Vehicle ride performance with semi-active suspension system using modified skyhook algorithm and current generator model

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Abstract: This paper presents the simulation of a semi-active suspension system by using designed parameter model of magnetorheological (MR) damper. The design is based on the performance of the original equipment shock absorber of a passenger vehicle. A 7DOF of vehicle ride model was developed and validated in order to study the performance of a passive suspension system and the designed semi active suspension system. A controller known as modified skyhook algorithm and current generator model was used in the semi-active suspension system. The simulation results show that the semi active suspension system give significant improvement on vehicle’s ride comfort.
Keywords: magnetorheological damper; ride performance; modified skyhook; semi active suspension system.


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1 Introduction

The design process, especially in engineering field, may require compromises in satisfying the design specification. The design of automotive suspensions falls into this category. The familiar trade-off between ride comfort and vehicle stability has been the driving force for advancement in the design of automotive suspensions. Though subjective, ride comfort can be quantified by the amount of energy transmitted through the suspension into the passenger compartment (sprung mass).

A typical vehicle suspension is made up of two components: a spring and a damper. The spring is chosen based solely on the weight of the vehicle, while the damper is the component that defines the suspension’s placement on the compromise curve. Depending on the type of vehicle, a damper is chosen to optimise vehicle performance. Ideally, the damper should isolate passengers from low-frequency road disturbances and absorb high-frequency road disturbances. Passengers are best isolated from low-frequency disturbances when the damping is high. However, high damping provides poor high frequency absorption. Conversely, when the damping is low, the damper offers sufficient high-frequency absorption, at the expense of low-frequency isolation. The need to reduce the effects of this compromise has given rise to several new advancements in automotive suspensions. The simplest, and the most common, type of suspension system is passive in the sense that no external sources of energy are required.

Recently, there has been significant research activity in a new class of so-called advanced suspension system. Advanced suspensions can be divided into four categories: fully active, active, slow active and semi-active.

A fully active suspension uses powered actuators to replace the conventional spring and damper arrangement. This system operates over a wide range of frequencies and attempts to control the motion of both the vehicle body and the wheels (Sharp and Hassan, 1986). The bandwidth and power consumption requirements are severe and the hardware costs are significant; so, fully active systems are only feasible for special high performance applications.

A semi-active suspension typically consists of controllable damper and passive spring. Such a system can only dissipate energy, in contrast with fully active and slow active systems, which can supply energy. The performance of a semi-active suspension system typically lies between the active and passive systems.

Many simulation studies on the performance of semi-active suspension system have been reported. Most of the simulation studies of semi-active suspension systems are based on the skyhook control theory (Sharp and Hassan, 1987; Alleyne and Hedrick, 1992; Satoh et al., 1990; Sunwoo and Cheok, 1990; Wilkinson, 1994; Leighton and Pullen, 1994). A quarter car model to simulate the behaviour of a semi-active suspension system has been reported by Sharp and Hassan, 1987; Alleyne and Hedrick, 1992; Satoh et al., 1990; Sunwoo and Cheok, 1990; Wilkinson, 1994; Leighton and Pullen, 1994, while a full vehicle model has been reported by Wilkinson (1994) and Leighton and Pullen (1994). A semi-active and an active suspension system, based on linear optimal control theory, have been studied by Thompson (1976). Thompson (1984) and Wilson et al. (1986). The state feedback controller was introduced by ElMadany and Abdul Jabbar (1991).
The purpose of this paper is to study the performance of a semi-active suspension system using the designed MR damper (based on the Original Equipment (OE) shock absorber performance) with the modified skyhook control algorithm used as the controller to reject road disturbances. The development of a current generator model, which is the main contribution in this study, has enabled the designed MR damper to track and produce the damping force as estimated by the control algorithm. The use of 7DOF vehicle ride model in this study will enable all the responses in the vehicle body’s vertical, rolling and pitching motion to be observed and studied.

This paper consists of seven sections. Section 1 is the introduction, and describes the types of suspension and some of the research work done in improving vehicle’s ride comfort. Sections 2 and 3 describe the mathematical equations of 7DOF vehicle model and MR damper model, respectively. Section 4 describes the current generator model that is used to supply the required current to the MR damper model in order to generate the required damping force. Section 5 explains the modified skyhook controller that is used in rejecting the road disturbance. Section 6 discusses the results of the simulation and the last section presents the conclusion of the studies. Section 7 presents the conclusion.

2 Seven degrees of freedom of full car modelling and model validations

In this study, a ride model is derived based on the work done by Ikenaga et al. (1999, 2000) and Hudha et al. (2003). The model consists of seven degrees of freedom, namely roll, pitch, bounce and vertical motion of each four wheels. Figure 1 shows the vehicle model while Figures 2 and 3 show the side and front view of the model, respectively. A full derivation of the full vehicle model for a ride performance study will make it easy to build a ride model using MATLAB/SIMULINK block diagrams. The results gained from the simulations are then compared with the experimental results in order to validate the model. The purpose of this validation is to make sure that the simulation model of the vehicle can be further used to investigate on the performance of semi-active system.

Figure 1 7DOF full car model
Based on the 7DOF ride model in Figures 1–3, the displacements of the sprung masses are given by:

\[ Z_{sf} = Z_b + \frac{a}{2} \theta - \frac{b}{2} \alpha \]  
(1)

\[ Z_{sr} = Z_b - \frac{a}{2} \theta - \frac{b}{2} \alpha \]  
(2)

\[ Z_{ol} = Z_b + \frac{a}{2} \theta + \frac{b}{2} \alpha \]  
(3)

\[ Z_{or} = Z_b - \frac{a}{2} \theta + \frac{b}{2} \alpha \]  
(4)

where \( Z_{sj} \) is the total sprung mass displacement (\( i = f \) for front, \( r \) for rear and \( j = l \) for left, \( r \) for right), \( Z_b \) is the sprung mass vertical displacement at the centre of gravity, \( \theta \) is the roll angle and \( \alpha \) is the pitch angle. For the forces acting at each of the
suspensions ($F_{ij}$), it is actually the sum of the spring force ($F_{ij}s$) and damper force ($F_{ij}d$). These suspension forces are given by:

\[ F_{ij} = F_{ij}s + F_{ij}d \]  \( (5) \)

\[ F_{ijr} = F_{ijr}s + F_{ijr}d \]  \( (6) \)

\[ F_{ijl} = F_{ijl}s + F_{ijl}d \]  \( (7) \)

\[ F_{ijrr} = F_{ijrr}s + F_{ijrr}d \]  \( (8) \)

The spring forces of the suspension system are given by:

\[ F_{ij}s = K_{ij}(Z_{ij} - Z_{ij}) \]  \( (9) \)

\[ F_{ijr}s = K_{ijr}(Z_{ijr} - Z_{ijr}) \]  \( (10) \)

\[ F_{ijl}s = K_{ijl}(Z_{ijl} - Z_{ijl}) \]  \( (11) \)

\[ F_{ijrr}s = K_{ijrr}(Z_{ijrr} - Z_{ijrr}) \]  \( (12) \)

where $K_{ij}$ is the spring stiffness of the spring while $Z_{ij}$ and $Z_{ijr}$ are the unsprung mass vertical displacement and the sprung mass vertical displacement, respectively, at each side of the vehicle. The damper forces are given by

\[ F_{ij}d = C_{ij}(Z_{ij} - Z_{ij}) \]  \( (13) \)

\[ F_{ijr}d = C_{ijr}(Z_{ijr} - Z_{ijr}) \]  \( (14) \)

\[ F_{ijl}d = C_{ijl}(Z_{ijl} - Z_{ijl}) \]  \( (15) \)

\[ F_{ijrr}d = C_{ijrr}(Z_{ijrr} - Z_{ijrr}) \]  \( (16) \)

where $C_{ij}$ is the damping coefficient of the dampers while $Z_{ij}$ and $Z_{ijr}$ are the unsprung mass vertical velocity and the sprung mass vertical velocity, respectively.

For the vehicle tyres, it is modelled as a spring and the force acting at the tyres is usually known as dynamic tire loads, $F_{tij}$. The dynamic tyre load of the tyres is given by

\[ F_{tij} = K_{tij}(Z_{tij} - Z_{tij}) \]  \( (17) \)

\[ F_{tijr} = K_{tijr}(Z_{tijr} - Z_{tijr}) \]  \( (18) \)

\[ F_{tijl} = K_{tijl}(Z_{tijl} - Z_{tijl}) \]  \( (19) \)

\[ F_{tijrr} = K_{tijrr}(Z_{tijrr} - Z_{tijrr}) \]  \( (20) \)

where $K_{tij}$, $Z_{tij}$, and $Z_{tijr}$, are the tyre stiffness, road input displacement and unsprung mass displacement, respectively.

Using Newton’s Second Law at the vehicle’s sprung mass, the body vertical acceleration, $\ddot{Z}_b$ can be determined by
where $m_0$ is the total mass of the vehicle. Angular acceleration during the roll effect, $\theta$ is given by;

$$\frac{(F_{fr} + F_{rr})a}{2} - \frac{(F_{fl} + F_{rl})a}{2} = I_{xx} \dot{\theta}$$

(22)

where $a$ is the vehicle’s track width and $I_{xx}$ is the moment of inertia about x-axis. The angular acceleration while the vehicle is in pitch effect, $\alpha$ is given by;

$$\frac{(F_{fr} + F_{rr})b}{2} - \frac{(F_{fl} + F_{rl})b}{2} = I_{yy} \alpha$$

(23)

with $b$ and $I_{yy}$ are the vehicle’s wheelbase and moment about y-axis, respectively.

Acceleration of each wheel can be calculated using

$$F_{fl} - F_{fl} - F_{rl} = m_{fl} \ddot{Z}_{fl}$$

(24)

$$F_{fr} - F_{fr} - F_{rl} = m_{fr} \ddot{Z}_{fr}$$

(25)

$$F_{rl} - F_{rl} - F_{rl} = m_{rl} \ddot{Z}_{rl}$$

(26)

$$F_{rr} - F_{rr} - F_{rl} = m_{rr} \ddot{Z}_{rr}$$

(27)

Based on the derived equations, a vehicle model has been build using Matlab/Simulink block as shown in Figure 4. The parameter values used in this simulation can be referred in Appendix A.

Figure 4 Matlab/Simulink vehicle’s ride model block
2.1 Seven DOF vehicle ride model validations

The validation of the developed ride model was done by comparing the simulation results with experimental results. The purpose of the validation is to determine the validity of the model before it can be used for further study.

The experimental data were collected by using eight accelerometers to collect the vertical motion data of sprung mass and unsprung mass. An accelerometer is placed at each sprung mass and unsprung mass at all corners of the vehicle. The heave motion data, rolling motion data and pitching motion data at the centre of gravity of the vehicle were collected using the tri-axis accelerometer sensor. All of the data obtained from the sensors were collected by IMC-FAMOS data acquisition system.

For these validations, a pitch mode test was performed where the vehicle hit a bump with the height, width, and length of 7.5 cm, 40 cm and 244 cm, respectively, at a speed of 20 km/h. It is assumed that both front wheels (and rear wheels) hit the bump symmetrically. The parameters (velocity and bump dimension) used during the test were then used as inputs in the developed ride model. The comparison of the results both from experiment and simulation is shown in Figure 5. The generated simulations results from the developed 7DOF model correlate and have the same trends as the experimental data. This validates the developed ride model and, further, improves the reliability of the model used in this study.

![Experimental and simulation results comparison](see online version for colours)

3 Fundamentals of MR damper

MR damper uses magnetorheological fluid, which is considered to be a smart fluid. It has a lot of potential to be explored in the control-based application. The MR fluid that consists of micro or nano sized iron particles in a carrier fluid, when subjected to magnetic field, will change its viscosity into a semi-solid state. The behaviour of MR fluid is manipulated by researchers and engineers to provide controllable damping resisting force.

The key design of MR damper is the design of the piston inside the damper. There are two types of pistons that can be used; valve mode and direct shear mode. In valve mode as shown in Figure 6, the flow of MR fluid will be restricted by the chain-like iron particles, which are subjected to the magnetic field between two fixed plates.
The pressure drop that exists for this mode can be represented by (Poynor, 2001):

$$\Delta P = \Delta P_v + \Delta P_r = \frac{12\eta Q L}{g^2 w} \frac{c \tau_L}{g}. \quad (28)$$

In equation (28), the pressure drop ($\Delta P$) is assumed to be the sum of viscous component ($\Delta P_v$) and a field dependent induced yield stress component ($\Delta P_r$) where $Q$ represents the pressure driven MR fluid flow, $\eta$ represents the plastic viscosity of MR fluid, $\tau_r$ represents field dependent yield stress of MR fluid, $L$ represents the length, $g$ and $w$ represent the fluid gap and the width of fluid flow, respectively. The constant $c$ can be approximated by:

$$c = 2.07 + \frac{1}{1 + 0.43} \quad (29)$$

$$\exists = \frac{wg^2 \tau_0}{12Q\eta}. \quad (30)$$

In direct shear mode valve design, as shown in Figure 7, the damping resistance force exists when the MR fluid flows under the effect of the magnetic field between two poles, with one pole moving relative to a fixed pole.

The equation of damping resistance force for this design mode is (Poynor, 2001):

$$F = F_\eta + F_\tau = \frac{\eta S A}{g} + \tau S \quad (31)$$

where $F$ represents the total resistant force generated between the pole plates, $F_\eta$ is the viscous shear force, $F_\tau$ is the magnetic field dependant shear force, $A$ is the pole plate area and $S$ is the relative velocity between pole plates.
In general, equations (28) and (31) can be further manipulated to give the volume of activated MR fluid, \( V \), which can be represented by:

\[
V = k \frac{\delta \eta}{\zeta \tau} \dot{W}_m \tag{32}
\]

where \( \lambda \) is the desired control ratio and \( W_m \) is the mechanical power dissipation. The parameters in equation (32) for valve mode design can be calculated as

\[
k = \frac{12}{c} \tag{33a}
\]

\[
\lambda = \frac{\Delta P_r}{\Delta P\eta} \tag{33b}
\]

\[
W_m = Q\Delta P_r. \tag{33c}
\]

While for the shear mode valve design can be calculated as:

\[
k = 1 \tag{34a}
\]

\[
\lambda = \frac{F\tau}{F\eta} \tag{34b}
\]

\[
W_m = F_S. \tag{34c}
\]

### 3.1 MR damper design parameters and model development

The design of the MR damper is based on the OE shock absorber used in the passenger vehicle. Due to this fact, the design parameter and the performance of the OE shock have to be studied. The parameters such as the OE shock absorber tube diameter, the tube length and the stroke length have been taken into consideration in designing the MR damper, as these parameters will become the constraints in designing a MR damper that will be used as a retrofitting damper.

The performance of the OE shock absorber was also investigated and taken as a benchmark in designing the MR damper. The OE shock absorber was tested using the Material Testing System (MTS) machine and the performance of OE shock absorber is shown in Figure 8.

The design of the MR damper is done by considering the geometric design and magnetic circuit design. In order to assist the development of the MR damper, an Excel Spreadsheet such as in Figure 9 was developed.

The design started by determining several parameters that are known or desired by the designer. For this design, parameters such as the wire hole inside the hollow shaft \( (D1) \), internal diameter of the piston ring \( (D2) \), outer diameter of piston ring \( (D4) \), fluid flow orifice or fluid gap \( (g) \), piston overall length and flange thickness \( (W) \) are the parameters that need to be determined by the designers.
The selection value of wire hole \( (D_1) \) should be large enough to enable the wire to be inserted into the hollow shaft and it also should be practical enough for the manufacturing process. The outer diameter of piston ring \( (D_4) \) was chosen based on the internal diameter of OE shock absorber tube. The value of the fluid gap \( (g) \) and the overall piston length was chosen by considering the fail-safe characteristics of the designed MR damper. According to Poynor (2001), the off-state condition of MR damper should be designed with at least 50% less than the OE shock absorber performance.
The piston core diameter \((D_3)\) must be selected iteratively until the number of coil turns produces the desired maximum operating current. A good design will provide a reasonable number of coil turns. The remaining parameters, which are the piston major diameter and the coil recess length, can be calculated as the function of the input parameters.

The most important aspect in designing the magnetic circuit is to focus on the region where the MR fluid will be activated. The activation region is the region where the highest reluctance exists. The efficiency of the generated reluctance depends on the fluid gap \((g)\) value chosen. The fluid gap \((g)\) should be minimised to give the highest reluctance effect when the MR damper is activated. However, it also needs to be compromised so that the damping force generated during the off-state condition is not too low. Also, to have an efficient magnetic circuit, the piston cross-sectional area \((A)\), the piston ring cross sectional area \((C)\) and the piston radial root area \((B)\) should be the same. This is to avoid a ‘bottleneck’ of MR fluid flow during on-state condition.

The overall performance of the designed MR damper should envelope the performance of the OE shock absorber by making sure that the damping force during maximum supplied current is at least 150% more than the OE shock absorber performance during the on-state condition and at least 50% less than the OE shock absorber performance during the off-state condition. The advantage of this general rule (Poynor, 2001) is that the MR damper will have the capability of providing any value of resisting force, including the resisting force value from the OE shock absorber.

Figure 10 shows the result of the developed MR damper model using the design parameters shown in the Excel Spreadsheet in Figure 9. It is very interesting to observe that from this model, supplying a large amount of current to the MR damper does not necessarily promise a higher damping resisting force. There will be a time when the magnetic circuit will become fully saturated and any higher supplied current will only be a waste of the power source. The plotted characteristics in Figure 10 also show the current operating range which, further, will provide very useful information which is needed when applying the MR damper to a control based application.

**Figure 10** MR damper characteristic based on the designed parameters (see online version for colours)
Figure 11 shows the damping force range between the designed MR damper and the OE shock absorber. It shows the capability of the designed MR damper to produce variable damping forces, including the damping forces of an OE shock absorber.

**Figure 11** Damping force range of the design MR damper and OE shock absorber (see online version for colours)

4 Current generator model

A simple and unique current generator for the MR damper was developed to control the supplied current to the designed MR damper. The current generator was developed based on the designed MR damper characteristics, as shown previously in Figure 10.

The current required by the MR damper to be able to catch up the desired damping force estimated by the control algorithm at a specific response time is produced by IF-THEN rules given:

\[
IF \ F_d \geq F_i \ AND \ F_d \leq F_{i+1} \ THEN \ I = \frac{F_d - F_i}{\Delta F_{i(i+1)}} - I_0 + I_i
\]

(35)

where

- \( F_d \) = desired force estimated from control algorithm
- \( F_i \) = damping force at a given current supply (data from MR damper characteristics)
- \( I \) = desired current
- \( I_i = I_0, I_1, I_2 \ and \ I_3 \)
- \( I_0 = 0 \) ampere
- \( I_1 = 1 \) ampere

\[
\text{Current Range Between MR Damper and OE Shock Absorber}
\]

- 0 Ampere
- OE Shock Absorber
- 3 Ampere

- Velocity (m/s^2)
$I_2 = 2 \text{ ampere}$

$I_3 = 3 \text{ ampere}$.

5 Modified skyhook control algorithm

The modified skyhook control algorithm is chosen to provide the desired force in attenuating the road harshness in the simulation. It was proposed by Bessinger et al. (1995) as a modification of the original skyhook control algorithm back in the year 1995. The modified skyhook control algorithm includes both passive damper and the skyhook damper effects in order to overcome the problem due to the application of the original skyhook controller known as the water hammer (Miller and Nobles, 1990; Tong, 2001). The water hammer problem is one where the passenger experiences unwanted audible noise and harsh jerks caused by force discontinuity (due to high damping switches to low damping or vice versa). The equation of the modified skyhook control algorithm is given by

$$F_s = C_m \tilde{u} \ddot{Z}_s - Z_s \ddot{Z}_s + (1 - \alpha) \ddot{Z}_s \dot{\varphi}$$

(36)

where

$\alpha$: Passive to skyhook ratio

$C_m$: Damping constant of modified skyhook

$Z_u$: Unsprung mass velocity

$Z_s$: Sprung mass velocity.

In this simulation the $\alpha$ value was chosen to be 0.5. An optimal value of $C_m$ was chosen to be as 5000 as this value will cause the desired force estimated from this control algorithm to be within the range of damping forces of the designed MR damper.

6 Simulation results

Figure 12 shows the layout of the closed-loop simulation model to investigate the ride performance of the passenger vehicle, attached with the designed MR damper and the current generator model. In this simulation model layout, the modified skyhook controller uses the steady state information (sprung mass velocity and unsprung mass velocity) from the full vehicle model to estimate the desired damping force required, in order to attenuate and isolate the road harshness from the vehicle body. The current generator block which has been mapped with the characteristics of the designed MR damper will generate the required current to the MR damper model such that the MR damper model will produce the same amount of damping force as the desired force, estimated by the modified skyhook control algorithm.
The ride performance of a passenger vehicle model is evaluated using a 3-mode test via simulation: the rolling mode test, pitching mode test and a combination of these two modes. In the rolling mode test, both front wheels of the vehicle’s simulation model will hit the bump simultaneously, followed by the rear wheels. For the pitching mode test, only one side of the vehicle will hit the bump. For the combination of rolling and pitching mode test, the developed vehicle model is subjected to the road input, which will provide the rolling and pitching effect simultaneously to the vehicle model.

6.1 Roll effect simulation

In roll mode simulation, as described previously, only one side of the vehicle model (left hand side) was subjected to the road input. A step input was used for this purpose. In this simulation test, the front left and rear left side of the vehicle will hit a bump with height, width and length of 7.5 cm, 40 cm and 244 cm, respectively, at a speed of 20 km/h. Figure 13 shows the results of the simulation of the vehicle model with a passive suspension system as compared with semi-active suspension system.

The results of the simulation test for the rolling effect simulation show a significant improvement in the semi-active suspension system. It improves most of the body motion parameters; that is, jerk, sudden vertical movement, body heave, body roll rate and body roll angle, except for the body vertical acceleration. Table 1 shows the improvement achieved by the semi-active suspension system based on Root-Mean-Square (RMS) values.

It can be seen from Table 1 that the applications of the semi-active suspension system in a vehicle model can improve the studied responses during the rolling effect, and the most significance improvements were on the vehicle’s heave, and roll angle. The improvements in these parameters are 70.32% and 65.74%, respectively.

The damping force and the amount of current supplied to the MR damper model during the roll mode test can be seen in Figures 14 and 15, respectively. Figure 14 shows the amount of damping force produced by the MR damper model by following the estimated damping forces provided by the controller. The damping force range estimated by the controller during the roll mode test lies between 7500 N (bounce) and –10,250 N (rebounce). The amount of current that was supplied by the current generator to the MR damper model in order to produce this amount of damping force is between 0 amperes and 0.53 amperes. The amount of current is sufficient to produce the estimated damping force due to the fact that the MR damper is operating at a velocity of ±3 m/s during the roll mode test.
Figure 13  Simulation of vehicle body motion during rolling effect (see online version for colours)

![Graph showing body jerk, vertical acceleration, roll rate, and roll angle for passive and semi-active suspension systems.](image)

Table 1  RMS values of passive and semi-active suspension system for roll effect simulation

<table>
<thead>
<tr>
<th></th>
<th>Passive</th>
<th>Semi active</th>
<th>Improvement (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jerk</td>
<td>1624.01</td>
<td>1473.35</td>
<td>9.28</td>
</tr>
<tr>
<td>Body vertical accel.</td>
<td>13.31</td>
<td>13.69</td>
<td>-2.84</td>
</tr>
<tr>
<td>Body vertical heave</td>
<td>0.05</td>
<td>0.02</td>
<td>65.74</td>
</tr>
<tr>
<td>Body roll rate</td>
<td>0.24</td>
<td>0.19</td>
<td>19.72</td>
</tr>
<tr>
<td>Body roll angle</td>
<td>0.06</td>
<td>0.02</td>
<td>70.32</td>
</tr>
</tbody>
</table>

Figure 14  Force tracking by MR damper model during roll mode test (see online version for colours)

![Graph showing force tracking for MR damper model.](image)
The evaluation of the vehicle’s ride performance during the roll mode test in the frequency domain was also investigated in order to study the effect of the designed semi-active suspension system on the vehicle’s body. It is important that the maximum peak response in time domain analysis occurs in the frequency range of 0–2 Hz. This is the frequency range of a good ride comfort. The time domain results were transformed into the frequency domain by using the Fast Fourier Transform function. The periodogram graph (power spectrum vs. frequency) for the studied time domain results are plotted. Figure 16 shows the periodogram graphs using the linear frequency scale for each of the studied parameters during the roll mode test.
As shown in Figure 16, all the major peaks in the periodogram graphs for body jerk, body heave, body roll rate and body roll angle in roll mode test occur within the frequency range of 0–2 Hz and this means that the improvement made by the semi-active suspension system does not sacrifice the passenger’s ride comfort. However, in the periodogram for body acceleration, the second major peak in that periodogram occurs at 2.2 Hz. This slight disadvantage is compromised due to the fact that the other observed responses are within the frequency range of 0–2 Hz.

6.2 Pitch effect simulation

In the pitch effect simulation, both front wheels will hit the bump simultaneously followed by the rear wheels. The simulation road input (bump dimension) and the velocity are still the same as for the roll effect simulation. Figure 17 shows the simulation results both for the passive suspension system and for the semi-active suspension system. The performance of vehicle body motion by the semi-active suspension system is still consistent, with body jerk, body vertical heave, body pitch rate and body pitch angle showing improvements. Table 2 shows the RMS values for the discussed output parameters. The improvement in the vehicle’s vertical heave and vehicle’s pitch angle are more than 60%.

Figure 17  Simulation results of body motion during pitching (see online version for colours)
The damping forces and the currents supplied to the MR damper model during the pitch mode test can be seen in Figures 18 and 19, respectively. It can be seen that the damping range of the force during the pitch mode test lies between 8000 N (bounce) and –10,000 N (rebounce) and the estimated damping force by the controller is tracked almost perfectly by the MR damper model. The amount of current that is supplied to the MR damper model to produce the damping force as estimated by the controllers during this pitch mode test is between 0 amperes and 0.7 amperes.

**Figure 18** Force tracking by MR damper model during pitch mode test (see online version for colours)

![Force tracking by MR damper model during pitch mode test](image)

**Figure 19** Current generated from current generator model to MR damper model during roll mode test

![Current generated from current generator model to MR damper model during roll mode test](image)

The evaluation of the ride comfort frequency made by the designed semi-active suspension system to the vehicle’s body during the pitch mode test is showed in Figures 20 and 21. It can be seen that maximum peaks of the power spectrum in all periodogram graphs occurs in the frequency range of 0–2 Hz except for the periodogram graph for body vertical acceleration where the second highest peak of the semi active curve occurs at slightly more that 2 Hz. This is expected because the RMS value table (Table 2) shows a slight disadvantage for the body vertical acceleration for the semi-active suspension system.

**Table 2** RMS value of passive and semi active suspension system for pitch effect simulation

<table>
<thead>
<tr>
<th></th>
<th>Passive</th>
<th>Semi active</th>
<th>Improvement (%)</th>
</tr>
</thead>
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<tr>
<td>Jerk</td>
<td>3148.45</td>
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<tr>
<td>Body vertical acceleration</td>
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<td>25.44</td>
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<tr>
<td>Body vertical heave</td>
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<tr>
<td>Pitch rate</td>
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<tr>
<td>Pitch angle</td>
<td>0.08</td>
<td>0.03</td>
<td>67.18</td>
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6.3 Roll and pitch effect simulation

In roll and pitch effect simulation, the road input signals for the simulation model were obtained by arranging the bumps to give both roll and pitch effect simultaneously to the vehicle model when the wheels hit the bump. Figure 22 shows the results of the body when subjected to these inputs. Overall, the performance of the semi active suspension system, when subjected to the combination of roll and pitch effect, is still consistent with a significant improvement for vehicle vertical heave, roll angle and pitch angle. Table 3 shows the RMS values for passive and semi-active suspension systems when subjected to these inputs.

The RMS values in Table 3 show improvement in vehicle’s vertical motion, roll motion and pitch motion due to the designed semi-active suspension system, with significant improvements in the vehicle’s vertical heave, roll angle and pitch angle. The vertical heave improves by 71.5% and the roll and pitch angle improve by 71.4% and 67.8%, respectively, when subjected to the disturbances.
Table 3  
RMS value of passive and semi active suspension system for roll and pitch effect simulation

<table>
<thead>
<tr>
<th></th>
<th>Passive</th>
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<tr>
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<td>Body pitch rate</td>
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<tr>
<td>Body pitch angle</td>
<td>0.11</td>
<td>0.04</td>
<td>67.8</td>
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</table>

Figure 22  Simulation of body motion during rolling and pitching effect (see online version for colours)

Figures 23 and 24 shows the results of force tracking made by the MR damper model and the amount of current supplied to the MR damper model during the combination mode test. The range of the damping forces produced by the MR damper model during this
combination mode test is between 7500 N (bounce) to –9700 N (rebound) and the amount of current supplied to the MR damper model to create these damping forces is between 0 amperes and 0.7 amperes.

Figure 23 Force tracking by MR damper model during combination mode test (see online version for colours)

Figure 24 Current generated from current generator model to MR damper model during combination mode test (see online version for colours)

Figure 25 shows the periodogram results for the combination test. It can be seen that the major peaks of the periodograms are in the range of 0–2 Hz except for the periodograms for the body jerk and body acceleration where the frequency for the major peak of power
spectrum in these two periodograms are 2.79 Hz and 2.65 Hz, respectively. Even though this can be considered as a harsh ride comfort, this is compromised by small amplitudes of body vertical heave, body roll angle and body pitch angle as shown in Figure 22.

**Figure 25** Frequency domain analysis for combination mode test (see online version for colours)
7 Conclusion

The simulation of the verified 7DOF vehicle model with passive and semi-active suspension systems shows the potential of the designed semi-active suspension system. In this simulation, the designed MR damper, its simulation model and developed current generator model, were found to improve passenger vehicle’s ride performance significantly when subjected to the road harshness (bump). The developed MR damper model was also found to be able to generate the damping forces as estimated by the controllers, due to the effectiveness of the developed current generator model. The amount of current supplied during all the mode tests is also small, with the current generated by the MR damper model being less than 1 ampere.

The overall performance by the designed semi-active suspension is also found to be very consistent even when different disturbances are applied to the simulation model. General improvements made by the designed MR damper when used with modified skyhook control algorithm are the reduction of the studied responses magnitudes (i.e., jerk, vertical heave etc.) as well as the reduction of the settling. The improvement made by the designed semi-active suspension also did not sacrifice the vehicle’s ride comfort and if a harsh ride comfort occurs, it will be compromised with small amplitudes of body vertical heave, body roll angle and body pitch angle.

Acknowledgements

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References


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Appendix A: 7DOF vehicle model parameters

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