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Vibration control of electrorheological seat suspension with human-body model using sliding mode control

Seung-Bok Choi*, Young-Min Han

Smart Structures and Systems Laboratory, Department of Mechanical Engineering, Inha University, Incheon 402-751, Republic of Korea

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Abstract

This paper presents vibration control performance of a semi-active electrorheological (ER) seat suspension system using a robust sliding mode controller (SMC). A cylindrical type of ER seat damper is manufactured for a commercial vehicle seat suspension and its field-dependent damping force is experimentally evaluated. A vertical vibration model of human-body is then derived and integrated with the governing equations of the ER seat suspension system. The integrated seat-driver model featured by a high order degree-of-freedom (dof) is reduced through a balanced model reduction method. The SMC is then designed based on the reduced model and the state observer is formulated to estimate feedback states which cannot be directly measured from sensors. By imposing a semi-active actuating condition, the synthesized SMC is experimentally realized. In the experimental implementation, a driver directly sits on the controlled seat. Both vertical displacement and acceleration are measured at seat frame and driver's head, respectively. Control performances are evaluated under various road conditions and compared with those obtained from conventional passive seat suspension system.

1. Introduction

Commercial vehicle drivers are typically exposed to extended driving hours and severe working environment. Recently, many people have focused their attention on the ride quality of vehicle which is directly related to driver fatigue, discomfort, and safety. Although many efforts to improve the ride quality have been made through primary and secondary (cabin) suspensions [1–5], drivers exposure to low-frequency and high-amplitude vibration dominant in commercial vehicles, which is a major factor in health disorders [6,7]. Thus, seat suspension has been employed as a simple and effective method to attenuate unwanted vibration. Moreover, the seat suspension should be particularly effective in the low frequency range because the seat vibration energy is concentrated at low frequency below 10 Hz, and to which human-body is very sensitive. Therefore, the transmitted vibration energy needs to be effectively alleviated by installing semi-active or active seat suspension systems with control scheme considering a mathematical model of the human-body.

*Corresponding author. Tel.: +82 32 860 7319; fax: +82 32 868 1716.
E-mail address: seungbok@inha.ac.kr (S.-B. Choi).
URL: http://www.ssslab.com (S.-B. Choi).

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Recently, a semi-active seat suspension system featuring electrorheological (ER) fluids has been investigated for vibration isolation with various approaches such as control algorithm and nonlinear vehicle model [8–11]. This type of the semi-active seat suspension system has several advantages such as fast response time, continuously controllable damping force and low energy consumption compared with active seat suspension systems. However, most of previous researches for the ER seat suspension system have been limited their scope to vibration isolation of a rigid dummy mass on the seat [8,9]. The research on the ride quality of the ER seat suspension considering human vibration model is considerably rare. The seat suspension dynamics with human-body model strongly influence ride discomfort and safety. Therefore, a mathematical model of the human-body should be considered for controller synthesis to enhance vibration attenuation performance of the seat suspension.

The seated human-body exposed to vibration is a complex dynamic system. So far, various biomechanical models have been developed to describe the human vibration motion such as ISO CD 5982 model [12] and others [13,14]. These lumped parameter models consider the human-body as several rigid bodies, springs, and dampers. Moreover, the models are featured by a high order degree-of-freedom (dof), and various uncertainties nevertheless remain. Therefore, the model should be reduced by considering seat vibration in frequency domain, and it needs a robust control algorithm such as sliding mode controller (SMC), which is very robust to external disturbances and parameter uncertainties.

The main contribution of this paper is to show vibration control performance of an ER seat suspension system via SMC. The proposed seat suspension system is featured by semi-active ER damper and vibration model of human-body. In order to achieve this goal, a cylindrical ER seat damper is devised and its damping force is experimentally evaluated with respect to the intensity of the input electric field. The ER seat damper is then installed in the commercial seat suspension system by replacing a conventional passive oil damper. A 4-dof human-body model is adopted and integrated with the governing equation of the seat suspension, which consists of seat, spring, and ER damper. A reduced seat-driver model is then derived via a balanced model reduction method. A sliding model control algorithms with state observer is designed to attenuate unwanted vibration and experimentally realized where a person is sitting on the seat. Based on the measured displacement and acceleration at seat frame and driver's head, respectively, vibration control responses of the proposed semi-active ER seat suspension are evaluated in both time and frequency domains under bump and random road conditions.

2. ER seat damper

A cylindrical type of ER seat damper is designed and manufactured on the basis of the required damping force level of a conventional seat damper for a commercial truck. An ER fluid was synthesized using arabic gum and silicone oil as particles and base liquid, respectively. In general, ER fluid behaves as the Bingham fluid [15] whose constitutive equation is given by

$$\tau = \eta\dot{\gamma} + \tau_y(E), \quad \tau_y(E) = \alpha E^{\beta}, \tag{1}$$

where τ is the shear stress [Pa], η is the dynamic viscosity [Pa s], $\dot{\gamma}$ is the shear rate [s^{-1}] and $\tau_y(E)$ is the dynamic yield stress of the ER fluid. It has been proved that $\tau_y(E)$ is a function of the electric field E [kV/mm] and exponentially increases with respect to the electric field. The parameters of α and β are characteristic values of the ER fluid to be determined by experiment. The viscosity of the silicone oil used in this work is 0.09 Pa s and the weight ratio of the particles to the fluid is 30%. The yield stress is evaluated by $24.4E^{2.07}$ Pa at room temperature. This experimentally obtained Bingham model is to be incorporated with the damping force model of the proposed ER seat damper.

The ER seat damper proposed in this study is shown in Fig. 1. The ER seat damper is divided into the upper and lower chambers by the piston, and it is fully filled with the ER fluid. By the motion of the piston, the ER fluid flows through the duct between inner and outer cylinders from one chamber to the other. The gas chamber located outside of the lower chamber acts as an accumulator of the ER fluid induced by the motion of the piston. In order to simplify the analysis of the ER seat damper, it is assumed that the ER fluid is incompressible and the pressure in the gas chamber is uniformly distributed. Furthermore, it is assumed that frictional force between oil seals and fluid inertia is negligible. For laminar flow in electrode gap, the fluid

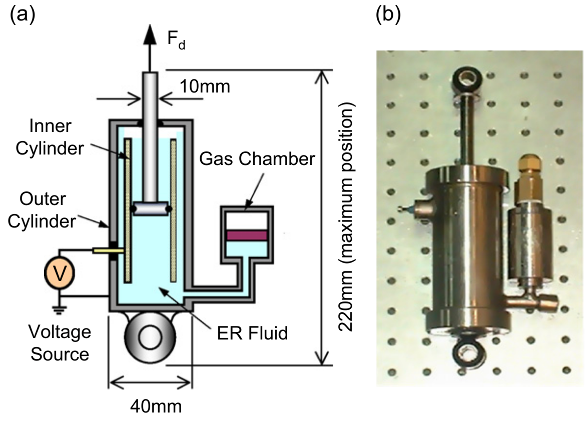

Fig. 1. The proposed ER seat damper.

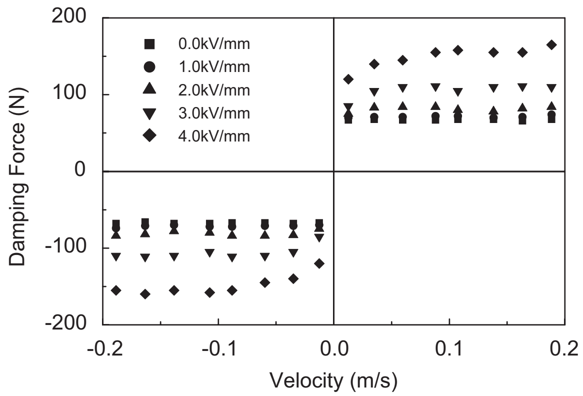

Fig. 2. Damping force characteristics of the proposed ER seat damper.

resistance from the duct is given by

$$R_e = \frac{12\eta l}{bh^3}, \tag{2}$$

where l is the electrode length, b is the electrode width, and h is the electrode gap. On the other hand, the pressure drop due to the increment of the yield stress of the ER fluid is given as

$$P_{\mathrm{ER}} = 2\frac{l}{h}\tau_y(E) = 2\frac{l}{h}\alpha E^{\beta}. \tag{3}$$

Now, the damping force of the ER seat damper is obtained by

$$F_d = \frac{A_r^2}{C_g}x_p + (A_p - A_r)^2 R_e \dot{x}_p + (A_p - A_r)P_{\mathrm{ER}}\,\mathrm{sgn}(\dot{x}_p), \tag{4}$$

where x_p and $\dot{x}_p$ are the excitation displacement and velocity, respectively, C_g is the gas compliance of the gas chamber, A_r is the piston-rod area, A_p is the piston area, and sgn$(\cdot)$ is the signum function. It is noted that the third term in Eq. (4) is the controllable damping force as a function of the intensity of the electric field. The damping force given by Eq. (4) can be rewritten by

$$F_d = k_e x_p + c_e \dot{x}_p + F_{\mathrm{ER}}, \tag{5}$$

where k_e is the effective stiffness due to the gas pressure, and c_e is the effective damping coefficient due to the viscosity of the ER seat damper.

The size and level of required damping force adopted in this work are determined on the basis of a conventional passive oil seat damper for a commercial truck. The damping force given by Eq. (5) is analyzed with respect to the electric field in order to determine principal design parameters. The designed and manufactured ER seat damper has the following geometric specifications: electrode length; 85 mm; electrode width: 122 mm; electrode gap: 0.95 mm and diameter of the piston head: 34 mm.Fig. 2 presents the measured damping force characteristics of the proposed ER seat damper. It is identified that the damping force of 62 N in the absence of the input field is increased up to 156 N by applying the input electric field of 4 kV/mm. It is clearly observed that the damping force increases as the input electric field increases. This directly indicates that a certain desired damping force can be achieved by controlling the input electric field regardless of the piston velocity.

3. Dynamic model of ER seat suspension with driver

A seat-driver model consisting of the semi-active ER seat suspension and human-body is shown in Fig. 3. The excitation input from the road is transmitted to the cabin floor. For the simplification of the dynamic modeling, it is assumed that there exists only the vertical motion of the vehicle. Both pitching and rolling motions are ignored in this study. As shown in Fig. 3, the buttocks with thigh of the human-body contact with the seat cushion, and their connection force can be obtained as follows:

$$F_h = c_3(\dot{x}_3 - \dot{x}_2) + k_3(x_3 - x_2), \tag{6}$$

where k_3 and c_3 are the spring constant and the damping coefficient of buttocks with thigh, respectively.

The seat suspension system consists of the ER seat damper, linear spring, seat frame and cushion. The spring constant is held to be constant and the damping force varies according to control input. The seat cushion is modeled by a set of linear springs and dampers whose characteristics are constant. Therefore, the governing equation of motion of the ER seat suspension can be obtained as follows:

$$\begin{aligned} m_1\ddot{x}_1 &= -c_1\dot{x}_1 - k_1x_1 + c_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) - U + E_p, \\ m_2\ddot{x}_2 &= -c_2(\dot{x}_2 - \dot{x}_1) - k_2(x_2 - x_1) + F_h, \end{aligned} \tag{7}$$

where m_1 and m_2 are the mass of the seat suspension and cushion, k_1 and c_1 are the spring constant and damping coefficient of the suspension, and k_2 and c_2 are the spring constant and damping coefficient of the cushion, respectively. From the ER seat damper model, it can be easily obtained that $k_1 = k_s + k_e$ where k_s is the spring constant of the suspension spring, and $c_1 = c_s + c_e$ where c_s is the effective damping coefficient of the suspension frame. U represents controllable damping force (F_{ER}), and F_h is connection force between seat cushion and human-body. E_p is calculated by $c_1\dot{x}_0 + k_1x_0$, where x_0 is excitation input. It is noted that the coefficients of springs and dampers adopted in this work were chosen based on parameter values of commercial truck seat.

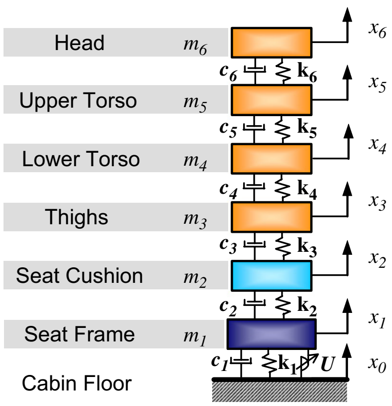

Fig. 3. Vibration model of the ER seat suspension system with the human-body.

The driver is modeled as a multi-segment mass composed of four human-body segments which include head, upper torso, lower torso and thighs. The arms and legs are combined with the upper torso and thigh, respectively [16]. A mathematical model of the human-body can be obtained as follows:

$$\begin{aligned} m_3\ddot{x}_3 &= -F_h + c_4(\dot{x}_4 - \dot{x}_3) + k_4(x_4 - x_3), \\ m_4\ddot{x}_4 &= -c_4(\dot{x}_4 - \dot{x}_3) - k_4(x_4 - x_3) + c_5(\dot{x}_5 - \dot{x}_4) + k_5(x_5 - x_4), \\ m_5\ddot{x}_5 &= -c_5(\dot{x}_5 - \dot{x}_4) - k_5(x_5 - x_4) + c_6(\dot{x}_6 - \dot{x}_5) + k_6(x_6 - x_5), \\ m_6\ddot{x}_6 &= -c_6(\dot{x}_6 - \dot{x}_5) - k_6(x_6 - x_5), \end{aligned} \tag{8}$$

where m_3, m_4, m_5 and m_6 are the mass of the thighs, lower torso, upper torso and head, respectively. k_4, k_5 and k_6 are the spring constant of lumbar spine, thoracic spine and cervical spine, respectively. c_4, c_5 and c_6 are the corresponding damping coefficient, respectively. Table 1 shows system parameters adopted in this study.

We describe interior forces between human-body segments with springs and dampers whose coefficients are obtained from Ref. [17], and the forces are adjusted to the body of the person who takes part in the experiment. The segmented masses are also determined by the same manner. By considering connection force (6), Eq. (8) is incorporated with the dynamic model (7) of the seat suspension system. Thus, by defining the state vector as $\mathbf{X} = [x_1 \;\; \dot{x}_1 \;\; x_2 \;\; \dot{x}_2 \;\; x_3 \;\; \dot{x}_3 \;\; x_4 \;\; \dot{x}_4 \;\; x_5 \;\; \dot{x}_5 \;\; x_6 \;\; \dot{x}_6]^{\mathrm{T}}$, the seat-driver model can be obtained in a state space form as follows:

$$\begin{aligned} \dot{\mathbf{X}} &= \mathbf{A}\mathbf{X} + \mathbf{B}U + \mathbf{D}E_p, \\ \mathbf{Y} &= \mathbf{C}\mathbf{X}, \end{aligned} \tag{9}$$

where

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ a_{21} & a_{22} & a_{23} & a_{24} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ a_{41} & a_{42} & a_{43} & a_{44} & a_{45} & a_{46} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & a_{63} & a_{64} & a_{65} & a_{66} & a_{67} & a_{68} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & a_{85} & a_{86} & a_{87} & a_{88} & a_{89} & a_{8,10} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & a_{10,7} & a_{10,8} & a_{10,9} & a_{10,10} & a_{10,11} & a_{10,12} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & a_{12,9} & a_{12,10} & a_{12,11} & a_{12,12} \end{bmatrix},$$

$$a_{21} = -\frac{k_1 + k_2}{m_1}, \quad a_{22} = -\frac{c_1 + c_2}{m_1}, \quad a_{23} = \frac{k_2}{m_1}, \quad a_{24} = \frac{c_2}{m_1},$$

Table 1
System parameters of the proposed ER seat suspension

Mass (kg)		Damping coefficient (Ns/m)		Spring constant (N/m)	
m_1	15	c_1	830	k_1	31000
m_2	1	c_2	200	k_2	18000
m_3	12.78	c_3	2064	k_3	90000
m_4	8.62	c_4	4585	k_4	162800
m_5	28.49	c_5	4750	k_5	183000
m_6	5.31	c_6	400	k_6	310000

$$a_{41} = \frac{k_2}{m_2}, \quad a_{42} = \frac{c_2}{m_2}, \quad a_{43} = -\frac{k_2 + k_3}{m_2}, \quad a_{44} = -\frac{c_2 + c_3}{m_2}, \quad a_{45} = \frac{k_3}{m_2}, \quad a_{46} = \frac{c_3}{m_2},$$

$$a_{63} = \frac{k_3}{m_3}, \quad a_{64} = \frac{c_3}{m_3}, \quad a_{65} = -\frac{k_3 + k_4}{m_3}, \quad a_{66} = -\frac{c_3 + c_4}{m_3}, \quad a_{67} = \frac{k_4}{m_3}, \quad a_{68} = \frac{c_4}{m_3},$$

$$a_{85} = \frac{k_4}{m_4}, \quad a_{86} = \frac{c_4}{m_4}, \quad a_{87} = -\frac{k_4 + k_5}{m_4}, \quad a_{88} = -\frac{c_4 + c_5}{m_4}, \quad a_{89} = \frac{k_5}{m_4}, \quad a_{8,10} = \frac{c_5}{m_4},$$

$$a_{10,7} = \frac{k_5}{m_5}, \quad a_{10,8} = \frac{c_5}{m_5}, \quad a_{10,9} = -\frac{k_5 + k_6}{m_5}, \quad a_{10,10} = -\frac{c_5 + c_6}{m_5}, \quad a_{10,11} = \frac{k_6}{m_5}, \quad a_{10,12} = \frac{c_6}{m_5},$$

$$a_{12,9} = \frac{k_6}{m_6}, \quad a_{12,10} = \frac{c_6}{m_6}, \quad a_{12,11} = -\frac{k_6}{m_6}, \quad a_{12,12} = -\frac{c_6}{m_6},$$

$$\mathbf{B} = \begin{bmatrix} 0 & -\frac{1}{m_1} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}^{\mathrm{T}},$$

$$\mathbf{C} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{k_5}{m_6} & \frac{c_5}{m_6} & -\frac{k_6}{m_6} & -\frac{c_6}{m_6} \end{bmatrix},$$

$$\mathbf{D} = \begin{bmatrix} 0 & \frac{1}{m_1} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}^{\mathrm{T}}.$$

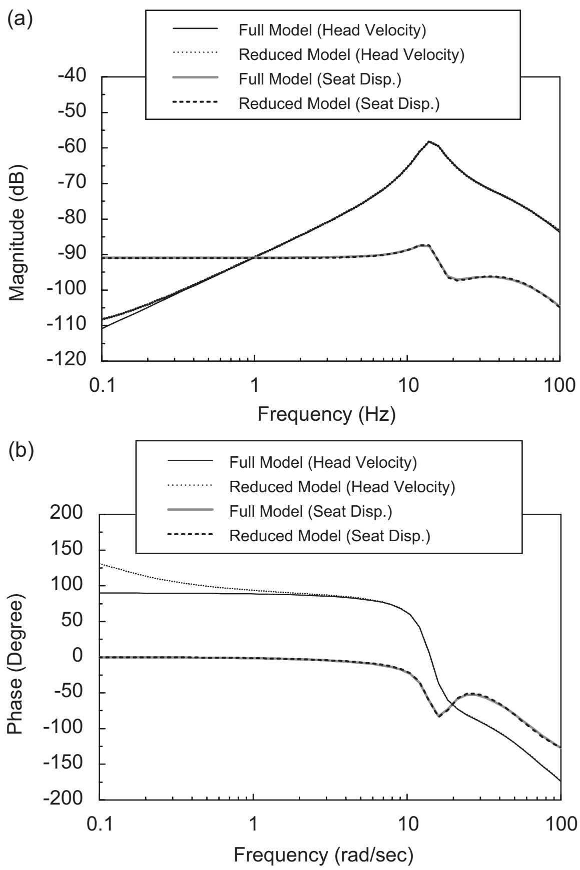

Fig. 4. Frequency–response characteristics of the reduced seat-driver model.

In this work, the order of the developed full model (9) is 12, which is relatively high and makes it difficult to design a controller. Therefore, a simple control system can be obtained by the balanced model reduction method [18]. The balanced realization of the system (8) is then given by

$$\mathbf{G} = \begin{bmatrix} \hat{\mathbf{A}} & \hat{\mathbf{B}} \\ \hat{\mathbf{C}} & 0 \end{bmatrix} = \begin{bmatrix} \mathbf{TAT}^{-1} & \mathbf{T}\begin{bmatrix} \mathbf{B} & \mathbf{D} \end{bmatrix} \\ \mathbf{CT}^{-1} & 0 \end{bmatrix} \tag{10}$$

where $\mathbf{T}$ is the similarity transformation matrix by which the state vector $\mathbf{X}$ is transformed to the new state vector $\mathbf{X}_r = \mathbf{TX}$. The realization then gives the balanced Gramian for controllability and observability. By investigating the Hankel singular values ($\sigma_1 > \sigma_2 > \cdots \sigma_{12}$) of the balanced Gramian, the less controllable and less observable states corresponding to $\sigma_7, \cdots \sigma_{12}$ can be truncated which do not affect the degree of loss of both controllability and observability of the system (9). By defining the reduced state vector $\mathbf{X}_r = [x_{r1}\ x_{r2}\ x_{r3}\ x_{r4}\ x_{r5}\ x_{r6}]^{\mathrm{T}}$, the following reduced six order model is obtained:

$$\dot{\mathbf{X}}_r = \mathbf{A}_r\mathbf{X}_r + \mathbf{B}_r U + \mathbf{D}_r E_p, \quad \mathbf{Y} = \mathbf{C}_r\mathbf{X}_r. \tag{11}$$

It is noted that $\mathbf{Y} = [\,x_1 \quad \ddot{x}_6\,]$ are directly measured. The specific values of the matrices for the proposed ER seat suspension system are in Ref. [19]. Fig. 4 compares the frequency responses between the twelve order full model (9) and the reduced six-order model (11), in which input is the damping force produced by ER damper whereas output is driver's head velocity and seat displacement. It is clearly seen that the accuracy of the reduced model is fairly acceptable for both head velocity and seat displacement.

4. Controller formulation

The impending control issue is to achieve vibration suppression considering human vibration by using ER seat suspension system. Thus, a structured control algorithm based on the seat-driver model needs instead of the conventional semi-active skyhook controller. Furthermore, as mentioned earlier, there exist parameter uncertainties in the human vibration model. This requires to use robust control schemes such as SMC which is very effective for nonlinear and parameter uncertain system [20]. Fig. 5 shows the proposed control block-diagram. The first step to formulate the SMC is to define a sliding surface which guarantees stable sliding mode motion on it. In this work, the following sliding surface is defined which is asymptotically stable:

$$\begin{aligned} s &= g_1x_{r1} + g_2x_{r2} + g_3x_{r3} + g_4x_{r4} + g_5x_{r5} + g_6x_{r6} \\ &= \mathbf{gX}_r, \end{aligned} \tag{12}$$

where $\mathbf{g}$ is the gradient matrix. Then, a sliding mode exists on the sliding surface whenever the following sliding mode condition is satisfied:

$$s\dot{s} < 0. \tag{13}$$

Prior to formulating the SMC, parameter variations of a driver are imposed as follows:

$$m_i = m_{i0} + \delta m_i, \quad |\delta m_i| < \alpha m_{i0}, \quad i = 3, 4, 5, 6, \tag{14}$$

where m_i is the mass of each segment of human-body such as thighs, lower torso, upper torso and head, and δm_i is corresponding variation. After incorporating parameter variations in Eq. (14) into the governing

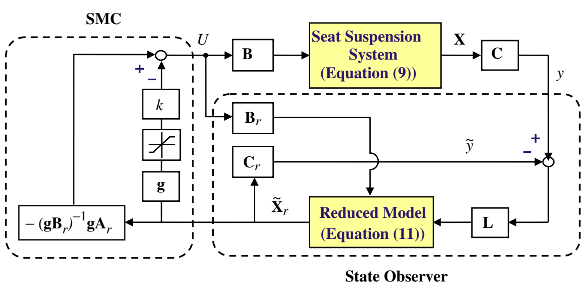

Fig. 5. Block-diagram of the proposed sliding mode control.

equation (9), it is reduced to the six-order model (11). The sliding mode condition (13) is then applied to the reduced system (11). Consequently, the SMC is obtained as follows:

$$
\begin{aligned}
U &= -\frac{(g_1 a_{r11} + \cdots + g_6 a_{r61})x_{r1} + \cdots + (g_1 a_{r16} + \cdots + g_6 a_{r66})x_{r6}}{(g_1 b_{r11} + \cdots + g_6 b_{r61})} - k\,\mathrm{sgn}\,(s) \\
&= -(\mathbf{gB}_r)^{-1}\mathbf{gA}_r\mathbf{X}_r(t) - k\,\mathrm{sgn}\,(s),
\end{aligned} \tag{15}
$$

where k is the discontinuous gain. Now, it can be shown that the control system in Eq. (11) with the proposed SMC in Eq. (15) satisfies the sliding mode condition in Eq. (13). Its detailed procedure for the controller design can be found in Ref. [19]. As mentioned earlier, the state $\mathbf{X}_r$ is to be estimated by the state $\tilde{\mathbf{X}}_r$ of the state observer. The observer based SMC can be expressed as follows:

$$
\begin{aligned}
U(t) &= -(\mathbf{gB}_r)^{-1}\mathbf{gA}_r\tilde{\mathbf{X}}_r(t) - k\,\mathrm{sgn}(\tilde{s}), \\
\dot{\tilde{\mathbf{X}}}_r(t) &= \mathbf{A}_r\tilde{\mathbf{X}}_r(t) + \mathbf{B}_r U(t) + \mathbf{L}(y_1 - \mathbf{C}_r\tilde{\mathbf{X}}_r(t)), \\
\tilde{s} &= \mathbf{g}\tilde{\mathbf{X}}_r(t),
\end{aligned} \tag{16}
$$

where $\mathbf{L}$ is the observer gain matrix. In practice, it is not desirable to use the discontinuous control law in Eq. (16) due to the chattering. Therefore, the discontinuous control law is normally approximated by a continuous one inside boundary layer width. This is accomplished by replacing sgn(s) in Eq. (16) by a saturation function [20]. In this study, $U(t)$ is the controllable damping force (F_{ER}) of the seat suspension system. It needs to be controlled depending upon the motion of the suspension travel. Therefore, the following actuating condition is normally imposed.

$$
U = \begin{bmatrix} F_{\mathrm{ER}}, & \text{for } F_{\mathrm{ER}}\,(\dot{x}_1 - \dot{E}_e) > 0 \\ 0, & \text{for } F_{\mathrm{ER}}\,(\dot{x}_1 - \dot{E}_e) \leqslant 0 \end{bmatrix}. \tag{17}
$$

The above actuating conditions physically imply that the activating of the controller only assures the increment of energy dissipation of the stable system. Once the control input is determined, the input electric field to be applied to the ER seat damper is obtained as follows.

$$
E = \left[\frac{U}{A_p - A_r}\frac{h}{2L\alpha}\right]^{1/\beta}. \tag{18}
$$

5. Performance evaluation and discussion

In order to evaluate control performance of the proposed system, an experimental setup is established as shown in Fig. 6. A driver directly sits on the controlled seat, and the floor under the seat is excited from the hydraulic power unit with two types of profiles; bump and random road conditions. The excited floor displacement is calculated and then the displacement signal is converted to the hydraulic control unit for controlling the excitation input to the platform. When the vibration is transmitted to the ER seat suspension, the dynamic response signals, which are acquired from the linear variable differential transformer (LVDT) and accelerometer, are then converted to digital signals via a 12bits A/D converter. The SMC given by Eq. (16) is then activated in the microprocessor to reduce the vibration level at the driver seat. In this study, gradient matrix (**g**) of sliding surface is determined by [120 −3 15 1 −1 800], and discontinuous k gain is 250.

In order to evaluate performances of a suspension system, a random profile is normally used to evaluate the frequency response such as power spectral density (PSD) whereas a bump profile is used to reveal the transient response characteristic. In this work, ride quality evaluation is performed by using fatigue-decreased-proficiency boundary, vibration dose value (VDV) and crest factor instead of PSD. First of all, the transmissibility of the seat-driver is firstly measured. The transmissibility is highly dependent on the mechanical impedance of the human-body. The isolation efficiency of the seat suspension called seat effective amplitude transmissibility (SEAT) [21] is then calculated with root mean square (rms) value of acceleration, which compares the vibration severity on the seat with on the floor. VDV is also calculated, which is more

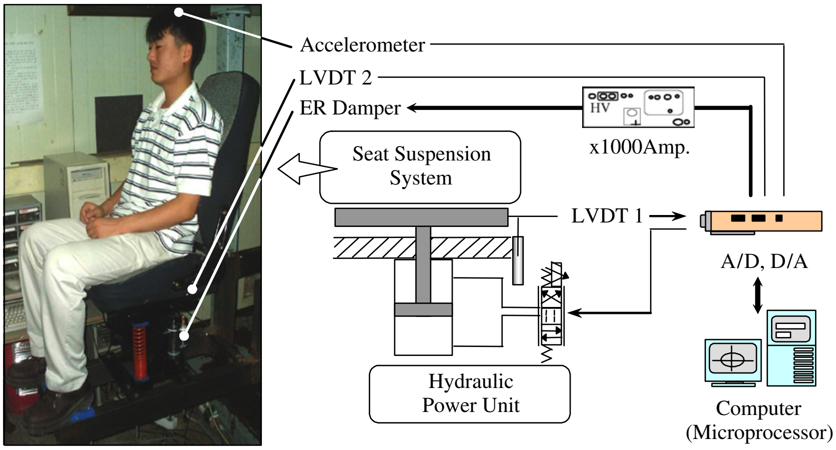

Fig. 6. Experimental configuration of the proposed ER seat suspension system.

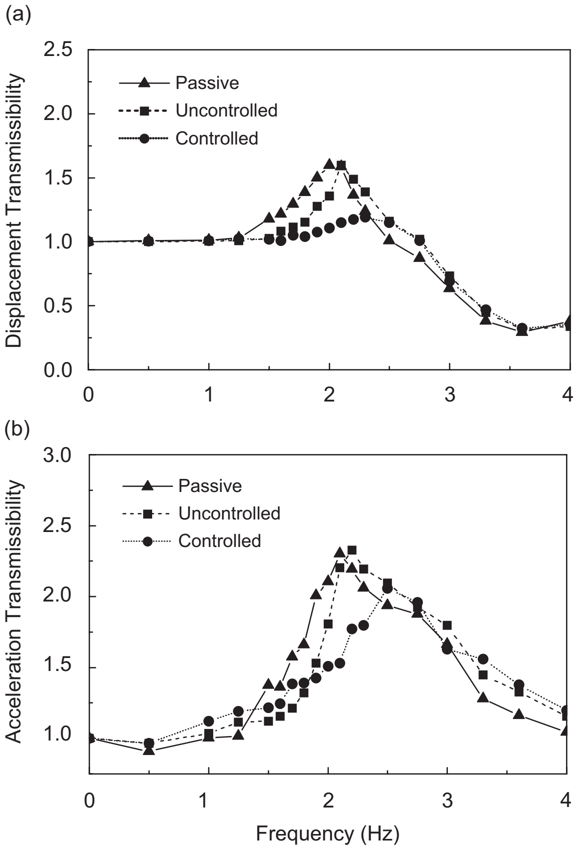

Fig. 7. Transmissibility of the ER seat suspension system.

sensitive to peak. In conformity with the parameters defined in ISO 2631-1 [22] for assessing the human-body exposure to vibration, control performance evaluation of the proposed seat suspension is performed on the basis of the time history of the frequency-weighted value in order to consider vibration sensitivity of human-body. The SEAT value and VDV ratio are obtained as follows:

$$a_w = \left[\frac{1}{T}\int_0^T \{a_w(t)\}^2\,\mathrm{d}t\right]^{1/2}, \quad \mathrm{SEAT} = \frac{a_{w,\mathrm{seat}}}{a_{w,\mathrm{base}}},$$
$$\mathrm{VDV} = \left[\int_{t=0}^{t=T} a_w^4(t)\,\mathrm{d}t\right]^{1/4}, \quad \mathrm{VDV\ ratio} = \frac{\mathrm{VDV}_{\mathrm{seat}}}{\mathrm{VDV}_{\mathrm{floor}}}, \tag{19}$$

where $a_w(t)$ is the time history of the frequency-weighted acceleration and T is the duration of measurement. The severity of the vibration exposures causes discomfort, pain and injury. However, it is generally assumed that the total exposure rather than average exposure is important. In this work, the ride quality is evaluated using the frequency-weighted rms acceleration at driver's head, and compared with the vibration fatigue-decreased proficiency boundary defined in ISO 2631-1 [22]. Furthermore, the crest factor is evaluated to show how much impact is occurring in the vibration because the impact can cause discomfort of driver. The crest factor is mathematically expressed as follows:

$$f_c = \frac{[a_w(t)]_{\max}}{a_w}. \tag{20}$$

In this work, two road profiles are adopted to evaluate the proposed system. The bump profile is firstly used to reveal the transient response characteristic, which is described by

$$z = z_b[1 - \cos(w_r t)], \tag{21}$$

where $w_r = 2\pi V_s/D$ and z_b is the half of the bump height. D is the width of the bump and V_s is the vehicle velocity. In the bump excitation, the vehicle travels the bump with constant velocity of 3.08 km/h (= 0.856 m/s). The

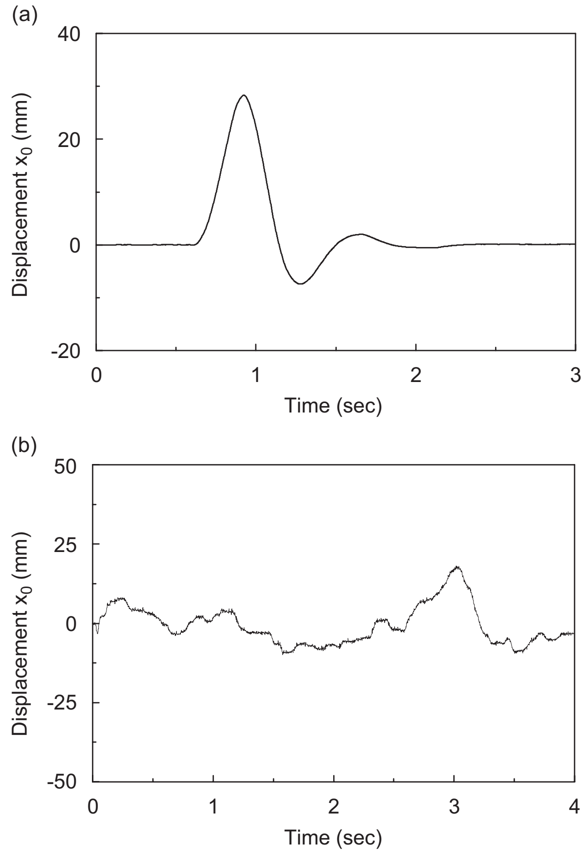

Fig. 8. Bump (a) and random (b) road excitation profiles.

second type of road profile, normally used to evaluate the frequency response, is a stationary random process with zero mean described by

$$\dot{z} + \rho_r V_s z = V_s W_n, \tag{22}$$

where W_n is the white noise with intensity calculated by $2\sigma^2\rho_r V_s$, ρ_r is the road roughness parameter, and σ^2 is the covariance of road irregularity. In the random excitation, the vehicle travels on the paved road with constant velocity of 72 km/h (= 20 m/s).

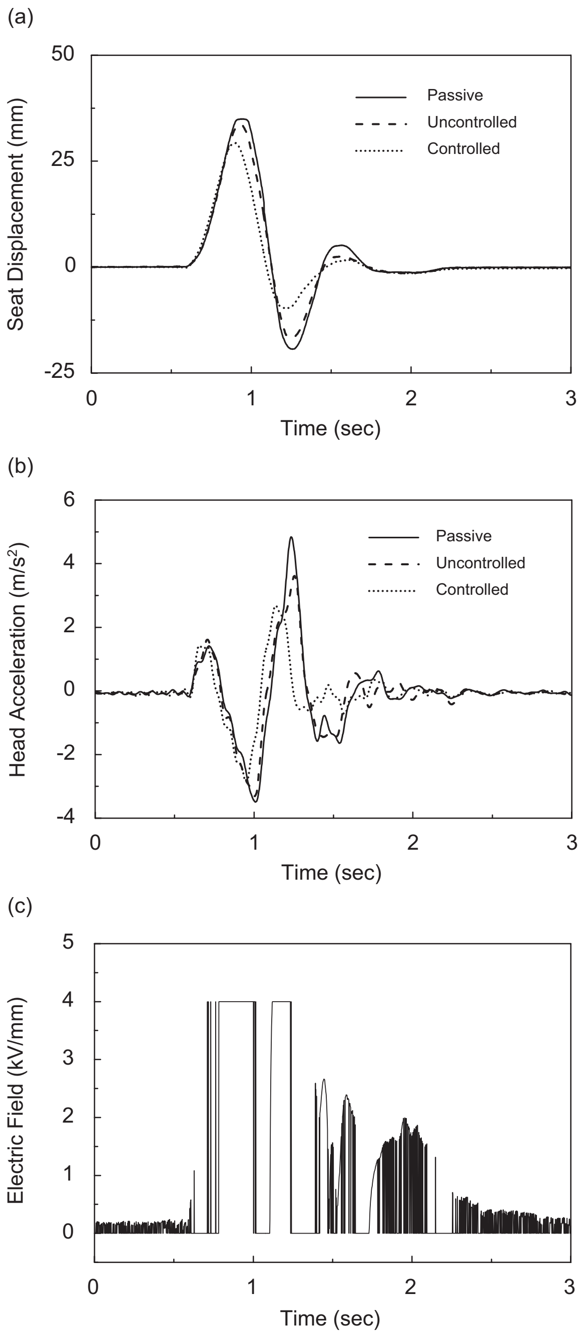

Fig. 9. Control responses of the ER seat suspension system under bump excitation.

Fig. 7 presents the measured results of the displacement transmissibility and acceleration transmissibility of the proposed ER seat suspension system at seat frame and driver's head. The measured natural frequency of the ER seat suspension system without electric field is 2.1 Hz, which is the same as for the passive seat suspension system for commercial truck. It is clearly observed that the peak of each transmissibility is substantially reduced by employing the control electric field determined from the SMC, and its frequency is slightly increased. It is calculated that the transmissibilities at seat and driver's head are reduced by 32% and 23%, respectively. Fig. 8 shows road profiles such as bump and random excitation. Fig. 9 presents the measured bump responses of the seat suspension system in the time domain. As clearly observed from the results, the output responses at the seat and driver's head have been reduced up to 30% by activating the proposed ER seat damper. It is also seen from Fig. 9(c) that the electric field has been supplied to the ER seat damper according to the semi-active control conditions. It is remarked that the passive result was obtained by replacing the ER seat damper by a commercially available passive oil seat damper for large trucks. From the result, it can be assured that the uncontrolled performances of the ER damper well agree with those of conventional passive damper on the concept of fail safe.

Fig. 10 shows the measured random responses of the seat suspension system. As shown in Fig. 10(a), the rms values of the frequency-weighted accelerations at head are evaluated and compared with the fatigue-decreased proficiency boundary in ISO 2631-1 as a function of frequency and time. As clearly observed from the results, the ride quality has been improved with respect to preservation of impaired working efficiency or fatigue. It is seen that the controlled rms value is distinctly reduced at around 2 and 6 Hz at which there exists resonance of human-body. The SEAT values and VDV ratio of the proposed ER seat suspension are also shown in Fig. 11. From the results, it is shown that the SEAT value and VDV ratio are reduced by 9% and 15%, respectively, which means that the proposed scheme is very effective for severe vibration. Fig. 12 shows

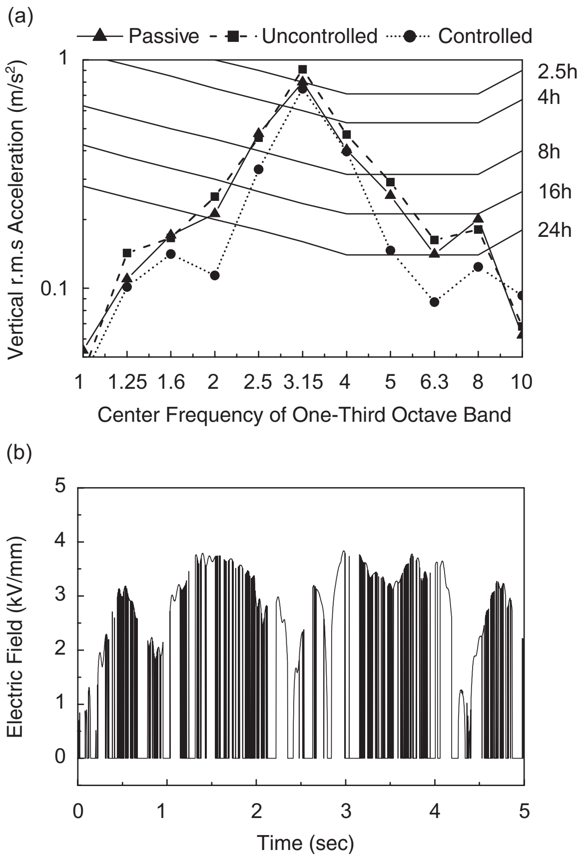

Fig. 10. Control responses of the ER seat suspension system under random excitation.

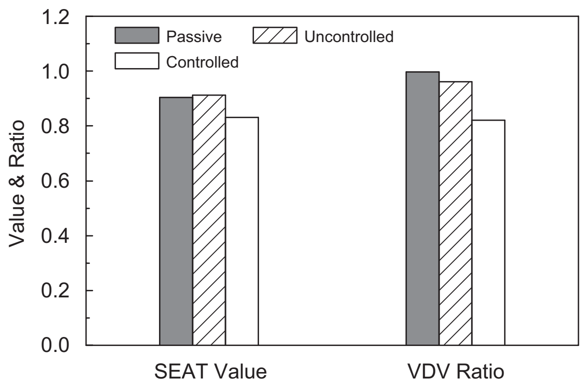

Fig. 11. SEAT value and VDV ratio of the ER seat suspension system.

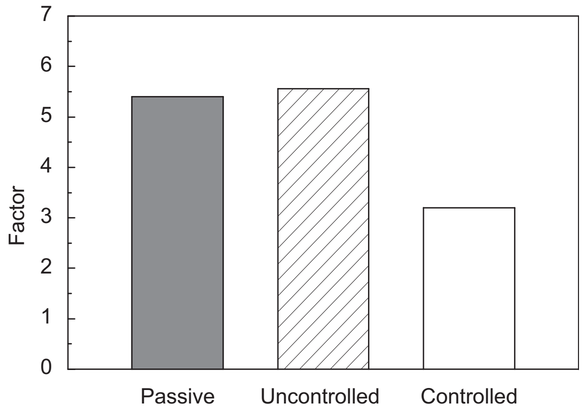

Fig. 12. Crest factor of the ER seat suspension system.

that the crest factor for random input is reduced up to 50% by activating controller. From these results for two typical road profiles, it can be assured that the proposed control scheme can improve ride quality under any other profile road.

6. Conclusion

A semi-active ER seat suspension system was proposed for commercial vehicles and its control performances were investigated by realizing the sliding mode controller (SMC). A 12-dof seat-driver model has been derived and reduced to 6-dof via the balanced model reduction. The SMC with state observer was then designed and experimentally realized to attenuate unwanted vertical vibration to a seated driver. In the experiment, a driver directly sat on the controlled seat, and vertical displacement and frequency-weighted rms accelerations were measured at the seat frame and driver's head, respectively. Then, it has been demonstrated that transmitted vibration levels are substantially reduced by adopting the semi-active ER seat damper. In addition, the ER seat suspension associated with the SMC was shown to reduce SEAT value by more than 10%, and to reduce VDV ratio by more than 15%. In the near future, a field test of the proposed ER seat suspension system will be undertaken on road environment as a second phase of this study.

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