Analysis of Train Suspension System Using MR dampers

DVA RamaSastry1,2, K V Ramana1, N Mohan Rao2, SVR Siva Kumar1 and T G L Priyanka1

1Department of Mechanical Engineering, K.L. University, Guntur, AndhraPradesh, India
2Department of Mechanical Engineering, University college of Engineering, Kakinada, JNT University, Kakinada, AndhraPradesh, India
e-mail: dvarsasstry@kluniversity.in

Abstract: This paper deals with introducing MR dampers to the Train Suspension System for improving the ride comfort of the passengers. This type of suspension system comes under Semi-active suspension system which utilizes the properties of MR fluid to damp the vibrations. In case of high speed trains, the coach body is subjected to vibrations due to vertical displacement, yaw and pitch movements. When the body receives these disturbances from the ground, the transmission of vibrations to the passenger increases which affect the ride comfort. In this work, the equations of motion of suspension system are developed for both conventional passive system and semi-active system and are modelled in Matlab/Simulink and analysis has been carried out. The passive suspension system analysis shows that it is taking more time to damp the vibrations and at the same time the transmissibility of vibrations is more. Introducing MR dampers, vertical and angular displacements of the body are computed and compared. The results show that the introduction of MR dampers into the train suspension system improves ride comfort.

Keywords: Magnetorheological dampers, high speed trains, secondary and primary suspension systems.

1. Introduction

With a concern of improving effectivity of public transport systems and also for cleaner environment, there is a rise in interest on high speed trains. But high speed trains are prone to problems in ride comfort and vehicle stability due to increased vibrations in the coach body [1]. The quality of ride and vehicle stability is influenced by acceleration and displacement of disturbances respectively [2]. The passive suspension systems will not be enough to cater the needs of the suspension of these trains, as design of such systems involves compromise in one of these two factors. The active suspension systems are complex and costlier to implement [3]. The evolution of semiactive suspension system has lead towards better suspension systems in high speed trains. Now a days, Magnetorheological fluids are extensively studied for damping the vibrations in various applications [4]. Magnetorheology is a branch of Rheology that deals with the flow and deformation of the materials under an applied magnetic field. In this work, MR dampers consisting of Magnetorheological fluid which consists of silicone oil along with carbonyl iron particles and additives are considered. MR fluids are able to react reversibly and instantaneously with the application of magnetic field [5]. They can change from a free-flowing liquid to a semi-solid within a few milliseconds with controllable yield strength, when subjected to a magnetic field [6]. In the absence of an applied field, an MR fluid is reasonably well...
approximated as a Newtonian liquid flowing freely with a consistency similar to mineral oil [7]. When a magnetic field is applied, the ferrous particles begin to align along the flux path, eventually forming particle chains in the fluid. Such chains resist and restrict fluid movement. As a result, a yield stress develops in the fluid [8]. In this work, the equations of motion of train coach suspension system are developed for both conventional passive system and semi-active system and are modelled in Matlab/Simulink and analysis has been carried out to study their response to track disturbances. Even though there are various parametric models available to model MR damper, in this study, Spencer model is considered, as it is identified to be more appropriate model in modelling the behaviour of MR damper [9].

2. Modelling of train Coach body Suspension system
In this study, a single bogie which consists of two trucks namely, front truck and rear truck and four wheel sets is considered as shown in Figure 1. The complete system is connected with the help of primary suspension system and secondary suspension system. The suspension system between the wheels and the frame is termed as primary suspension system, which consists of conventional springs and viscous dampers. The suspension system between the coach body and the truck frame is termed as secondary suspension system which again consists of conventional springs and four viscous dampers. In this case, four MR dampers have been incorporated instead of these conventional viscous dampers in secondary suspension to study the improvement in the damping properties.

![Figure 1. Schematic Diagram of Train Suspension System](image)

2.1. Governing Equation of Motion for Train Suspension System
The equations of motion for the entire train suspension system as shown in figure 1 are derived from basic theory of vibrations and are shown below. The nomenclature and force relations are shown in appendix.

2.1.1. Equation of motion for Complete Train Suspension System with passive suspension.

\[ M_1 \ddot{d} = F_{sszl}r + F_{sszl}l + F_{ssztr} + F_{ssztl} \]

\[ I_2 \ddot{\beta} = (F_{sszl}r + F_{sszl}l)h_i + (F_{ssztr} + F_{ssztl})h_i - (F_{sszl}r + F_{sszl}l)p + (F_{ssztr} + F_{ssztl})p \]
\[ I_f \ddot{\alpha} = -(Fssylr + Fssyll) h_1 - (Fssytr + Fssytl) h_1 + (Fsszlr + Fssztl)m - (Fsszll + Fssztl)m \]

2.1.2. Equation of motion for front truck.

\[ M_2 \ddot{f}_j = -(Fsszlr + Fsszll) + (Fpszlr + Fpsz2r) + (Fpszl + Fpsz2l) \]

\[ I_f \dot{\phi}_1 = (Fssxlr + Fssxll) h_2 + (Fpsxlr + Fpsxll) h_3 + (Fpsx2r + Fpsx2l) h_3 \\
- (Fpszlr + Fpszll) v + (Fpsz2r + Fpsz2l) v \]

\[ I_f \dot{\gamma} = -(Fssylr + Fssyll) h_2 - (Fsszlr - Fsszll)m - (Fpsy1r + Fpsy2r) h_3 \\
- (Fpsyll + Fpsy2l) h_3 + (Fpszlr + Fpsz2r) n - (Fpszl + Fpsz2l)n \]

2.1.3. Equation of motion for Rear truck.

\[ M_2 \ddot{f}_r = -(Fssztr + Fssztl) + (Fpsz3r + Fpsz3l) + (Fpsz4r + Fpsz4l) \]

\[ I_f \dot{\phi}_3 = (Fssxtr + Fssxtl) h_2 + (Fpsx3r + Fpsx3l) h_3 + (Fpsx4r + Fpsx4l) h_3 \\
- (Fpsz3r + Fpsz3l) v + (Fpsz4r + Fpsz4l) v \]

\[ I_f \dot{\delta} = -(Fssytr + Fssytl) h_2 - (Fssztr - Fssztl)m - (Fpsy3r + Fpsy4r) h_3 \\
- (Fpsy3l + Fpsy4l) h_3 + (Fpsz3r + Fpsz4r) n - (Fpsz3l + Fpsz4l)n \]

2.2. Equation of motion for Complete Train Suspension System with MR damper.

\[ M_1 \ddot{d} = Fsszlr + Fsszll + Fssztr + Fssztl - fnmr - fmrl - fnmtr - fnmr \]

\[ I_f \dot{\beta} = (Fssxlr + Fssxll) h_1 + (Fssxtr + Fssxtl) h_1 - (Fsszlr + Fsszll) p + (Fssztr + Fssztl) p + \\
(fnmr + fmrl)p - (fnmtr + fnmr)p \]

\[ I_f \dot{\alpha} = -(Fssylr + Fssyll) h_1 - (Fssytr + Fssytl) h_1 + (Fsszlr + Fssztl)m - (Fsszll + Fssztl)m - \\
(fnmr + fnmtr)m + (fmrl + fnmr)m \]

2.2.1. Equation of motion for front truck

\[ M_2 \ddot{f}_j = -(Fsszlr + Fsszll) + (Fpszlr + Fpsz2r) + (Fpszl + Fpsz2l) + (fnmr + fmrl) \]

\[ I_f \dot{\phi}_1 = (Fssxlr + Fssxll) h_2 + (Fpsxlr + Fpsxll) h_3 + (Fpsx2r + Fpsx2l) h_3 \\
- (Fpszlr + Fpszll) v + (Fpsz2r + Fpsz2l) v \]

\[ I_f \dot{\gamma} = -(Fssylr + Fssyll) h_2 - (Fsszlr - Fsszll)m - (Fpsy1r + Fpsy2r) h_3 \\
- (Fpsyll + Fpsy2l) h_3 + (Fpszlr + Fpsz2r) v - (Fpszl + Fpsz2l)v + (fnmr - fnmtr)m \]

2.2.2. Equation of motion for Rear truck

\[ M_2 \ddot{f}_r = -(Fssztr + Fssztl) + (Fpsz3r + Fpsz3l) + (Fpsz4r + Fpsz4l) + (fnmtr + fnmr) \]

\[ I_f \dot{\phi}_3 = (Fssxtr + Fssxtl) h_2 + (Fpsx3r + Fpsx3l) h_3 + (Fpsx4r + Fpsx4l) h_3 \\
- (Fpsz3r + Fpsz3l)v + (Fpsz4r + Fpsz4l)v \]
\[ I_3 \dot{\delta} = -(F_{ssytr} + F_{ssytl})h_2 - (F_{ssztr} - F_{ssztl})m - (F_{psy3r} + F_{psy4r})h_3 \\
-(F_{psy3l} + F_{psy4l})h_3 + (F_{psz3r} + F_{psz4r})n - (F_{psz3l} + F_{psz4l})n + (f_{mrtr} - f_{mrtil})m \]

3. **Modelling of suspension system in MATLAB/SIMULINK**

The above equations of motion of passive suspension system for train are modelled in Matlab/Simulink considering the values for various parameters of Train Suspension system as shown in Table 1.

Table 1: Train Suspension model Parameters

| Parameter | Value         | Parameter | Value         | Parameter | Value         |
|-----------|---------------|-----------|---------------|-----------|---------------|
| M_1       | 39600 kg      | M_2       | 3250 kg       | I_6       | 250500 kgm²   |
| I_1       | 88500 kgm²    | I_2       | 2460000 kgm²  | I_3       | 4270 kgm²     |
| I_3       | 3060 kgm²     | I_4       | 3020 kgm²     | K_5       | 700000 N/m    |
| K_4       | 4000000 N/m   | K_5       | 3250000 N/m   | C_6       | 150000 Ns/m   |
| C_4       | 0             | C_5       | 0             | K_5       | 290000 N/m    |
| K_2       | 150000 N/m    | K_1       | 150000 N/m    | C_1       | 80000 Ns/m    |
| C_2       | 0             | C_3       | 50000 Ns/m    | h_2       | -0.452 m      |
| h_1       | 0.217 m       | h_1       | 1.207 m       | a         | 0.7466 m      |
| m         | 1 m           | p         | 9 m           | n         | 1 m           |
|           |               |           |               |           |               |

4. **Modelling of MR damper using Spencer model**

Various parametric models were proposed to simulate the behaviour of MR dampers. The model proposed by Spencer [9], a modified Bouc-Wen model, is the most appropriate model among all the available parametric models. The Spencer model is represented in Figure 2.

![Spencer model of MR damper](image)

According to Spencer model, the damping force produced by the MR damper is given by

\[ f_{MR} = \alpha z + C_0(\dot{x} - \dot{y}) + k_0(x - y) + k_1(x - x_0) \]
\[ \dot{z} = -\gamma |\dot{x} - \dot{y}|(|z|^{n-1} - \beta(\dot{x} - \dot{y})|z|^n + A(\dot{x} - \dot{y})) \]
\[ \dot{y} = \frac{1}{\varepsilon_0 + \varepsilon_1} \left[ \alpha z + C_0(\dot{x}) + k_0(x - y) \right] \]

Table 2: Spencer damper model parameters [10]

| System Parameter | Values at 0.4 amp | System Parameter | Values at 0.4 amp |
|------------------|-------------------|------------------|-------------------|
| Alpha            | 86665 N/m         | K_1              | 675.30 N/m        |
| Beta             | 1198420.0 m⁻¹     | K_0              | 2807.8 N/m        |
| Gamma            | 52260.00 m⁻¹      | C_1              | 20626.5 Ns/m      |
| n                | 2                 | C_0              | 1965.10 Ns/m      |
| A                | 29.37             |                  |                   |
These equations are modelled in MATLAB/SIMULINK and shown in the Figure 3. The values of various parameters[10] mentioned in the equation, at 0.4amps current supply are shown in Table2.

![Simulink modelling-phenomenological Spencer model of MR damper](image)

**Figure 3.** Simulink modelling-phenomenological Spencer model of MR damper

The figures 4, 5 and 6 shows Force Vs Displacement, Force Vs Velocity and Force Vs Time plots respectively for MR damper, when Spencer model is applied. These plots show the heuristic and non-linear behaviour of MR damper to the most appropriate extent.

This Spencer model of MR damper is used to incorporate MR damper in the place of secondary viscous damper and the semi active suspension is modelled in Matlab/Simulink.
5. Analysis of Passive suspension system
In order to analyse the passive suspension system, the wheels which are on the left side of the coach body are given an initial disturbance of 1cm on the track. Due to this initial disturbance the body starts vibrating and this vibration is transferred to the coach body through primary and secondary suspension systems. The behaviour of the suspension system is observed by plotting magnitude of vertical displacement, vertical acceleration, yaw movement, pitch movement, Yaw acceleration, Pitch acceleration of coach body with respect to time.

6. Analysis of semi-active suspension system
Similarly the same train model is analysed incorporating MR dampers in the secondary suspension system replacing the secondary viscous dampers in Matlab/Simulink. To activate the MR damper, operating current is to be supplied. The train system is analysed for MR damper at 0.4amp. Similar disturbances that were given to passive system were applied to this system also. The plots obtained for maximum vertical displacement, maximum vertical acceleration, maximum yaw movement, maximum pitch movement, Yaw acceleration, Pitch acceleration of coach body in both the cases relatively were shown in figures 7 to 12. The maximum amplitude values of all these parameters for both the cases were represented in Table. 3.
Figure 8. Angular displacement (pitch) for Passive Vs Semi-active Suspension system (MR damper at 0.4 amps)

Figure 9. Angular displacement (yaw) for Passive Vs Semi-active Suspension system (MR damper at 0.4 amps)

Figure 10. Vertical Acceleration for Passive Vs Semi-active Suspension system (MR damper at 0.4 amps)
The full coachbody is analyzed for vertical displacement, yaw, pitch, acceleration, yaw acceleration and pitch acceleration with passive and MR dampers (at 0.4 amps). The results obtained clearly show the effectiveness of MR dampers and improvement of ride comfort for passengers with semiactive suspension system. Table 3 also shows the percentage of improvement possible in each of the effecting parameters of the ride quality and vehicle handling of train.

### Table 3: Identified values of Train suspension system for passive and semiactive

| Train suspension parameters         | Passive  | Semiactive | % improvement |
|------------------------------------|----------|------------|---------------|
| Vertical displacement of coach body| 10 mm    | 9 mm       | 10.0          |
| Maximum vertical acceleration      | 0.125 m/s² | 0.105 m/s² | 16.0          |
| Maximum yaw moment                | 0.92 rad | 0.75 rad   | 18.5          |
| Maximum pitch moment              | -2.7 rad | -2.5 rad   | 7.4           |
| Yaw acceleration                   | 0.0175 rad/s² | 0.014 rad/s² | 20.0          |
| Pitch acceleration                 | 0.045 rad/s² | 0.043 rad/s² | 4.4           |

The full coachbody is analyzed for vertical displacement, yaw, pitch, acceleration, yaw acceleration and pitch acceleration with passive and MR dampers (at 0.4 amps). The results obtained clearly show the effectiveness of MR dampers and improvement of ride comfort for passengers with semiactive suspension system. Table 3 also shows the percentage of improvement possible in each of the effecting parameters of the ride quality and vehicle handling of train.

### 7. Conclusion

The analysis shows that semi-active suspension system has reduced the affect for the given disturbance. The results show that replacement of Passive dampers with MR dampers have significant affect of damping in both displacement and acceleration of vertical, yaw and pitch directions. Hence, it can be concluded that by introducing MR dampers in train suspension systems, the better ride quality and vehicle handling can be achieved.
Appendix

\[ K_1 = \text{Vertical Stiffness of secondary suspension} \]
\[ K_2 = \text{Longitudinal Stiffness of secondary suspension} \]
\[ K_3 = \text{Lateral Stiffness of secondary suspension} \]
\[ K_4 = \text{Vertical stiffness of primary suspension} \]
\[ K_5 = \text{Longitudinal stiffness of primary suspension} \]
\[ K_6 = \text{Lateral stiffness of primary suspension} \]
\[ C_1 = \text{Vertical viscous damping of secondary suspension} \]
\[ C_2 = \text{Longitudinal viscous damping of secondary suspension} \]
\[ C_3 = \text{Lateral viscous damping of secondary suspension} \]
\[ C_4 = \text{Vertical viscous damping of primary suspension} \]
\[ C_5 = \text{Longitudinal viscous damping of primary suspension} \]
\[ C_6 = \text{Lateral viscous damping of primary suspension} \]
\[ \alpha = \text{Angular displacement of coach body} \]
\[ \beta = \text{Angular displacement of coach body about perpendicular axis} \]
\[ \gamma = \text{Angular displacement of front truck} \]
\[ \phi_1 = \text{Angular displacement of rear truck about perpendicular axis} \]
\[ \phi_2 = \text{Angular displacement of front truck about perpendicular axis} \]
\[ \delta = \text{Angular displacement of rear truck} \]
\[ d = \text{Vertical displacement of train body} \]
\[ p = \text{length of the train} \]
\[ d_{t1} = \text{Vertical displacement of front truck} \]
\[ d_{t2} = \text{Vertical displacement of rear truck} \]
\[ h_1 = \text{Distance between coach body C.O.G and secondary suspension} \]
\[ h_2 = \text{Distance between truck frame C.O.G and secondary suspension} \]
\[ h_3 = \text{Distance between wheel set C.O.G and secondary suspension} \]
\[ V = \text{wheel base/2} \]
\[ n = \text{Primary suspension gap/2} \]
\[ m = \text{Secondary suspension gap/2} \]
\[ M_1 = \text{Mass of the Train body} \]
\[ M_2 = \text{Mass of front truck} \]
\[ M_3 = \text{Mass of Rear truck} \]
\[ I_1 = \text{Yaw moment of inertia of truck} \]
\[ I_2 = \text{Pitch moment of inertia of truck} \]
\[ I_3 = \text{Yaw moment of inertia of train} \]
\[ I_4 = \text{Pitch moment of inertia of train} \]

MR Damper

\[ Z = \text{displacement of Bouc-wen model} \]
\[ Y = \text{displacement of dashpot} \]
\[ C_1 = \text{coefficient of viscous damping at higher velocities} \]

Forces Explanation

In order to get the equations in a simplified manner we have introduces force terms in different directions. They are mentioned below.

\[ -K_1(d - p\beta - x + m(\alpha - \gamma)) - C_1(d - p\beta - x + m(\alpha - \gamma)) = F_{sszlr} \]
\[ -K_1(d - p\beta - x - m(\alpha - \gamma)) - C_1(d - p\beta - x - m(\alpha - \gamma)) = F_{sszll} \]
\[ -K_1(d + p\beta - y + m(\alpha - \delta)) - C_1(d + p\beta - y + m(\alpha - \delta)) = F_{ssztr} \]
\[ -K_1(d + p\beta - y - m(\alpha - \delta)) - C_1(d + p\beta - y - m(\alpha - \delta)) = F_{ssztl} \]
\[ -K_2(h_1\beta + h_2\phi) - C_2(h_1\beta + h_2\phi) = F_{ssxlr} \]
\[ -K_2(h_1\beta + h_2\phi) - C_2(h_1\beta + h_2\phi) = F_{ssxll} \]
\[ -K_3(h_1\beta + h_2\phi) - C_3(h_1\beta + h_2\phi) = F_{ssylr} \]
\[ -K_3(h_1\beta + h_2\phi) - C_3(h_1\beta + h_2\phi) = F_{ssyll} \]
$K_s(h\alpha + h_2\delta) + C_s(h\alpha + h_2\dot{\delta}) = F_{ssytr}$

$-K_s h_3\phi_1 - C_s h_3 \dot{\phi}_1 = F_{psx1r}$

$-K_s h_3\phi_1 - C_s h_3 \dot{\phi}_1 = F_{psx1l}$

$-K_s h_3\phi_1 - C_s h_3 \dot{\phi}_1 = F_{psx2r}$

$-K_s h_3\phi_1 - C_s h_3 \dot{\phi}_1 = F_{psx2l}$

$-K_s h_3\phi_2 - C_s h_3 \dot{\phi}_2 = F_{psx3r}$

$-K_s h_3\phi_2 - C_s h_3 \dot{\phi}_2 = F_{psx3l}$

$-K_s h_3\phi_2 - C_s h_3 \dot{\phi}_2 = F_{psx4r}$

$-K_s h_3\phi_2 - C_s h_3 \dot{\phi}_2 = F_{psx4l}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy1r}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy1l}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy2r}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy2l}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy3r}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy3l}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy4r}$

$K_s(h_3\gamma + C_s h_3 \dot{\gamma}) = F_{psy4l}$

References

[1] Satoshi Koizumi 2013 *Advance In Railway Vehicle Technology And Future Prospects Mainly In Relation To Bogie* (Nippon Steel & Sumitomo Metal Technical Report No. 105)

[2] Grag V K and Dukkipati R V 1984 *Dynamics of Railway Vehicle Systems*, (Ontario : Academic Press Inc.)

[3] Sinasi Arslan 1999 Control of Active Vehicle Suspension Systems, *Proc. Int. Conf. on Electrical and Electronics Engineering* (Bursa, Turkey) pp 160-65.

[4] Mark R Jolly, Jonathan W Bender, and J David Carlson. Properties and Applications of Commercial Magnetorheological Fluids *Proc. 1998 SPIE 3327, Smart Structures and Materials 1998: Passive Damping and Isolation* (L. Porter Davis, San Diego, CA), 3327, pp 262–75

[5] Guglielmino E, Sireteanu T, Stammers CW, Ghita G andGiuclea M2008*Semi-Active Suspension Control: Improved Vehicle Ride and Road Friendliness.* (Springer-Verlag London)

[6] Carlson J D 1999 *Magnetorheological Fluid Actuators : Adaptronics And Smart Structures* (Springer-Verlag Berlin) pp 180–95
[7] Carlson J D and Spencer Jr B F 1996 Magneto-Rheological Fluid Dampers for Semi-Active Seismic Control. *Proc 3rd Int Conf on Motion and Vibration Control* (Chiba Japan) 3 pp 35-40.

[8] Kordonsky W 1993 Magnetorheological Effect as a base of New Devices and Technologies. *Journal of Magnetism and Magnetic Materials*, **122** pp 395-98.

[9] Spencer Jr B F, Dyke S J, Sain M K and Carlson J D 1997 Phenomenological Model For Magneto-Rheological Dampers. *ASCE Journal of Engineering Mechanics* **123** (3) pp 230-38

[10] Sapinski B and Filus J 2003 Analysis of Parametric Models of MR Linear Damper *Journal Of Theoretical and Applied Mechanics* **41**(2) pp 215-40.