Modeling and control of hybrid MR seat damper and whole body vibration evaluation for bus drivers

Olivier Munyaneza¹ and Jung Woo Sohn²

Abstract
This paper describes the design, simulation, and performance evaluation of hybrid MR damper on quarter bus semi-active seat suspension coupled with human biodynamic model. Also, the whole body vibration (WBV) exposures were evaluated based on the international standard ISO 2631 (1997), and its parameters were used to measure the level of discomfort for bus drivers. The hybrid MR damper was proposed to enhance the damping force within low current supplied and achieve a fail-soft capability in case of electrical failure. The characteristics of the proposed hybrid MR damper were compared to the conventional MR damper by considering the same size, materials, and current input. The designed damper was incorporated to seat suspension system coupled with biodynamic lumped model, and the governing equations of motion of the full model were derived. Skyhook controller was used to control the amount of current to be supplied to hybrid MR damper. The controlled semi-active hybrid MR and conventional MR seat suspension are compared to uncontrolled system for two types of road excitation. The simulated results show that the driver seat comfort was improved by the skyhook controller than the uncontrolled case. The evaluated WBV showed that the hybrid MR damper can improve the driver life from fairly uncomfortable to little discomfort.

Keywords
Magnetorheological fluid, hybrid magnetorheological damper, whole body vibration, biodynamic lumped model, skyhook controller

Introduction
Driver’s fatigue and lower back pain are the common issues related to ride discomfort. The source of discomfort is mostly caused by the vibration from the road roughness which is transmitted to the driver seat. Most of researchers have revealed that professional bus drivers are prone to low back pain (LBP) than others.¹ To know how the vibration affects the driver’s health, there are a lot of research studies concerning whole body vibration (WBV) exposure for professional drivers. Thamsuwan et al.² conducted a comparison study of measuring the WBV exposure in high floor coach and low floor city bus. They have found that the seats only attenuate 10% of the transmitted vibration and attenuate it on the speed humps. Okunribido et al.³ conducted a study about city bus driving and LBP and they have found that the city bus drivers spent more than 60% of their work time driving. To reduce the vibration risks for the health of the drivers, different techniques were developed to improve the driver seat comfort. Tengler et al.⁴ proposed an optimal springing control of a seat for driver comfort improvement. In their work, they were aiming to calibrate the existing system and an improvement of 58% and 68% was achieved for smooth bump and sharp bump, respectively. All the above authors were focused on improving ride comfort.

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either by optimizing or introducing active system to the existing passive model. However, all the models have the benefits and drawback. Passive system which is equipped by spring, shock absorber and sometime stoppers is well known for their reliability and low cost but has poor rider comfort and limited operational range. On the other hand, active suspension system which uses actuators to generate force was provided to improve the ride comfort significantly. However, its higher cost and extra space requirement for the system has been a challenge for the car manufacturers. To overcome the higher cost of active suspension and keep harnessing the reliability and easiness of passive suspension, a semi-active suspension was introduced. Unlike actuators, magnetorheological (MR) damper is commonly used as semi-active device. A lot of works about semi-active suspension were conducted and different equipment featuring MR dampers are already available at the market. Currently, semi-active MR damper is still a hot topic due to its fast response time, controllable damping force, and lower energy consumption compared to active system. Du et al. proposed a semi-active control for full car with seat suspension and driver body using ER damper, and their results showed that semi-active control can provide better ride comfort than passive. Metered and sika presented a vibration control of semi-active seat suspension using fuzzy logic to control MR damper force.

To contribute in this study, a hybrid MR fluid damper for seat suspension is proposed. Unlike the conventional MR damper, this model incorporates a permanent magnet (PM) which helps to enhance the yield stress of MR fluid and keep the activeness of hybrid MR damper in case electrical circuit failure. Also, the hybrid MR damper is evaluated under eight DOF of quarter bus coupled with human biodynamic model to show its effectiveness on vibration isolation. The results presented in this work are obtained through computer simulation due to the time consuming, cost and much effort required to conduct experimental work. The rest of this paper is organized as follows: in section Modeling of hybrid MR damper, hybrid MR damper is modeled; section Seat suspension system, biodynamic lumped model coupled with quarter bus model is developed; section Simulation results and discussions is for Simulation results and evaluation; and section Whole body vibration analysis is for conclusion.

**Modeling of hybrid MR damper**

Generally, the conventional MR dampers generate low damping force in off-state condition. The damping force is increased as the current is supplied to the electromagnetic coil. However, in case of electric circuit failure, the MR damper retains the behavior of passive damper which also called fail-soft. The hybrid MR damper is composed by two PMs apart from electromagnetic coil. In off-state condition, the two PMs will generate low magnetic field in the annular gap. In the on-state condition, the electromagnetic coil will also generate enough magnetic field depend on the supplied current and these magnetic field will be added to the initial one generated by PM to form the total magnetic field in the damper valve. The use of hard magnets in our model help to overcome the fail-soft issue when the smaller current is supplied to the coil. The size of hard magnet should be carefully selected as it can lead to fail-hard also called higher initial damping force which is not beneficial to our application. Figure 1 shows a full schematic configuration of the proposed hybrid MR seat damper and structural dimension which are also presented. Figure 2 represents the cross section of piston valve of the proposed hybrid MR seat damper. Where part 1 is core, 2 is PM, 3 is coil, 4 is coil insulator, 5 is piston rod, 6 is flux return, 7 is bottom pole, 8 is upper pole, and 9 is piston outer seal.

Due to the nonlinearity of the magnetic materials which compose the piston valve, it is difficult to measure analytically the magnetic induction and yield stress of MR fluid in the annular gap. The finite element analysis using ASNYS MAXWELL software was used to simulate and estimate the magnetic flux density and field intensity respect to various current input. Figure 3 shows a 2D symmetric model for hybrid MR valve and it shows how the magnetic flux pass through the annular gap for both magnets and primary coil. All the dimensions for hybrid MR piston valve are mentioned in Table 1. Figure 4 shows the distribution of magnitude field lines and magnetic flux density from both conventional and hybrid MR dampers. The physical properties of MR fluid (MRF-132DG) from LORD Corp was taken as controllable fluid for our damper. According to Ref. 7, the yield stress induced by input current in annular gap is obtained as follows

\[
\tau_y = \alpha H^\beta
\]

where \(H\) is magnetic field intensity of MR fluid obtained through simulation. \(\alpha\) and \(\beta\) are the intrinsic values of MR fluid which are experimentally obtained. In off-state condition, the two PMs produce a magnetic field which is fairly low. When the current is supplied to the primary coil, a significant magnetic field is generated to enhance the low flux density generated by magnets. As you keep varying the current, the yield stress in the annular gap is also increasing which in return changes the velocity profile of the fluid. To analyze the performance of hybrid MR valve, an approximated laminar flow containing
Figure 1. Schematic configuration of hybrid magnetorheological damper.

Figure 2. Structural configuration of piston valve for hybrid magnetorheological damper.
MR fluid, which follows Bingham plastic model, was assumed to be incompressible with negligible inertia force. The governing equation for laminar flow in the absence of magnetic field was given as follows

$$\frac{\Delta P}{Q} = \frac{12\eta L}{bT_g^3}$$  \hspace{1cm} (2)$$

where $\Delta P$ is the pressure drop, $Q$ is the fluid flow rate in annular gap, $L$ is the annular length, $\eta$ is the viscosity of MR fluid, $b$ is the valve width, and $T_g$ is the annular gap. The pressure drop due to magnetic field is expressed as follows

![Figure 3. Magnetic circuit of hybrid type magnetorheological damper.](image)

| Parameter symbol | Description                          | Dimension, mm |
|------------------|--------------------------------------|---------------|
| $R_o$            | Radius of piston rod                 | 4.95          |
| $L_p'$           | Length of minor pole                 | 1.4           |
| $T_g$            | Annular gap size                     | 0.8           |
| $L_P$            | Length of main pole                  | 4             |
| $R_c$            | Radius of core                       | 11            |
| $m$              | Width of magnet                      | 2.75          |
| $R_p$            | Inner radius of hybrid MR valve      | 13.5          |
| $R$              | Outer radius of hybrid MR valve      | 18.1          |
| $L$              | Length of hybrid MR valve            | 23.8          |
| $T_c$            | Thickness of flux return             | 4             |

Note: MR: magnetorheological.
\[ \Delta P_{MR} = \frac{C L \tau_y}{T_g} \] (3)

where \( C \) is the dependent coefficient of flow velocity profile which usually varies between 2.07 and 3.07 and it can be approximated as

\[ C = 2.07 + \frac{12 Q \eta}{12 Q \eta + 0.8 \pi (R_p + 0.5 T_p) T_g^2 \tau_y} \] (4)

where \( L \) is the active length \((L = 2L_P + 2L_P)\). The dynamic yield stress \((\tau_y (KPa))\) of MR fluid related to the magnetic flux density in the annular fluid gap is derived as follows

\[ \tau_y (B_{MR}) = C_0 + C_1 B_{MR} + C_2 B_{MR}^2 + C_3 B_{MR}^3 + C_4 B_{MR}^4 \] (5)

For

\[ C_0 = 0.1442; \quad C_1 = 13.7; \quad C_2 = 158.79; \quad C_3 = 176.51; \quad C_4 = 42.962 \]

By neglecting the effect of gas compliance and friction force, the total damping force for the proposed hybrid MR damper is derived as follows

\[ F_D = \Delta P_{VP} (A_P - A_r)^2 + \Delta P_{MR} (A_P - A_r) \text{sgn}(V_P) \] (6)

where \( A_P \) and \( A_r \) are the piston valve and piston rod cross-sectional areas. \( V_P \) is the piston valve velocity and \( \text{sgn}(.) \) is the signum function.

From equation (6), it is clearly seen that the damping force is composed by two parts. Part one is related to fluid viscosity of MR fluid and the structure of hybrid MR damper; it is also called viscous damping force. On the other hand, there is damping force which is related to yield stress of MR fluid in on-state condition and it varies with the magnetic field strength induced. The controlled force is also called Coulomb damping force.

Figure 5(a) and (b) illustrates the simulation results of damping force against piston velocity at various magnetic fields obtained from both conventional and hybrid MR damper, respectively. As shown in Figure 5(a), at the off-state condition \((I = 0 \text{ A})\), the damping force resulted from the conventional MR damper is revealed to be around 3.92 N–5.74 N at different piston velocities. On the other hand, as shown in Figure 5(b), the damping force of the hybrid MR damper varies from 40.8 N up to 55.0 N. At this point, the hybrid MR damper outperformed the conventional MR damper in terms of providing a fail-safe solution in case of electric circuit failure. When the input current is surged up, the yield stress force is gradually
increasing due to the MR effect from both cases. For instance, at the varying piston velocity of 0–0.08 m/s and the input current of $I = 0.4 \, \text{A}$; the yield stress force for conventional MR damper varying from 315.2 N to 353.2 N. While at the same condition, the damping force from the hybrid MR damper varies from 705.5 N to 749.5 N which is more than two times the conventional MR damper. Finally, at the maximum current of $I = 0.8 \, \text{A}$, the hybrid MR damper and conventional MR damper models are capable of producing the maximum damping force of 1300.7 N and 1158.3 N respectively. It is clearly seen that the proposed hybrid MR damper has proved to be more effective than the current conventional model in terms of producing higher yield stress force at the same current input. Also, a smaller damping force generated by conventional MR damper is not enough to solve the failure-safe issue which is the opposite of the proposed model.

**Seat suspension system**

Since a decade ago, different constructions from lower to higher degree of freedoms (DOFs) for seat suspension have been proposed. Metered et al. proposed a single DOF for seat suspension. A simplest form with 4 DOF was proposed by Gündoğdu. Also higher degrees of freedom for seat coupled with human model were presented by Refs and . Since we will deal with the WBV for bus driver, we preferred to work with a model of eight DOF as shown at Figure 6, where 4 DOF are for human model, two DOF for seat system and two DOF for quarter bus model.

For model simplification, the horizontal motion was ignored; translational and rotational motions were also ignored and backrest posture was assumed to be at right angle. The damper system for seat suspension is leaned at an angle ($\phi$) of 60° at an idle position. The human model was adopted from Ref. with assumed a total mass of 60.67 Kg. The four rigid mass for human body are lower torso or pelvis ($M_3$), viscera ($M_4$), upper torso ($M_5$), and head ($M_6$). All the design parameters for the proposed model are illustrated in Table 2.

The dynamic model equations for the system are described below from their static equilibrium state:

**First case:** Quarter bus model equations

$$
\begin{align*}
M_S \ddot{X}_S &= -K_S (X_S - X_U) - C_S \left( \dot{X}_S - \dot{X}_U \right) \\
M_U \ddot{X}_U &= K_S (X_S - X_U) + C_S \left( \dot{X}_S - \dot{X}_U \right) - K_r (X_U - X_r)
\end{align*}
$$

**Second case:** seat and human model equations
Table 2. Model parameters.

| Mass (Kg) | Spring constant (N/m) | Damping coefficient (N.S/m) |
|-----------|-----------------------|-----------------------------|
| M_0 = 87.15 | K_4 = 200000          | —                           |
| M_5 = 1749.4 | K_3 = 20000           | C_5 = 15000                 |
| M_1 = 40   | K_1 = 7414.86         | C_1 = 950                   |
| M_2 = 4    | K_2 = 20000           | C_2 = 500                   |
| M_3 = 36   | K_3 = 49340           | C_3 = 2475                  |
| M_4 = 5.5  | K_4 = 20000           | C_4 = 330                   |
| M_5 = 15   | K_5 = 144000          | C_5 = 909.1                 |
| M_6 = 4.17 | K_6 = 10000           | C_4 = 200                   |
| —          | K_7 = 166990          | C_7 = 310                   |

\[
\begin{align*}
M_1\ddot{Z}_1 &= -K_1(Z_1 - Z_0) - C_1 \sin \theta \left(\dot{Z}_1 - \dot{Z}_0\right) \\
&\quad + K_2(Z_2 - Z_1) + C_2 \left(Z'_2 - \dot{Z}_1\right) - FMR \sin \theta
\end{align*}
\]  
(8)
The ideal skyhook controller is approximated by the following control logic

\[
F_{\text{Sky}} = \begin{cases} 
    C_{\text{sky}} \left( \dot{Z}_1 - \dot{Z}_3 \right), & \text{if } \dot{Z}_1 \left( \dot{Z}_1 - \dot{Z}_3 \right) > 0 \\
    0, & \text{if } \dot{Z}_1 \left( \dot{Z}_1 - \dot{Z}_3 \right) \leq 0
\end{cases}
\]

Once the control output \( F_{\text{Sky}} \) is generated, the current input to be supplied to the hybrid MR damper is obtained based on

\[
I = \frac{2H_m}{N} \left[ F_{\text{Sky}} \frac{H_m}{8 \sin \phi \alpha L_m (A_p - A_s)} \right]^{1/\beta}
\]

where \( N \) is the number of coils turn, \( Lm \) is the MR orifice length, and \( Hm \) is the width of MR gap.

**Simulation results and discussions**

All the results presented in this paper were simulated via MATLAB/Simulink software, and both uncontrolled and controlled results are compared. To assess the ride comfort of the driver, the displacement of seat, head, and their respective accelerations are provided. During simulation, two types of road excitation, bump and random roads were used. In this work, the input disturbance to the seat system is taken as \( X_s \) (sprung displacement response), while \( X_r \) is taken as road input to quarter bus model. However, in most of the previous research studies about seat suspension (\( X_r = X_s \)) have been applied.

**Case 1:** bump road excitation

In this case, the pavement profile road \( X_r(S) \) can be obtained from differential equation as follows

\[
X_r = a (1 - \cos(\omega_r t))
\]

where \( a = 0.035 \text{ m} \) is the half bump amplitude, \( \omega_r = 2\pi V/D \) (\( D = 0.8 \text{ m} \)) is the bump width. \( V \) is the vehicle speed. At the bump, the vehicle speed was taken as \( V = 0.856 \text{ m/s} \).

**Case 2:** random road excitation

In this case, the pavement profile road \( X_r(S) \) can be obtained from differential equation as follows

\[
M_2 \ddot{Z}_2 = -K_2 (Z_2 - Z_1) - C_2 \left( \dot{Z}_2 - \dot{Z}_1 \right) + K_3 (Z_3 - Z_2) + C_3 \left( \dot{Z}_3 - \dot{Z}_2 \right) \tag{9}
\]

\[
M_3 \ddot{Z}_3 = -K_2 (Z_3 - Z_2) - C_2 \left( \dot{Z}_3 - \dot{Z}_2 \right) + K_4 (Z_4 - Z_3) + C_4 \left( \dot{Z}_4 - \dot{Z}_3 \right) \tag{10}
\]

\[
M_4 \ddot{Z}_4 = -K_4 (Z_4 - Z_3) - C_4 \left( \dot{Z}_4 - \dot{Z}_3 \right) + K_5 (Z_5 - Z_4) + C_5 \left( \dot{Z}_5 - \dot{Z}_4 \right) \tag{11}
\]

\[
M_5 \ddot{Z}_5 = -K_5 (Z_5 - Z_3) - C_5 \left( \dot{Z}_5 - \dot{Z}_3 \right) - K_6 (Z_6 - Z_5) + C_6 \left( \dot{Z}_6 - \dot{Z}_5 \right) \tag{12}
\]

\[
M_6 \ddot{Z}_6 = -K_7 (Z_6 - Z_5) - C_7 \left( \dot{Z}_6 - \dot{Z}_5 \right) \tag{13}
\]
where $\delta(t)$ is a white random process with a PSD of $2\alpha V^2$. The velocity $V = 72$ km/h is assumed. The value of $\alpha = 0.127 \text{ m}^{-1}$ and $\sigma^2 = 300 \text{ mm}^2$.

The bump response for seat displacement is shown in Figure 7, by comparing peak to peak values, and the controlled responses for both proposed and conventional models are slightly high compared to uncontrolled response. However, they have a fast settling time. This might be caused by displacement of sprung mass which was taken as input to seat system. Also, the proposed model seems to amplify the output displacement response than conventional MR damper. On the other hand, the skyhook controller has managed to suppress all the unpleasant vibration supplied to the driver seat. Figure 8(a) and (b) show the acceleration response for seat frame and driver head acceleration, and it is clearly seen that the proposed model has performed well than the conventional MR damper and uncontrolled responses in terms of both peak to peak values and RMS values. Figure 8(c) represents acceleration response for seat cushion, and it is also shown that the controlled response by skyhook controller for proposed model is better than that of conventional MR damper and uncontrolled one.

The bump responses were compared in term of root mean square (RMS), and all their values are summarized in Table 3. Figure 9 shows the acceleration responses in frequency domain for seat frame, driver head, and seat cushion in the range of 0–20 Hz. It is clearly evident that the lowest peak resonances for seat and driver head were achieved by skyhook controller where proposed model performed well than conventional MR damper. Also, frequency domain responses clearly showed how the controller attenuated both seat and driver head energy from the sprung displacement and improve the ride comfort.

For the case of random vibration, the acceleration and frequency domain for uncontrolled, HMR, and MR controlled responses were illustrated in Figures 10 and 11, respectively. In this case also, the controlled hybrid MR damper outperformed well the conventional MR damper in terms of both attenuate the vibration and minimize the energy transferred to the seat frame and driver body. For better visualization of how all responses performed, the RMS values and percentage reduction for accelerations were calculated and presented in Table 4. The controlled MR damper has reduced the RMS by 10.42% for seat frame, 15.28% for driver head, and 12.6% for seat cushion compared to uncontrolled responses. On the other hand, the controlled HMR damper compared to uncontrolled responses has managed to reduce the RMS by 14.58% for seat frame, 18.05% for driver head, and 16.53% for seat cushion. Figure 11 shows the frequency domain for acceleration responses and they revealed that the highest peaks occur around the frequency of 2 Hz and diminished around 3.8 Hz. The frequency domain results clearly show that the controlled hybrid MR damper responses attenuate the peak acceleration better than conventional MR damper.

**Whole body vibration analysis**

To evaluate the driver body exposure to vibration and how bad those vibrations might affect driver health, ISO 2631 (International standard for evaluating human exposure to WBV) is applied to evaluate the vertical motion of seat...
Figure 8. Acceleration responses in time domain for bump road.
Table 3. RMS value and Percentage improvement for bump road.

|                | Performance evaluation (RMS) |
|----------------|-------------------------------|
| Acceleration (m/s²) | Uncontrolled | Proposed model | Conventional model | HMR % reduction | MR % reduction |
| Seat frame      | 0.78            | 0.60            | 0.66              | 23.08            | 15.38            |
| Driver head     | 0.65            | 0.46            | 0.51              | 29.23            | 21.54            |
| Seat cushion    | 1.05            | 0.75            | 0.84              | 28.57            | 20.00            |

Figure 9. Acceleration responses in frequency domain for bump road. (a) Seat frame, (b) driver head, and (c) seat cushion.
Figure 10. Accelerations responses in time domain for random road. (a) Seat frame, (b) driver head, and (c) seat cushion.
Figure 11. Accelerations responses in frequency domain for random road. (a) Seat frame, (b) driver head, and (c) seat cushion.

Table 4. RMS values and percentage improvement for random road.

| Acceleration (m/s²) | Performance evaluation (RMS) | HMR % reduction | MR % reduction |
|---------------------|------------------------------|-----------------|----------------|
| Seat frame          | Uncontrolled: 0.96           | Proposed model: 0.82 | 0.86           | 14.58          | 10.42          |
| Driver head         | Uncontrolled: 0.72           | Proposed model: 0.59 | 0.61           | 18.05          | 15.28          |
| Seat cushion        | Uncontrolled: 1.27           | Proposed model: 1.06 | 1.11           | 16.53          | 12.60          |
Table 5. Average weighted acceleration and vibration dose value.

| Exposure parameters | Uncontrolled | Proposed model | Conventional model |
|---------------------|--------------|----------------|-------------------|
|                     | Base         | Seat frame     | Seat cushion      | Base         | Seat frame     | Seat cushion      | Base         | Seat frame     | Seat cushion      | Body         | Body         | Body         |
| $A_w^{(w,k,z)}$      | 0.45         | 0.57           | 0.80              | 0.5           | 0.43           | 0.56           | 0.36         | 0.48           | 0.63           | 0.40         |
| VDV$^{(w,k,z)}$      | 1.71         | 1.77           | 2.47              | 1.56         | 1.34           | 1.74           | 1.18         | 1.47           | 1.95           | 1.26         |

Figure 12. Comparison of whole body vibration performances. (a) Driver head, (b) seat frame, and (c) seat cushion.
suspension. The whole body exposure parameters evaluated are as follows: weighted RMS acceleration $A_w$ in (m/s²), the vibration dose value (VDV) in (m/s⁰.⁷⁵), and seat effective amplitude transmissibility (SEAT). The formulas used to calculate both $A_w$ and VDV are given below

\[
A_w = \left[ \frac{1}{T} \int_0^T A_w(t)^2 dt \right]^{1/2}
\]

\[
VDV = \left[ \frac{1}{T} \int_0^T A_w(t)^4 dt \right]^{1/4}
\]

where $A_w(t)$ is the frequency weighted acceleration and $T$ is the exposure time. The SEAT value is calculated using either $A_w$ or VDV values

\[
\text{SEAT} (%) = 100 \times \frac{A_w(\text{seat})}{A_w(\text{base})}
\]

\[
\text{SEAT} (%) = 100 \times \frac{VDV(\text{seat})}{VDV(\text{base})}
\]

When the value of SEAT is greater than 100%, this indicates that the seat is amplifying vibration. However, if the SEAT is less than 100%, the SEAT is attenuating vibration. To compute the exposure parameter values, we only considered vertical motion for our quarter bus model. The calculated values for evaluated WBV for both controlled proposed model and conventional model were compared to uncontrolled responses in Figure 12. It shows the WBV performance of the three parameters for driver head, seat frame, and seat cushion acceleration. The controlled values are labeled in brown and orange colors for conventional and proposed models, respectively. The uncontrolled one is labeled in blue color. It is clearly seen that the proposed model values for weighted acceleration ($A_w$) have significantly reduced by 24.6%, 28%, and 34.9% for seat frame, driver head, and seat cushion acceleration, while the conventional MR model suppresses the vibrations by 5.8%, 20%, and 21.25% for seat frame, driver head, and seat cushion, respectively. The SEAT values were calculated from both obtained $A_w$ and VDV values. It is obviously seen that the obtained SEAT values obtained from VDV are better than those from $A_w$. From Figure 12(a), the uncontrolled SEAT values for driver head acceleration are 111.1% and 91.2% for $A_w - \text{SEAT}$ and $VDV - \text{SEAT}$, respectively. On the other hand, the HMR controlled SEAT values for driver head are 80.0% and 69.0% for $A_w - \text{SEAT}$ and $VDV - \text{SEAT}$. While for MR controlled SEAT, driver head values were 88.9% and 73.68% for $A_w - \text{SEAT}$ and $VDV - \text{SEAT}$, respectively. The obtained results indicate how the controlled semi-active system might attenuate better the vibrations induced by the seat frame to the driver. It is clearly seen that the uncontrolled responses are

Table 6. SEAT value percentages calculated based on $A_w$ and VDV.

| SEAT (%) | Uncontrolled | Proposed model | Conventional model |
|----------|-------------|----------------|--------------------|
|          | Seat frame | Seat cushion | Body               | Seat frame | Seat cushion | Driver head | Seat frame | Seat cushion | Body               |
| $A_w - \text{SEAT}$ | 126.66     | 177.78       | 111.11             | 95.55      | 124.44       | 80.0        | 106.67     | 140.0        | 88.89             |
| $VDV - \text{SEAT}$ | 103.51     | 144.44       | 91.23              | 83.6       | 101.75       | 69.0        | 85.96      | 114.03       | 73.68             |

Note: SEAT: seat effective amplitude transmissibility; VDV: vibration dose value.

Table 7. Scale of discomfort suggested by ISO 2631.

| Weighted vibration magnitude (m/s²) | Likely reaction in public transport |
|------------------------------------|-----------------------------------|
| Less than 0.315                    | Not uncomfortable                   |
| 0.315–0.63                         | Little uncomfortable                |
| 0.5–1                              | Fairly uncomfortable                |
| 0.8–1.6                            | Uncomfortable                      |
| 1.25–2.5                           | Very uncomfortable                  |
| Greater than 2                     | Extremely uncomfortable             |

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worse before introducing a controller. Figure 12(c) shows the SEAT values for seat cushion acceleration, and it is clear seen that all controlled and uncontrolled responses were amplified due to the low stiffness of cushion material. All the detailed values calculated for WBV are mentioned in Tables 5 and 6, respectively. From Table 5, we can conclude that the car driver will suffer no discomfort based on the obtained values and compare them to the scale of discomfort from ISO 2631 which is summarized in Table 7.

Conclusion

In this work, the hybrid MR damper for vehicle seat suspension was proposed, and its effectiveness was compared to conventional MR damper through MATLAB software. At higher current of 1A, the damping force obtained was higher in hybrid MR damper than in conventional one. Semi-active seat for ¼ bus coupled with human model featuring hybrid MR damper was modeled and skyhook controller was used to control the vertical vibration of the system. For the case of bump input road, the HMR damper was proven to be effective than conventional MR damper. Similarly, to the case of random road, the hybrid MR damper outperformed the conventional MR damper in terms of suppressing the unwanted vibrations. Also, for both cases of controlled HMR and MR responses, there was a great improvement in ride comfort than in uncontrolled responses for both bumpy and random roads. To ensure that the vibrations exposed to the driver are not harmful, the WBV parameters were calculated and the little discomfort to the driver was confirmed based on the discomfort chart by ISO 2631 (1997). SEAT values revealed that the uncontrolled system amplifies the acceleration, while the skyhook controller attenuates the acceleration responses.

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