Active suspension of truck seat

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The driver's seat of a heavy duty truck is usually mounted on a spring–damper assembly anchored to the cab floor. To improve riding comfort, this study investigated the effects of mounting a computer-controlled actuator in parallel with the traditional spring–damper assembly. A dynamic model of the seat is represented by a two degree-of-freedom system, including a cushion. In this paper, a control system is designed, using optimal control theory, which minimizes rms vertical acceleration at a point representing the driver's hip point. In this system, accelerations of the hip point, the seat frame and the cab floor are picked up and integrated to obtain the state variables to be fed back and fed forward to the actuator through a digital computer. The actuator is constructed with electric servo-motor and ball-screw mechanism. The experimental study was carried out on a shaker, which simulates the vibrations of the cab floor in actual service. Results were obtained for both a dummy and a real human body. The vibration test produced rms accelerations of the seat and the hip point of about 1.0 m/s^2 without the actuator, while the rms accelerations were suppressed to about 0.5 m/s^2 at a rms input voltage to the servo-motor of 1.0 V.

1. Introduction

Freight shipment by heavy duty truck is becoming more common recently, with longer distances and long transit times contributing to corresponding increases in driver's fatigue and driver's fatigue related accidents. As a result, improvement of the riding comfort of the driver's seat is seen as an important step in improving safety, and driver performance. Many studies on car seat vibration have been published to date; these studies are classified into 2 groups.

The first group is concerned with the measurement and/or analysis of seat vibration. For example, Pope et al. [11] studied the effects of various cushions on the dynamic response. Shinjo [20] examined transient riding comfort for heavy duty trucks, and Volfson [23] carried out simulations of the seat dynamics of off-road vehicles. Huston et al. [5] performed field measurements of seat vibrations, and Gu [4] measured vibration transmissibility using a mass dummy, Lewis and Griffin [9] also studied vibration transmissibility of a car seat with a suspended back-rest.

The second group is concerned with the isolation of seat vibration, and this group is further divided into 3 groups; the passive suspension, the semi-active suspension, and the full-active suspension. The passive suspension, composed of spring–damper assemblies, has been analyzed by Gouw et al. [3] in their study of driver comfort and safety in off-highway vehicles using optimal seat suspensions. Sankar and Afonso [14] promoted design and testing for lateral seat suspension of off-road vehicles.

The semi-active suspension, in which suspension parameters are adaptively shifted, Ranganathan and Sriram [12] designed a PC-based software for off-road vehicle seat suspensions, and Amirouche et al. [1] worked out an optimal driver seat suspension design for heavy trucks and heavy vehicle systems.

As for the full-active suspension, McCormac et al. [10] developed a dual-axis active seat suspension system using an electro-hydraulic actuator, and Stein and Ballo [21] developed a driver's active seat suspension for off-road vehicles using an electro-hydraulic actuator. Johnson [7] developed an active seat suspension to control low back injuries using an electro-hydraulic actuator. Stein [22] investigated an active vibration control system for a driver's seat using an electro-pneumatic actuator, and Ballo [2] examined the power requirements of an active vibration control system using an electro-pneumatic actuator. Ballo's results were discussed by Ryba [13]. Shimogo and his team [15–19] developed active suspension systems for heavy duty truck seats using an electric servo-motor.

In the case of the heavy duty truck, the driver's seat is usually supported by a spring–damper assembly connected to the cab floor. The cab-floor vibration has a relatively wide band width with a dominant frequency of 2.2 Hz. Although the general floor vi-

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bration is effectively isolated by this suspension, the frequency component centered around 2 to 3 Hz is not so well damped. In the typical design procedure of a vibration isolator, the natural frequency of the suspension should be in a lower frequency region than the dominant frequency of the disturbance. However, it is difficult to attain a very low disturbance frequency, say 2.2 Hz, due to the stroke limitation of the suspension. In the seat suspension discussed in this study, the natural frequency of the seat, including driver's mass, is 2.6 Hz, which was selected to be lower than the natural frequency of 5.4 Hz of the driver's mass on the seat cushion. Although the 5.4 Hz component is suppressed by this spring–damper suspension, it is difficult to suppress the 2–3 Hz components by the spring–damper suspension alone. According to the ISO2631-1:85, the 4–8 Hz component is most important to improve the ride comfort of the seat. However, the 2–3 Hz components also play a negative role and effect riding comfort. To suppress the 2–3 Hz components, an active suspension, in which an actuator is installed in parallel to the spring–damper assembly, is necessary. In the full-active suspensions proposed thus far, electro-hydraulic or electro-pneumatic actuators have been most commonly studied (McCormac et al. [10]; Stein and Ballo [21]; Johnson [7]; Stein [22]; and Ballo [2]).

In this study, an electric servomotor and ballscrew mechanism, installed in parallel to the spring–damper assembly, are applied to improve the riding comfort by damping the low frequency component of seat vibration. The electric servomotor system is simpler than the hydraulic or pneumatic actuators, and its loading capacity and frequency range are appropriate to seat vibration control. The control law is introduced using the optimal regulator theory on the basis of a simplified seat structure model of the truck. In particular, the feed-forward link of the cab floor vibration is provided along with the feed-back link of the seat vibration. The control performance was evaluated by computer simulation and experimental study on the shaker table realizing actual vibrations of the cab floor for both a dummy and a real human body.

2. Analytical model

The cab of a heavy truck is usually mounted on the chassis frame, which, in turn, is mounted on the axes, so the coupled vibration of the whole structure should be considered in an analysis of the vertical vibration

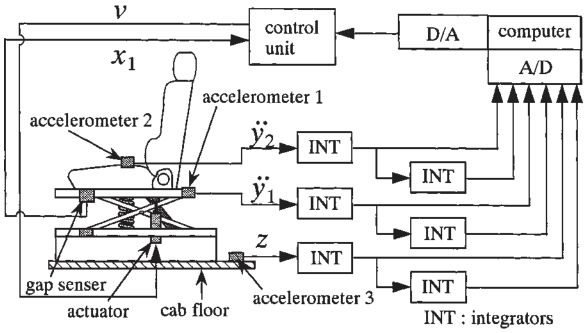

Fig. 1. Control system.

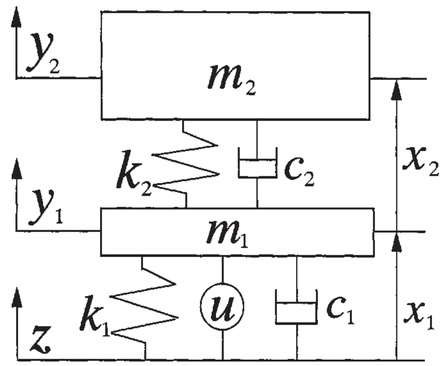

Fig. 2. Analysis model.

of the driver's seat. However, since the seat mass (including the driver's mass) is relatively small compared to that of the cab (the mass ratio is about $75/800$), the cab floor vibration is assumed to be transferred to the seat in the way of a cascade connection or in the one-way manner, that is, the seat vibration does not affect the cab vibration. The schematic illustration of the seat and the vibration control system are shown in Fig. 1. If the flexibility of the cushion on the seat frame is taken into account, the analytical model of the seat can be represented as a two degree-of-freedom system (Fig. 2), in which m_1 and m_2 denote the seat frame mass and the driver's mass, respectively. The friction between the driver's body and the seat back and the reaction between the driver's feet and the cab floor or between the driver's hands and the steering wheel are neglected in this study. The driver's mass is corrected by separating the leg mass from the body mass. The reduced mass of the driver on the seat set is at 80% to 90% of the whole body mass. In this study, the typical mass of the driver is assumed to be 70 kg, so the reduced mass is 60 kg, which is about 85.7% of the whole mass.

3. State equations

Motion equations of the system shown in Fig. 2 are:

$$
\begin{aligned}
m_1\ddot{y}_1 &= -c_1(\dot{y}_1 - \dot{z}) - k_1(y_1 - z) \\
&\quad - c_2(\dot{y}_1 - \dot{y}_2) - k_2(y_1 - y_2) + u, \\
m_2\ddot{y}_2 &= -c_2(\dot{y}_2 - \dot{y}_1) - k_2(y_2 - y_1).
\end{aligned} \tag{1}
$$

The control system shown in Fig. 1 is realized by the feedback of the vertical displacements and velocities of the cushion surface $y_2, \dot{y}_2$, and the seat frame $y_1, \dot{y}_1$, and by the feedforward of the floor displacement and the velocity $z_1, \dot{z}_1$. These variables are obtained by integrating the detected accelerations $\ddot{y}_2, \ddot{y}_1, \ddot{z}$. The actuator consists of a DC servomotor and a ballscrew mechanism, which are installed vertically under the seat. The relationship between the relative velocity of the suspension and the rotation speed of the servomotor is:

$$\dot{y}_1 - \dot{z}_1 = \alpha\dot{\theta}, \tag{2}$$

where α denotes the lead of the ballscrew. The rotation speed of the servomotor is expressed as a linear function of the input voltage v and the torque T as follows:

$$\dot{\theta} = a'v - b'T \tag{3}$$

(a', b' are constants of the servomotor). The relationship between the torque T and the vertical force u of the actuator is:

$$T = \alpha u. \tag{4}$$

Substituting Eqs (3) and (4) into Eq. (2), we obtain

$$
\begin{aligned}
&\dot{y}_1 - \dot{z}_1 = av - bu, \\
&u = -(\dot{y}_1 - \dot{z})/b + (a/b)v,
\end{aligned} \tag{5}
$$

where $a = a'\alpha$, $b = b'\alpha^2$. Substituting Eq. (5) into Eq. (1), the vertical force u is replaced by the input voltage v. The state equation of the seat suspension system is introduced from the motion equation, that is:

$$\dot{y}_\mathrm{s} = A_\mathrm{s} y_s + b_\mathrm{s} v + d_\mathrm{s} z_\mathrm{f}, \tag{6}$$

where

$$y = \begin{bmatrix} y_1\ y_2\ \dot{y}_1\ \dot{y}_2 \end{bmatrix}^\mathrm{T}, \qquad z_\mathrm{f} = \begin{bmatrix} z\ \dot{z} \end{bmatrix}^\mathrm{T}, \tag{7}$$

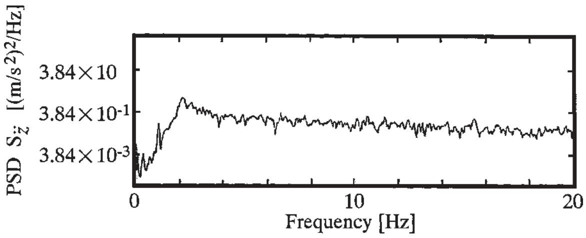

Fig. 3. Power spectral density of floor acceleration.

$$A_\mathrm{s} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\dfrac{k_1+k_2}{m_1} & \dfrac{k_2}{m_1} & -\dfrac{1+b(c_1+c_2)}{bm_1} & \dfrac{c_2}{m_1} \\ \dfrac{k_2}{m_2} & -\dfrac{k_2}{m_2} & \dfrac{c_2}{m_2} & -\dfrac{c_2}{m_2} \end{bmatrix},$$

$$b_\mathrm{s} = \begin{bmatrix} 0 & 0 & \dfrac{a}{bm_1} & 0 \end{bmatrix}^\mathrm{T},$$

$$d_\mathrm{s} = \begin{bmatrix} 0 & 0 & \dfrac{k_1}{m_1} & 0 \\ 0 & 0 & \dfrac{1+bc_1}{bm_1} & 0 \end{bmatrix}^\mathrm{T}.$$

The power spectral density of the cab floor vibration, shown in Fig. 3, was obtained from actual data taken from a heavy duty truck running on Tokyo metropolitan high way. As a single peak appears in the spectrum, the transfer function of the shaping filter to produce the floor vibration from the white noise is approximated by the expression of the second order system:

$$G_\mathrm{f}(s) = \frac{s^2}{s^2 + 2\zeta_\mathrm{f}\omega_\mathrm{f} s + \omega_\mathrm{f}^2}. \tag{8}$$

The parameters ω_f and ζ_f are determined by curve fitting to the actual data in the frequency domain of Fig. 3. The state equation of the shaping filter is derived from the transfer function as follows:

$$\dot{z}_\mathrm{f} = A_\mathrm{f} z_\mathrm{f} + d_\mathrm{f} w, \tag{9}$$

where

$$z_\mathrm{f} = \begin{bmatrix} z\ \dot{z} \end{bmatrix}^\mathrm{T},$$

$$A_\mathrm{f} = \begin{bmatrix} 0 & 1 \\ -\omega_\mathrm{f}^2 & -2\zeta_\mathrm{f}\omega_\mathrm{f} \end{bmatrix}, \qquad d_\mathrm{f} = \begin{bmatrix} 0 \\ 1 \end{bmatrix},$$

$w(t)$ is Gaussian white noise with mean zero.

So the state equation of the augmented system, including the shaping filter, is obtained:

$$\dot{y} = Ay + bv + dw, \tag{10}$$

where

$$y = \lfloor y_\mathrm{s}\ z_\mathrm{f} \rfloor^\mathrm{T} = \left[y_1\ y_2\ \dot{y}_1\ \dot{y}_2\ z\ \dot{z} \right]^\mathrm{T},$$

$$A = \begin{bmatrix} A_\mathrm{s} & d_\mathrm{s} \\ 0 & A_\mathrm{f} \end{bmatrix}, \quad b = \begin{bmatrix} b_\mathrm{s} \\ 0 \end{bmatrix}, \quad d = \begin{bmatrix} 0 \\ d_\mathrm{f} \end{bmatrix}. \quad (11)$$

4. Optimal control law

As the control purpose is to minimize the vertical acceleration of the hip point on the seat under the constraints of the relative displacement of the suspension and the input voltage to the servomotor, the criterion function is given as follows:

$$\begin{aligned} J &= E\lfloor q_1(y_1 - z)^2 + q_2(y_2 - y_1)^2 + q_3\dot{y}_1^2 \\ &\quad + q_4\dot{y}_2^2 + q_5 y_1^2 + q_6 y_2^2 + rv^2 \rfloor \\ &= E[y^\mathrm{T} Q y + rv^2], \end{aligned} \quad (12)$$

where

$$\begin{aligned} Q &= q_1 c_1^\mathrm{T} c_1 + q_2 c_2^\mathrm{T} c_2 + q_3 c_3^\mathrm{T} c_3 \\ &\quad + q_4 c_4^\mathrm{T} c_4 + q_5 c_5^\mathrm{T} c_5 + q_6 c_6^\mathrm{T} c_6. \end{aligned} \quad (13)$$

$$c_1 = \lfloor 1\ 0\ 0\ 0\ {-1}\ 0 \rfloor, \quad c_2 = \lfloor -1\ 1\ 0\ 0\ 0\ 0 \rfloor,$$
$$c_3 = \left[0\ 0\ 1\ 0\ 0\ 0 \right], \quad c_4 = \left[0\ 0\ 0\ 1\ 0\ 0 \right],$$
$$c_5 = \left[1\ 0\ 0\ 0\ 0\ 0 \right], \quad c_6 = \left[0\ 1\ 0\ 0\ 0\ 0 \right],$$

$q_1, q_2, q_3, q_4, q_5, q_6$ and r are the weighting parameters.

According to the optimal regulator theory, the optimal control law to minimize the criterion is :

$$\begin{aligned} &v = -Fy, \\ &F = r^{-1} b^\mathrm{T} P = \left[f_1\ f_2\ f_3\ f_4\ f_5\ f_6 \right], \end{aligned} \quad (14)$$

where P is obtained by solving the Riccati equation

$$PA + A^\mathrm{T} P - r^{-1} P b b^\mathrm{T} P + Q = 0, \quad (15)$$

as indicated in the literature, for example, by Kwakernaak and Sivan [8]. To obtain the optimal feedback gain F of the digital control system in the discrete time domain, the MATLAB CONTROL SYSTEM TOOLBOX lqrd of MATHWORKS Inc. was used.

Table 1
Specification of experimental model

Floor vibration model	$\omega_\mathrm{f} = 13.82$ rad/s, $\zeta_\mathrm{f} = 0.150$ $W = 4.4 \times 10^{-4}$ (m/s^2)^2/Hz rms $\ddot{z} = 1.158$ m/s^2
Seat suspension	$k_1 = 1.960 \times 10^4$ N/m $c_1 = 2.156 \times 10^3$ N s/m
Cushion	$k_2 = 6.907 \times 10^4$ N/m $c_2 = 7.247 \times 10^2$ N s/m
Seat mass	$m_1 = 15$ kg
Body mass	$m_2 = 60$ kg
Actuator	$\dot{x}_1 = av - bu$ $a = -4.640 \times 10^{-2}$ (m/s)V $b = -2.026 \times 10^{-6}$ (m/s)N

5. Experimental method

The experiment of the active seat suspension was performed by mounting a real truck seat on an electro-hydraulic shaker table (max. load: 10 t, frequency range: 0–50 Hz, max. stroke: ± 150 mm). Experiments were conducted by placing a dummy, as well as a real human body, on the seat. The specifications of the experimental system is shown in Table 1 (actuator specification: stroke $= \pm 37.5$ mm, frequency $= 0 \sim 10.8$ Hz, control force $= 200$ N). Although a hysteresis due to the friction produced by the sliding linkage is indicated along with a nonlinearity in the shock absorber, these characteristics are linearized with the assumptions that the relative displacement amplitude is 5 mm and the velocity amplitude is 0.3 m/s. The characteristics of the cushion are separately identified in the experiment.

6. Results of the experiment

If the actuator is not installed, the acceleration spectrum of the seat frame and the dummy hip point have a peak value at 2 to 3 Hz as shown in Fig. 4. This peak disappears in the controlled case as shown in Fig. 5 (weighting parameters: $q_1 = 10^5$, $q_2 = 0$, $q_3 = 10^6$, $q_4 = 10^3$, $q_5 = 10^2$, $q_6 = 10^2$, $r = 10^{1.5}$, and control gains: $f_1 = 16.8429 \times 10^3$, $f_2 = -18.2267 \times 10^3$, $f_3 = -2.7056 \times 10^3$, $f_4 = -0.2216 \times 10^3$, $f_5 = -3.7431 \times 10^3$, $f_6 = -18.7855 \times 10^3$).

Figure 6 indicates a corresponding results for real human body of 70 kg, similar to that of Fig. 5, because the effective mass of the human body discounting legs is nearly equal to the mass of the dummy.

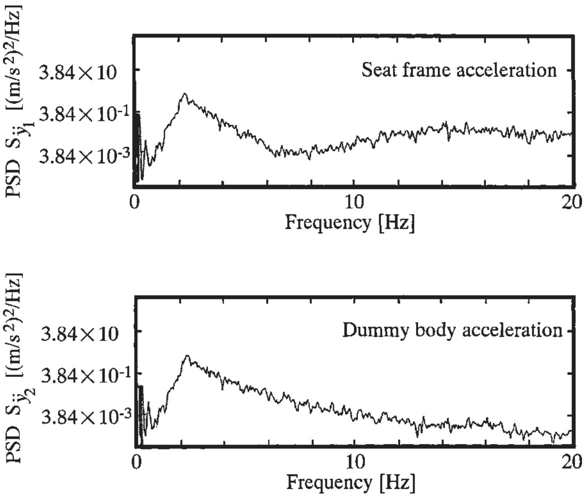

Fig. 4. Power spectral densities without actuator.

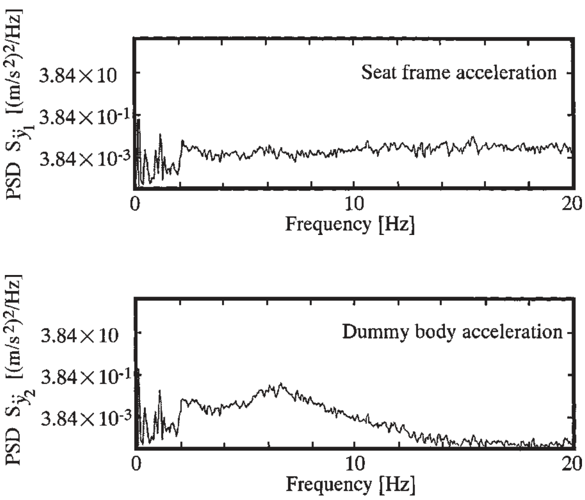

Fig. 5. Power spectral densities with actuator.

The difference over the range of 5 to 8 Hz between the human body PSD and the dummy body PSD may be due to differences in seating conditions. The dummy body rolled and pitched on the seat cushion during the vibration test in spite of attempts to constrain the dummy body neck to the seat back by a rope. In contrast, the human body can be seated in a stable manner on the seat cushion. The human body itself can easily suppress rolling and pitching, and the contact surface between the hip and the seat cushion is considerably softer when compared with that of the dummy body.

The trade-off plots of the acceleration of the seat frame (a), the acceleration of the hip point (b), and the relative displacement of the suspension (c), against the input voltage of the actuator are shown in Fig. 7.

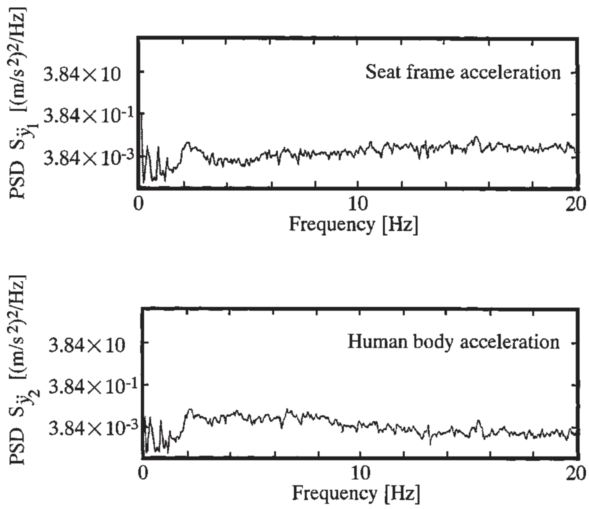

Fig. 6. Power spectral densities with actuator.

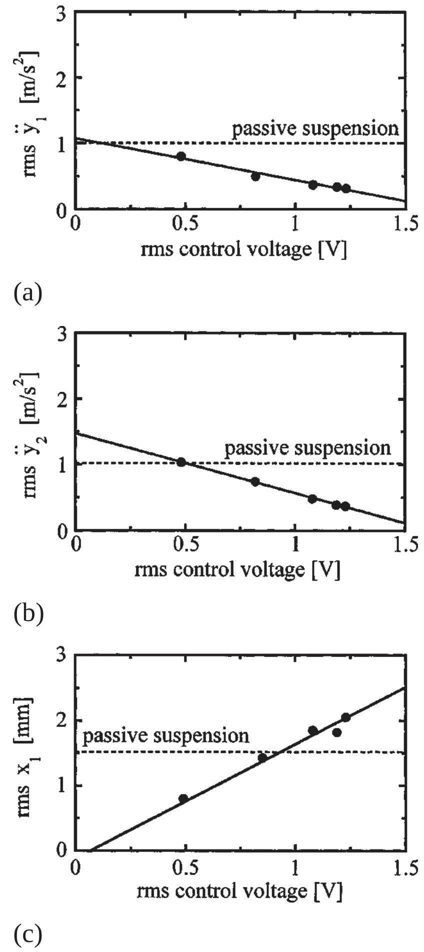

Fig. 7. Trade off curve. (a) Absolute acceleration of seat; (b) Absolute acceleration of hip point; (c) Relative displacement of suspension.

These results were obtained by varying the weighting parameter of the input voltage in the criterion function ($r = 10^{1.5}$ to $10^{3.5}$). If the rms input voltage is 1 V, for example, the rms acceleration of the hip point is about 1 $\mathrm{m/s^2}$ without the actuator, while it is reduced to about 0.5 $\mathrm{m/s^2}$ with the actuator.

7. Discussion

Then experimental data indicate that the frequency response of the hip point around 2 to 3 Hz is suppressed, and thus, the riding comfort is improved, by the active seat suspension proposed in this study.

Prediction of the control performance was made by numerical simulation using a digital computer to determine the values of the weighting parameters in the criterion function. The experimental results of control performance were usually lower than the simulation results, because the nonlinear characteristics of the real system was linearized to determine the optimal control law.

The feedforward link from the cab floor vibration to the controller is essential to suppress the seat vibration, because the cab floor vibration has a dominant frequency component. The control law was determined for the augmented system by including the shaping filter which generates the cab floor vibration, and state variables of the shaping filter are then fed forward to the controller.

Therefore, if the real cab floor vibration spectrum is changed from the specified spectrum used to design the control system, the control performance may degrade. To reduce the effects of such abnormal conditions, the robustness of the control system should be specified. However, in actuality, the cab floor vibration spectrum was hardly influenced by changes in road conditions in the running test. For very rough road surface conditions, the actuator should be deactivated. With the active control off, only the spring–damper assembly is effective because the actuator is installed in parallel with the spring–damper assembly. Some resistance will remain as a result of the servomotor and the ballscrew mechanism, even if the control is off.

In practice, if the absolute displacement of the hip point decreased, then the relative displacement between the cab floor and the hip point increases, and it has an effect on the operations of the pedals and the steering wheel.

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