy from the spring is transferred to the damper, the piston travels through pressurized oil, expending the energy. The viscosity of oil in the damper is an important factor in deciding the shock absorption capacity of the suspension system.

The whole vehicle weight is divided into two parts, called sprung mass (the vehicle mass supported by the spring/damper) and unsprung mass (not supported by spring/damper). The unsprung to sprung mass ratio should be maintained as low as possible for better vehicle dynamics. Suspension geometries also play a major role in vehicle dynamics. Double wishbone (Short long arm suspension) is a very common geometry used in commercial vehicles, consisting of a short (upper) and a long arm (lower) connecting the chassis and the knuckle. McPherson struts are also a very widely used geometry, consisting of a single wishbone control arm connecting the knuckle to the chassis, and a strut connecting the wishbone to the chassis. Cars with solid axle suspension are used in transporters and big vehicles, consisting of trailing arm suspension, leaf springs, etc. The relative amount of suspension travel for a given wheel travel is a significant parameter in the design of the suspension. It should be set according to the vehicle needs and its applications. The more the damper travel, the more
the wheel travels over the suspension. For the suspension to work properly, the wheel alignments, namely camber, caster and toe also needed to be studied and set accordingly [1].

A vehicle suspension system designed with elements of fixed performance characteristics is called the passive suspension system, whose design is generally a compromise of comfort against stability or vice versa [2]. For semi-active suspension system the damping effect is adjustable through a controller along with couple of sensors. Controller determines necessary control force depending on a pre-defined control strategy, and it operates automatically in response to a signal from sensors monitoring suspension parameters [3, 4]. On the other hand, the active suspension system makes use of an actuator and controller with significant energy input for producing the control force. Semi-active suspensions have achieved widespread acceptance in automobiles because to their greater reliability, reduced cost, and equivalent performance [5].

The variation of damping effect is achieved by using a magneto-rheological (MR) damper in which the viscosity of damping fluid changes in tune with the magnetic field acting on the fluid and is a reversible process [6]. MR fluids consist of a dispersion of magnetizable particles in a viscous carrier fluid (usually oil) and it changes from a thick fluid to a semi-solid state within a few milliseconds depending on the intensity of magnetic field applied on it [7]. Commercially available MR dampers can be characterized by experimentation and used appropriately in vehicle suspension systems [8, 9]. The non-linear parameters and characteristics of some materials like MR fluids are important for effective control of such systems as the shock absorber and dampers.

Experimental studies to characterize the response of commercial MR dampers were useful to model their hysteresis behaviour, following Bingham and Boc-Wen approach [10]. Sulaymon (2016) has reviewed the mathematical formulations and numerical simulating of hysteresis nature of MR damper, by modelling it according to Dahl, Bingham, Bouc-Wen and LuGre models, developed in MATLAB®/Simulink®. The models were evaluated for a quarter-car model alongside Golden Car parameters. It was concluded that the Bingham model performed better than other models for constant value excitations (step input) [11]. Suspension models based on Golden car parameters are known to produce a good correlation for the full spectrum of vehicles in USA [12]. The performance of semi-active suspension for a quarter car model was examined in MATLAB utilising a PID controller. The vehicle handling effectiveness and ride comfort were assessed by measurement of the car body displacement and wheel deflection and used as feedback by the PID controller to enhance the performance of the suspension [13]. Metered et al. suggested a suspension system that includes a controller to determine the intended damping force using a PID controller tuned by Particle Swarm Optimization technique. Further, a magnetorheological damper was fabricated using carbonyl iron powder and Silicone oil, and calibrated by establishing the command voltage required to keep track of the desired damping force [14]. Singh et al. used evolutionary computing technique to predict the current value in order to adjust the damper force for the given input working conditions and maintain the driver acceleration [15]. Sastry et al. conducted experimental evaluation of the behaviour of the MR damper to validate the Spencer model. The Spencer model of MR damper was then used to simulate a semi-active suspension system for quarter car and half car models. The results show that the displacement and acceleration in the car body were reduced by 30% and 25% respectively, for a quarter car model [16]. The performance of the MR suspension system with respect to ride comfort was assessed using finite element analysis, considering a half-vehicle model built on a variable structure control approach and it was shown to outperform the passive suspension system [17]. In different studies, it has been proven that MR fluid is the most suitable material in designing vibration dampers and shock absorbers [18-20]. The performance of a semi-active seat suspension for a quarter car integrated with a human model was implemented in MATLAB and compared with that of passive seat suspension. Different road profiles such as road bump and ISO road profile were used to run the simulation and it was found that the semi-active seat suspension with MR damper can significantly attenuate the vibration transmitted to the human head [21].

2. Methodology
The equations of motion describing a passive suspension system of a quarter car model are written considering a physics model of the suspension elements and used for simulating in MATLAB
SIMULINK. Following the same procedure, a semiactive suspension system is modelled, which incorporates the Bingham model of MR damper and then the performance of the two are compared. The better suspension system is found out by MATLAB simulation to evaluate the chassis displacement under step input and impulse input conditions relating to the road disturbance. The step input captures the model response to a roadblock or a pothole while the impulse input corresponds to a road bump. The result is plotted on the graph with car displacement in the Y-axis and time in the X-axis. The MR damper parameters used for reference in this study is RD-1005-3 (Lord Corporation). It can give a force variation of 500 to 1000N in response to magnetizing current variation from 0-1A, with a stroke length of 5mm and frequency of 0.5 to 1.5Hz sinusoidal excitation [22].

3. Mathematical Model of Quarter Car Suspension
A 2DOF quarter car model is considered for the mathematical modelling and simulation of the vehicle suspension system. Assuming that the vehicle load is evenly distributed to all the four wheels of the vehicle, only one-quarter of the total vehicle mass is borne by a wheel. We know that the suspension system is an assembly of the elements constituting the sprung mass (M_s) and unsprung mass (M_u). The sprung mass is characterized by its stiffness (K_s) and damping coefficient (C_s) and the unsprung mass is also characterized by its stiffness (K_u) and damping coefficient (C_u). Even though the damping effect in the unsprung mass is considered insignificant in suspension models, still it is considered that the tyre gives a damping effect. The road disturbance \( r(t) \) acts on the unsprung mass and produces a vertical acceleration, which is transmitted to the springs.

3.1 Passive Suspension System Model
The physical model of passive suspension of a quarter car is shown in figure 1. The equations of motion (vertical) can be thus written, considering that the suspension is disturbed only by the road roughness during its traction. According to the model in figure 1, the sprung mass and unsprung mass are supported through the spring and damper. In a passive suspension, these elements are designed with a fixed value spring stiffness and damping coefficient satisfying the desired operating requirements. Once designed the suspension geometry and other parameters are not altered. As the vehicle moves along the road, the wheel picks up the road undulations, which is represented by \( r(t) \), as a function of time. The unsprung mass reacts to the road disturbance with a vertical acceleration (\( z_u'' \)) and transmits the same to the sprung mass, resulting in its acceleration (\( z_s'' \)).

![Figure 1: Phenomenological model of passive suspension.](image)

The mathematical model of passive suspension is derived referring to figure 1 and Newton’s law of motion, as follows,
Representing the equation (1) and (2) in Simulink we get the passive suspension model in the form of MATLAB block diagram that is using the blocks such as Add, Gain, Input, Scope, etc. as shown in figure 2. The value of suspension parameters are input and stored in the model as shown in Table 1. The suspension parameter data for the simulations are chosen to match the ‘golden car’ parameters [11, 12]. Simulation of the passive suspension system is carried out to find its response to a step input and impulse input; the relative displacement of the car body for a period of 10 sec is recorded.

**Table 1. Vehicle suspension parameters and their values**

| Parameter                              | symbol | Parameter values used in suspension model |
|----------------------------------------|--------|------------------------------------------|
| Sprung mass                            | $M_s$  | 500 Kg                                   |
| Unsprung mass                          | $M_u$  | 70 Kg                                    |
| Spring stiffness (sprung mass)         | $K_s$  | 28500 N/m                                 |
| Spring stiffness (unsprung mass)       | $K_u$  | 293900 N/m                                |
| Damping coefficient (sprung mass)      | $C_s$  | 2700 Ns/m                                 |
| Damping coefficient (unsprung mass)    | $C_u$  | 0                                        |
| Displacement (sprung mass)             | $z_s$  |                                          |
| Displacement (unsprung mass)           | $z_u$  |                                          |
| Force generated by controller          | $U_c$  |                                          |
| Road roughness / (disturbance)         | $r(t)$ |                                          |

3.2. Semi-active Suspension with Bingham Model

As opposed to a passive suspension system, the semi-active suspension is designed to produce a variable damping effect when subject to road disturbance. This is brought about by introducing a damper functioning with magnetorheological fluid, whose characteristics can be controlled through an
external factor in relation to the road disturbance. Considering the physical model of semi-active suspension in figure 3, a variable control force \( U_c \) is introduced by the MR damper in response to the magnitude of road roughness induced vertical displacement and velocity of the vehicle. In this model, for \( U_c \), we use the Bingham model to produce the variable damping force.

Thus, expressing the equations of motion based on Newton’s laws for semi-active suspension system,

\[
M_s \ddot{z}_s + C_s (\dot{z}_s - \dot{z}_u) + K_s (z_s - z_u) = U_c
\]  
(3)

\[
M_u \ddot{z}_u + C_s (\dot{z}_u - \dot{z}_s) + K_s (z_u - z_s) + C_u \dot{z}_u + K_u z_u = -U_c + K_u r + C_u \dot{r}
\]  
(4)

![Figure 3: Phenomenological model of semi-active suspension.](image)

![Figure 4: Bingham plastic model of magneto-rheological damping [22].](image)

The Bingham model of MR damper is represented in figure 4, as proposed by [22] The Bingham plastic model was proposed to simulate and identify the optimal parameters of MR fluids. The control force \( U_c \) of semi-active suspension system is set to be equal to \( F_{mr} \) of the Bingham model, and is given by the equation,

\[
F_{mr} = F_c \text{ sgn}(\dot{z}) + C_0 \dot{z} + f_0
\]  
(5)

Here, \( F_{mr} \) is the force generated by the Bingham model, \( F_c \) is the friction force=100N, \( C_0 \) is the damping coefficient =320Ns/m, \( f_0 \) is the offset force (constant force) =10N and \( z \) is the piston relative displacement [10]. The Signum function will take care of the direction of friction force. In this model, \( z \) and its functions corresponds to the displacement of the sprung mass. The properties of the MR damper used in this work is in reference to the MR damper RD-1005-3 (Lord Corporation). Now with the mathematical equations of motion for semi-active suspension and the Bingham model of
damper, the two are integrated to create a Simulink block diagram for the simulation study of semi-active suspension, as shown in figure 5. In the part of the block diagram showing the Bingham model of MR damper, the velocity of sprung mass $\dot{z}_s$ is the input that directs the frictional force $F_c$ with the Signum function. As it can be seen in the block diagram, the road disturbance is given as the input to the unsprung mass. The simulation of semi-active suspension is carried out to find its response to a unit step input and the relative displacement of the car body for a period of 10 sec is recorded.

Figure 5: MATLAB model of semi-active suspension system with magneto rheological damping.

4. Results and Discussions
A thorough analysis of passive suspension system was performed with the MATLAB model and using the vehicle suspension parameters as described in Section 2. The simulation result is shown below in figure 6 for a step input of 0.1m magnitude. The graph shows the variation of body displacement over a period of 10 sec of simulation time. From the graph the maximum displacement is 40mm (0.04m), followed by another peaking point of 5mm and the settling time is around 2 sec.

For a vehicle with passive suspension system moving across a speed breaker, the response of MATLAB model is shown in figure 7. The road disturbance resulting out of a speed breaker is simulated using a pulse input in the MATLAB suspension model. In order to match with a typical speed breaker on road, the pulse input is defined with a step magnitude of 0.1m and pulse duration of 25 ms. The pulse duration was estimated at this value by considering a vehicle speed of 10km/hr and a speed breaker width of 0.4m to be crossed over by the vehicle. In this regard, operating speed of vehicles crossing speed breakers analyzed by Mohanty, et al., were useful [23]. In figure 7, the graph shows a sudden jerk of over 50mm (0.05m) within 0.2 sec but subsequently the jerky motion is averaged out and the vehicle body displacements reached a steady condition only after 1.2 sec.
The simulation result for the semi-active suspension with Bingham model is shown in figure 8 for a step input of 0.1m magnitude and in figure 9 for a pulse input of 0.1m magnitude. The graph in figure 8 shows the variation of body displacement over a period of 10 sec of simulation time. In this graph, there is a maximum vertical displacement of vehicle body up to 11mm and 1 sec of settling time. Oscillations of vehicle displacement are significantly reduced, though for the entire period of study, there is low order fluctuation in the vertical axis showing a smoothening of road disturbance by the MR damping.

Figure 9 shows the response of MATLAB model of semi-active suspension system for a pulse input to simulate the vehicle crossing a road bump. In order to match with a typical speed breaker on the road, the pulse input is defined with step magnitude of 0.1m and pulse duration of 25 ms. In figure 9, the graph shows a sudden jerk with a vertical displacement of 0.04m of the vehicle body as it comes across the speed breaker. Repeated jerks of reduced magnitude (less than 0.015m), is noted up to a duration of 0.3 sec followed by steady condition of vehicle ride after 0.9 sec, showing it is no better response than the passive suspension. It could be due to the pulse duration being very short compared to the response time of MR Fluid. Based on the MATLAB simulation results presented in figures 6-9, it is seen that the semi-active suspension with MR damper in general, provides better ride comfort.
5. Conclusions
The physical models of passive and semi-active vehicle suspension are analyzed in order to understand their behaviour and express the same by a mathematical model. The semi-active suspension model was integrated with the Bingham model of MR damping in MATLAB. The MATLAB models of passive and semi-active suspension are thus developed and simulation was performed for a step and pulse inputs representing different road disturbances. The vertical displacement of vehicle body is plotted for a simulation time of 10 sec for the four simulated conditions. The result shows that the semi-active suspension gives reduced vehicle body displacement and lesser settling time, compared to passive suspension.

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