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Abstract

In this paper, the control of roll vibration and vertical vibration of a seat suspension caused by uneven road under both sides of tyres is studied. The heavy duty vehicles are generally working under severe conditions, where the uneven road can cause roll vibration and vertical vibration of the vehicle body and can have an effect on seat suspension. The conventional single-degree of freedom (single-DOF) seat suspension can only isolate the vertical vibration while the roll vibration will be totally transferred to the driver's body. The high magnitude of driver body's lateral acceleration caused by roll vibration will influence drivers' health and have a negative effect on ride comfort. A two-layer multiple-DOF active seat suspension, which has a z-axis DOF in the bottom layer and a roll DOF in the top layer, is proposed in this paper. A non-singular terminal sliding controller is designed to control the top-layer to reduce the lateral acceleration and roll acceleration by tracking a desired roll angle. An H_∞ controller with disturbance compensation is applied to control the bottom-layer for vertical vibration isolation. Both controllers use the variables which can be measured or estimated in the practical application as feedback signals. Two inertial measurement units (IMUs) MPU9250 are used in order to estimate the rotary angles of the top and base platforms of the two-layer multiple-DOF active seat suspension. The effectiveness of the seat suspension and control method is validated by both simulations and experiments. A single-DOF active seat suspension and a well-tuned conventional passive one are applied for comparison. Based on ISO 2631, the vibration total value of frequency weighted root mean square (FW-RMS) acceleration of the multiple-DOF active seat surface has 29.8% and 23.6% reductions for evaluating its influence on health and ride comfort, respectively, when compared with the single-DOF active one. The proposed vibration isolation method can effectively reduce the whole body vibration (WBV) of heavy duty vehicle drivers, and it shows high potential for practical application.

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Control of a multiple-DOF vehicle seat suspension with roll and vertical vibration

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Abstract

In this paper, the control of roll vibration and vertical vibration of a seat suspension caused by uneven road under both sides of tyres is studied. The heavy duty vehicles are generally working under severe conditions, where the uneven road can cause roll vibration and vertical vibration of the vehicle body and can have an effect on seat suspension. The conventional single-degree of freedom (single-DOF) seat suspension can only isolate the vertical vibration while the roll vibration will be totally transferred to the driver's body. The high magnitude of driver body's lateral acceleration caused by roll vibration will influence drivers' health and have a negative effect on ride comfort. A two-layer multiple-DOF active seat suspension, which has a z-axis DOF in the bottom layer and a roll DOF in the top layer, is proposed in this paper. A non-singular terminal sliding controller is designed to control the top-layer to reduce the lateral acceleration and roll acceleration by tracking a desired roll angle. An H_∞ controller with disturbance compensation is applied to control the bottom-layer for vertical vibration isolation. Both controllers use the variables which can be measured or estimated in the practical application as feedback signals. Two inertial measurement units (IMUs) MPU9250 are used in order to estimate the rotary angles of the top and base platforms of the two-layer multiple-DOF active seat suspension. The effectiveness of the seat suspension and control method is validated by both simulations and experiments. A single-DOF active seat suspension and a well-tuned conventional passive one are applied for comparison. Based on ISO 2631, the vibration total value of frequency weighted root mean square (FW-RMS) acceleration of the multiple-DOF active seat surface has 29.8% and 23.6% reductions for evaluating its influence on health and ride comfort, respectively, when compared with the single-DOF active one. The proposed vibration isolation method can effectively reduce the whole body vibration (WBV) of heavy duty vehicle drivers, and it shows high potential for practical application.

Keywords: Vibration control, Multiple-DOF, heavy duty vehicle, whole body vibration, ride comfort.

1. Introduction

Whole body vibration (WBV) including vertical, lateral and longitudinal vibration has been studied in [1-4], which have confirmed that the long-term exposure to WBV will cause health problems, such as lower back pain [5]. Drivers of heavy duty vehicles, such as the agricultural vehicle, construction vehicle and mining vehicle, need to work in severe WBV conditions for a long time. In order to protect their health and improve their ride comfort, seat suspension has been widely applied to isolate vibration [1, 6]. The vibration control of a multiple-degree of freedom (multiple-DOF) seat suspension is studied in this paper.

Generally, three kinds of single-DOF seat suspensions for vertical vibration control have been extensively studied, namely, passive, semi-active and active seat suspensions. A negative stiffness structure for passive suspension is proposed for overcoming the vibration amplification problem in low frequency [7]. The semi-active seat suspension is designed with semi-active devices. Semi-active seat suspensions using an electrorheological (ER) fluid damper and a magnetorheological (MR) fluid damper are proposed for vertical vibration control [8, 9]. The active seat suspension is the most effective way to improve ride comfort; it has been designed with different actuators. An active seat suspension with two electromagnetic linear actuators has been presented in [10]. An active seat suspension system comprised of a hydraulic absorber and a controlled air-spring has been proposed in [11].

Some researchers have considered other DOFs of vibration besides the vertical one. An active seat suspension with two-DOFs for military vehicles is designed for controlling vertical and lateral vibration [12]. A lateral seat suspension for off-road vehicles is designed and tested in [13]. An MR elastomer-based isolator is designed for the horizontal vibration reduction of a driver seat [14]. A six-DOF active seat suspension with six actuators designed as Stewart Platform has been applied for a patent [15]. An integrated semi-active seat suspension for both longitudinal and vertical vibration isolation is proposed in [16]. By compensating longitudinal and lateral acceleration, a four-DOF active seat suspension is designed and tested in [17].

In the past decades, many reputable vibration control algorithms have been proposed, e.g. H_∞ control [18, 19], adaptive control [20], positive position feedback control [21], passivity-based control [22], and intelligent control [23]. The attitude motion tracking controller is proposed for active suspension systems in [24]. Sliding mode controllers have been proven to be effective for tracking control. The non-singular terminal sliding model control is proposed for rigid manipulators in [25].

On the whole, though it is widely accepted that WBV has great influence on drivers' health and ride comfort, especially on heavy duty vehicle drivers, most researchers proposed methods for overcoming

vertical vibration while the multiple-DOF vibration control has not been paid more attention yet. Thus, the seat suspension considering multiple-DOF vibration control should be designed and corresponding control algorithm should be developed.

This paper proposes a new two-layer multiple-DOF seat suspension prototype for both roll vibration and vertical vibration control. The roll angles of the top and base platforms of the seat prototype are estimated based on the measurement of two inertial measurement units (IMUs) MPU9250. A desired roll angle of the top platform is calculated according to the roll vibration of the base platform. Then, a non-singular terminal sliding mode controller is designed for the top platform to track the desired roll angle. The bottom layer of the seat suspension is controlled by an H_∞ controller with disturbance compensation, the effectiveness of which has been validated in our previous study [26]. By controlling the roll vibration, the lateral acceleration of the driver body can be effectively suppressed. With the proposed control method, the negative effects of WBV on driver health and ride comfort have been greatly reduced.

The remainder of the paper is organised as follows: Section 2 discusses the roll movement effect on ride comfort and health. Section 3 presents the multiple-DOF seat suspension prototype and its model. The controller design is presented in Section 4. Section 5 presents the simulation and experiment evaluation. Finally, Section 6 presents the conclusions of this research.

2. Roll movement effect on ride comfort and health

Generally, the vertical vibration isolation is considered when designing a seat suspension for conventional road vehicles but for heavy duty vehicles, such as agricultural vehicles and construction vehicles, the severe roll vibration caused by uneven roads under the left and right tyres of the vehicle will deteriorate the ride comfort greatly. Figure 1 shows a half-car model with chassis and seat suspension, where m_l , m_r , m_v and m_b are masses of the left tyre, right tyre, vehicle body and driver body, respectively; K and C are the corresponding stiffness and damping parameters. The uneven road under the left and right tyres will cause vehicle's vertical vibration, and at the same time, the roll vibration of vehicle chassis θ is generated. The vertical vibration and roll vibration will all be transferred to seat suspension. For normal road conditions, the magnitude of θ is small, thus, the roll vibration of driver body α is also small. By only controlling the vertical vibration, the ride comfort will be improved greatly. When the roll vibration magnitude is high, the driver body can be regarded as a mass in the top of an inverted pendulum; thus, the high magnitude lateral acceleration is caused by the rotary vibration:

$$a_{y4} = \ddot{\alpha}r \quad (1)$$

where $\ddot{\alpha}$ is the roll acceleration of driver body; r is the distance from the centre of driver body mass to its unknown rotary centre; a_{y4} is the acceleration along y_4 axis of driver body caused by roll acceleration.

Figure 1. Half car model with chassis and seat suspension

Generally, ISO 2631 is applied to evaluate human exposure to WBV [27]. The effect of vibration on driver health and ride comfort is dependent on the vibration frequency content. The frequency weighted root mean square (FW-RMS) acceleration a_w is determined by:

$$a_w = [\sum_i (W_i a_i)^2]^{\frac{1}{2}} \quad (2)$$

where a_i is the RMS acceleration for the i th one-third octave band; W_i is the weighting factor for the i th one-third octave band given in ISO 2631.

The fourth power vibration dose value (VDV) is more sensitive to peaks than the RMS method, so it is also applied to evaluate vibration magnitude, which is defined as:

$$VDV_{total} = \left\{ \int_0^T [a_w(t)]^4 dt \right\}^{\frac{1}{4}} \quad (3)$$

The vibration total value of FW-RMS acceleration is calculated as:

$$a_v = (k_x^2 a_{wx}^2 + k_y^2 a_{wy}^2 + k_z^2 a_{wz}^2)^{\frac{1}{2}} \quad (4)$$

where a_{wx} , a_{wy} and a_{wz} are FW-RMS accelerations with respect to the three orthogonal axes; k_x , k_y and k_z are multiplying factors which are defined in Table 1 with respect to health and ride comfort. The value of multiplying factors indicates that, with respect to the effect on ride comfort, the

vibrations along three axes have same weightings; but the longitudinal vibration (a_{wx}) and lateral vibration (a_{wy}) have more influence on health than vertical vibration (a_{wz}).

Table 1. Multiplying factors for WBV [27]

	Health	Comfort
k_x	1.4	1
k_y	1.4	1
k_z	1	1

In this paper, only the control of vibrations along z_4 and y_4 axes of the driver body is studied. The vibration along the x_4 axis of the driver's body is much more complicated considering the existing effects of road slope, seat backrest, and feet on the floor, thus, its control problem will not be discussed in the current work.

3. Seat suspension model

3.1 Prototype

A multiple-DOF seat suspension prototype is designed and fabricated as shown in Figure 2, which consists of two layers. The bottom layer of the multiple-DOF seat suspension, which is proposed in [26, 28], is applied for controlling vertical vibration. In the top layer, there is a universal joint as shown in Figure 3; two actuators are connected with the shafts of the universal joint, thus the roll and pitch movement of the top-layer suspension can be controlled; there are four springs in the four corners to provide rotational stiffness, which can be equivalent as two torsional springs in the universal joint. Four 400 W Panasonic servo motors (MSMJ042G1U) with gear boxes to amplify the torque output are applied in the seat suspension as actuators; two motors are assembled in the top-layer suspension for the roll and pitch movement and another two are used in the bottom-layer for the vertical vibration control. The maximum vertical force of the bottom layer generated can reach 350 N, and the maximum torque output of the roll and pitch actuators is 52 Nm. **In this work, it is assumed that the response of servo motor is sufficiently fast and the torque outputs of servo motors are accurately controlled by their own drives; thus, the actuator dynamics are not considered.** With the two-layer structure, the coupling between the vertical and roll vibrations is small which has been verified in the experiment.

Figure 2. Multiple-DOF seat suspension prototype

Figure 3. Universal joint.

3.2 System modelling

The roll acceleration caused lateral vibration and vertical vibration will be studied, thus, a Y-Z plane model of the multiple-DOF seat suspension is built in this paper. As shown in Figure 4, five coordinate frames are built: frame 0 is the fixed reference; frame 1 is in the base of the seat suspension; frame 2 and frame 3 are in the roll joint, and fixed at bottom and top-layer, respectively; frame 4 is in the mass centre of the driver's body and seat. Using right hand rule, the x axes of all the frames can be known. There are vibration sources to cause frame 1 to move along z_0 and rotate around x_0 , namely, the vertical vibration and roll vibration. m_1 is the total mass of driver and seat; m_2 is the mass between top and bottom-layer; h_b is the distance between frame 4 and frame 2; h_s is the distance between frame 2 and the mass centre of m_2 ; h_0 is the initial height of frame 2 to frame 1; $h(t)$ is the dynamic displacement of frame 2 referred to frame 1. The bottom layer has stiffness k_s , friction force f_{rt} and active force u_t . In the top-layer, there are rotational stiffness k_r , friction torque f_{rr} and active torque u_r .

Figure 4. Y-Z plane model of Multiple-DOF seat suspension.

The model is further decomposed into two parts as shown in Figure 5 where P and N are the interaction forces aligned with the axes of frame 2.

Figure 5. Model decomposition. (a) top-layer. (b) bottom-layer.

For deriving the seat suspension model, the 4×4 homogeneous transformation matrix $\mathbf{T}$ and 4×1 homogeneous coordinate are applied [29]:

$$\mathbf{T} \triangleq \begin{bmatrix} \mathbf{R} & \mathbf{p} \\ \mathbf{0} & 1 \end{bmatrix} \quad (5)$$

$$\mathbf{q} = \begin{bmatrix} \mathbf{p} \\ 1 \end{bmatrix} \quad (6)$$

where $\mathbf{R}$ is the 3×3 rotation matrix; $\mathbf{p}$ is a 3×1 translation vector. In this paper, the last term in the derivative and the second derivate of $\mathbf{q}$ is always 1, because it does not have practical physical meaning.

Thus, the transformation matrices of adjacent frames are defined as:

$$\mathbf{T}_0^1 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & c\theta & -s\theta & 0 \\ 0 & s\theta & c\theta & z_s \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (7)$$

$$\mathbf{T}_1^2 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & h + h_0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (8)$$

$$\mathbf{T}_2^3 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & c\beta & -s\beta & 0 \\ 0 & s\beta & c\beta & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (9)$$

$$\mathbf{T}_3^4 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & h_b \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (10)$$

where θ is the roll vibration from the cab floor; z_s is the vertical vibration displacement; β is the relative rotary angle of frames 2 and 3; s and c represent the trigonometric functions $\sin$ and $\cos$, respectively. Thus, the roll vibration of the driver body is $\alpha = \theta + \beta$.

Then, the transformation matrices of frames 2, 3 and 4 referred to frame 0 can be derived:

$$\mathbf{T}_0^2 = \mathbf{T}_0^1 \mathbf{T}_1^2 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & c\theta & -s\theta & -s\theta(h + h_0) \\ 0 & s\theta & c\theta & c\theta(h + h_0) + z_s \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (11)$$

$$\mathbf{T}_0^3 = \mathbf{T}_0^2 \mathbf{T}_2^3 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & c(\theta + \beta) & -s(\theta + \beta) & -s\theta(h + h_0) \\ 0 & s(\theta + \beta) & c(\theta + \beta) & c\theta(h + h_0) + z_s \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (12)$$

$$\mathbf{T}_0^4 = \mathbf{T}_0^3 \mathbf{T}_3^4 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & c(\theta + \beta) & -s(\theta + \beta) & -s\theta(h + h_0) - s(\theta + \beta)h_b \\ 0 & s(\theta + \beta) & c(\theta + \beta) & c\theta(h + h_0) + z_s + c(\theta + \beta)h_b \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (13)$$

The interaction force from upper joint on lower joint in frame 2 is defined as:

$$\mathbf{F}_2^a = [0 \quad -P \quad -N]^T \quad (14)$$

Thus, the reaction force on upper joint in frame 3 and frame 4 can be defined as:

$$\mathbf{F}_4^{\text{ra}} = \mathbf{F}_3^{\text{ra}} = -\mathbf{R}_2^{3\text{T}} \mathbf{F}_2^{\text{a}} = -\mathbf{R}_3^2 \mathbf{F}_2^{\text{a}} = \begin{bmatrix} 0 \\ c\beta P + s\beta N \\ -s\beta P + c\beta N \end{bmatrix} \quad (15)$$

The gravity of m_1 in frame 0 is:

$$\mathbf{G}_0^{\text{m1}} = [0 \quad 0 \quad -m_1 g]^{\text{T}} \quad (16)$$

Thus the gravity of m_1 in frame 4 is:

$$\mathbf{G}_4^{\text{m1}} = \mathbf{R}_4^0 \mathbf{G}_0^{\text{m1}} = \begin{bmatrix} 0 \\ -s\alpha m_1 g \\ -c\alpha m_1 g \end{bmatrix} \quad (17)$$

It is easy to obtain the coordinate of m_1 in frame 0 from (13):

$$\mathbf{q}_0^{\text{m1}} = \begin{bmatrix} 0 \\ -s\theta(h + h_0) - s\alpha h_b \\ c\theta(h + h_0) + z_s + c\alpha h_b \\ 1 \end{bmatrix} \quad (18)$$

And, its velocity and acceleration can be derived:

$$\dot{\mathbf{q}}_0^{\text{m1}} = \begin{bmatrix} 0 \\ -\dot{\theta}c\theta(h + h_0) - s\theta\dot{h} - \dot{\alpha}c\alpha h_b \\ -\dot{\theta}s\theta(h + h_0) + c\theta\dot{h} + \dot{z}_s - \dot{\alpha}s\alpha h_b \\ 1 \end{bmatrix} \quad (19)$$

$$\begin{aligned} \ddot{\mathbf{q}}_0^{\text{m1}} &= \begin{bmatrix} 0 \\ -\ddot{\theta}c\theta(h + h_0) + \dot{\theta}^2 s\theta(h + h_0) - 2\dot{\theta}c\theta\dot{h} - s\theta\ddot{h} - \ddot{\alpha}c\alpha h_b + \dot{\alpha}^2 s\alpha h_b \\ -\ddot{\theta}s\theta(h + h_0) - \dot{\theta}^2 c\theta(h + h_0) - 2\dot{\theta}s\theta\dot{h} + c\theta\ddot{h} + \dot{z}_s - \ddot{\alpha}s\alpha h_b - \dot{\alpha}^2 c\alpha h_b \\ 1 \end{bmatrix} \end{aligned} \quad (20)$$

For formulating the dynamic equation of m_1 , its acceleration vector is transferred into frame 4:

$$\begin{aligned} \mathbf{p}_4^{\ddot{\text{m1}}} &= \mathbf{R}_4^0 \mathbf{p}_0^{\ddot{\text{m1}}} \\ &= \begin{bmatrix} 0 \\ -s\beta[\dot{\theta}^2(h + h_0) - \ddot{h}] - c\beta[\ddot{\theta}(h + h_0) + 2\dot{\theta}\dot{h}] - \ddot{\alpha}h_b + s\alpha\ddot{z}_s \\ -c\beta[\dot{\theta}^2(h + h_0) - \ddot{h}] + s\beta[\ddot{\theta}(h + h_0) + 2\dot{\theta}\dot{h}] - \dot{\alpha}^2 h_b + c\alpha\ddot{z}_s \end{bmatrix} \end{aligned} \quad (21)$$

Then, its dynamic equation is formulated:

$$m_1 \mathbf{p}_4^{\ddot{\text{m1}}} = \mathbf{F}_4^{\text{ra}} + \mathbf{G}_4^{\text{m1}} \quad (22)$$

We can get two equations:

$$\begin{aligned} m_1 \{ &-s\beta[\dot{\theta}^2(h + h_0) - \ddot{h}] - c\beta[\ddot{\theta}(h + h_0) + 2\dot{\theta}\dot{h}] - \ddot{\alpha}h_b + s\alpha\ddot{z}_s \} \\ &= c\beta P + s\beta N - s\alpha m_1 g \end{aligned} \quad (23)$$

$$\begin{aligned} m_1 \{ &-c\beta[\dot{\theta}^2(h + h_0) - \ddot{h}] + s\beta[\ddot{\theta}(h + h_0) + 2\dot{\theta}\dot{h}] - \dot{\alpha}^2 h_b + c\alpha\ddot{z}_s \} \\ &= -s\beta P + c\beta N - c\alpha m_1 g \end{aligned} \quad (24)$$

The interaction force N is calculated for following derivation:

$$N = m_1[-\dot{\theta}^2(h + h_0) + \ddot{h} - \dot{\alpha}^2 h_b c \beta - \ddot{\alpha} h_b s \beta + c \theta \ddot{z}_s + c \theta g] \quad (25)$$

The rotational inertial of m_1 around x_4 axis is I_{m1} . Summing the moments about the centroid of the driver body, a dynamic equation is obtained:

$$I_{m1} \ddot{\alpha} = -k_r \beta - f_{rr} + u_r + (c \beta P + s \beta N) h_b \quad (26)$$

where the simplified friction model is applied $f_{rr} = F_{rr} \text{sgn}(\dot{\beta})$, F_{rr} is the Coulomb friction coefficient.

Combing equations (23) and (26):

$$(I_{m1} + m_1 h_b^2) \ddot{\alpha} = -k_r \beta - F_{rr} \text{sgn}(\dot{\beta}) + u_r - m_1 \{s \beta [\dot{\theta}^2(h + h_0) - \ddot{h}] + c \beta [\ddot{\theta}(h + h_0) + 2 \dot{\theta} \dot{h}] - s \alpha \ddot{z}_s - s \alpha g\} h_b \quad (27)$$

Because the coordinate of m_2 in frame 2 is:

$$\mathbf{q}_2^{m2} = [0 \quad 0 \quad -h_s \quad 1]^T \quad (28)$$

its coordinate in frame 0 can be defined as:

$$\mathbf{q}_0^{m2} = \mathbf{T}_0^2 \mathbf{q}_2^{m2} = [0 \quad -s \theta(h + h_0 - h_s) \quad c \theta(h + h_0 - h_s) + z_s \quad 1]^T \quad (29)$$

Its velocity and acceleration can be derived:

$$\dot{\mathbf{q}}_0^{m2} = \begin{bmatrix} 0 \\ -\dot{\theta} c \theta(h + h_0 - h_s) - s \theta \dot{h} \\ -\dot{\theta} s \theta(h + h_0 - h_s) + c \theta \dot{h} + \dot{z}_s \\ 1 \end{bmatrix} \quad (30)$$

$$\ddot{\mathbf{q}}_0^{m2} = \begin{bmatrix} 0 \\ \ddot{\theta} s \theta(h + h_0 - h_s) - \dot{\theta} c \theta(h + h_0 - h_s) - \dot{\theta} c \theta \dot{h} - \dot{\theta} c \theta \dot{h} - s \theta \ddot{h} \\ -\ddot{\theta} c \theta(h + h_0 - h_s) - \dot{\theta} s \theta(h + h_0 - h_s) - \dot{\theta} s \theta \dot{h} - \dot{\theta} s \theta \dot{h} + c \theta \ddot{h} + \ddot{z}_s \\ 1 \end{bmatrix} \quad (31)$$

For formulating the dynamic equation of m_2 , its acceleration vector is transferred into frame 2:

$$\mathbf{p}_2^{\ddot{m}2} = \mathbf{R}_2^0 \mathbf{p}_0^{\ddot{m}2} = \begin{bmatrix} 0 \\ -\ddot{\theta}(h + h_0 - h_s) - 2 \dot{\theta} \dot{h} + s \theta \ddot{z}_s \\ -\ddot{\theta}^2(h + h_0 - h_s) + \ddot{h} + c \theta \ddot{z}_s \end{bmatrix} \quad (32)$$

In the initial condition, the spring in the bottom-layer is compressed to support the gravity of m_1 and m_2 . Thus, the preload force of spring is:

$$f_p = (m_1 + m_2)g \quad (33)$$

The gravity of m_2 in frame 0 is:

$$\mathbf{G}_0^{m2} = [0 \quad 0 \quad -m_2 g]^T \quad (34)$$

The gravity of m_2 in frame 2 is:

$$\mathbf{G}_2^{m2} = \mathbf{R}_2^0 \mathbf{G}_0^{m2} = \begin{bmatrix} 0 \\ -s\theta m_2 g \\ -c\theta m_2 g \end{bmatrix} \quad (35)$$

Because the active actuator of the bottom layer can only control the suspension's dynamic along the z_2 axis, the dynamic equation of m_2 along the z_2 axis is built:

$$m_2 \ddot{p}_{2(z2)}^{m2} = -k_s h - F_{rt} \operatorname{sgn}(\dot{h}) + u_t + f_p + G_2^{m2}(z2) - N \quad (36)$$

It can be rearranged as:

$$\begin{aligned} (m_2 + m_1) \ddot{h} = & -k_s h - F_{rt} \operatorname{sgn}(\dot{h}) + u_t \\ & - m_1 [-\dot{\theta}^2 (h + h_0) - \dot{\alpha}^2 h_b c\beta - \ddot{\alpha} h_b s\beta + c\theta \ddot{z}_s + c\theta g - g] \\ & - m_2 [-\dot{\theta}^2 (h + h_0 - h_s) + c\theta \ddot{z}_s - g + c\theta g] \end{aligned} \quad (37)$$

The dynamic of the multiple-DOF seat suspension is determined by equations (27) and (37).

4. Controller design

The roll movement and vertical vibration are controlled independently in this research. This control strategy requires the controllers to be robust enough to overcome disturbance from the system coupling.

In terms of the practical implementation, the feedback variables of the controller should be measurable in application. The acceleration of m_2 and the dynamic displacement h , which can be easily obtained, are applied to control the bottom-layer. The roll angles of the multiple-DOF seat suspension base and top are used for controlling the top-layer. The roll angle cannot be measured directly, but it can be accurately estimated by MEMS (microelectromechanical systems) IMU (inertial measurement unit) sensors with proper algorithms. The IMU sensor consists of an accelerometer to measure three axes accelerations and a gyroscope to measure three axes rotational rates.

4.1 Roll vibration controller

Because the roll angle of the top seat suspension can be estimated, if we know a desired roll angle which can decrease the lateral acceleration, the tracking control can be applied and there are many methods to do this. The decision as to the desired roll angle can follow two rules, least lateral displacement and least roll angle, as shown in Figure 6. The first rule causes the centre of driver mass to have the least lateral displacement, thus, the desired angle is defined as:

$$\alpha_1 = -\operatorname{asin}\left(\frac{h + h_0}{h_b} \sin(\theta)\right) \quad (38)$$

The second rule reduces the roll movement of the driver's body; thus the desired angle is $\alpha_2 = 0$.

Combining these two rules, the desired roll angle of the driver body is designed as:

$$\alpha_d = \varepsilon(\alpha_1 + \alpha_2) \quad (39)$$

where $\varepsilon \in [0 \ 1]$ is a weighting parameter.

Figure 6. Desired roll angle of driver body. (a) Least lateral displacement. (b) Least roll angle.

The dynamics of the roll movement are shown in equation (27) which is nonlinear and is coupling with the vertical vibration. The non-singular terminal sliding mode controller is selected to track the desired roll movement. The roll movement dynamic is rearranged as:

$$(I_{m1} + m_1 h_b^2) \ddot{\alpha} = -k_r(\alpha - \theta) + u_r + \omega_1 \quad (40)$$

$$\begin{aligned} \omega_1 = & -F_{rr} \text{sgn}(\dot{\alpha} - \dot{\theta}) \\ & - m_1 \{ s\beta [\dot{\theta}^2(h + h_0) - \ddot{h}] + c\beta [\ddot{\theta}(h + h_0) + 2\dot{\theta}\dot{h}] - s\alpha \ddot{z}_s \\ & - s\alpha g \} h_b \end{aligned} \quad (41)$$

where ω_1 is assumed to have a bound as ξ_1 .

The tracking error and its derivative value is:

$$e = \alpha_d - \alpha, \quad \dot{e} = \dot{\alpha}_d - \dot{\alpha} \quad (42)$$

The non-singular sliding variable is selected as:

$$s = e + \frac{1}{\beta} \dot{e}^{p/q} \quad (43)$$

where $\beta > 0$, p, q ($p > q$) are positive odd integers, $1 < \frac{p}{q} < 2$.

Generally, the non-singular terminal sliding mode controller is designed as:

$$u_r = k_r(\alpha - \theta) + \eta \text{sgn}(s) + (I_{m1} + m_1 h_b^2) (\ddot{\alpha}_d + \beta \frac{q}{p} \dot{e}^{2-p/q}) \quad (44)$$

where $\eta > \xi_1 > 0$.

With this controller, theoretically, the sliding mode variable will converge to zero quickly without singular problem [25]. In practical implementation, it will have difficulties. The sliding variable s and the desired roll acceleration $\ddot{\alpha}_d$ are all derived from the roll vibration source θ which is estimated from the IMU sensor. The estimation error and measurement noise already exist in the value of θ . It is well known that the derivative of a measurement value will amplify its error and noise; double derivative will severely deteriorate its accuracy. Thus, high magnitude of noise may exist in $\ddot{\alpha}_d$. In addition, $\ddot{\alpha}_1$ is bounded and $\ddot{\alpha}_2 = 0$, so $\ddot{\alpha}_d(I_{m1} + m_1 h_b^2)$ should have a bound ξ_2 . The roll movement controller is designed as:

$$u_r = k_r(\alpha - \theta) + \eta \text{sgn}(s) + (I_{m1} + m_1 h_b^2) \beta \frac{q}{p} \dot{e}^{2-p/q} \quad (45)$$

where $\eta > \xi_1 + \xi_2 > 0$.

For further decreasing the chattering of the controller, the sign function is implemented as a hyperbolic function:

$$\text{sgn}(s) \approx \tanh(\varepsilon s) \quad (46)$$

where $\varepsilon > 0$ is applied to avoid the dramatically change when s is around 0.

Analysis of stability shows

$$\begin{aligned} \dot{s} &= \dot{e} + \frac{1}{\beta} \frac{p}{q} \dot{e}^{p/q-1} (\ddot{\alpha}_d - \ddot{\alpha}) \\ &= \dot{e} + \frac{1}{\beta} \frac{p}{q} \dot{e}^{p/q-1} \left(\ddot{\alpha}_d - \frac{1}{I_{m1} + m_1 h_b^2} (-k_r(\alpha - \theta) + u_r + \omega_1) \right) \\ &= \frac{1}{\beta} \frac{p}{q} \dot{e}^{p/q-1} \left(\ddot{\alpha}_d - \frac{1}{I_{m1} + m_1 h_b^2} (\eta \text{sgn}(s) + \omega_1) \right) \\ &= \Psi(-\eta \text{sgn}(s) - \omega_1 + \ddot{\alpha}_d(I_{m1} + m_1 h_b^2)) \end{aligned} \quad (47)$$

where $\Psi = \frac{1}{I_{m1} + m_1 h_b^2} \frac{1}{\beta} \frac{p}{q} \dot{e}^{p/q-1}$.

Because $1 < \frac{p}{q} < 2$, $0 < \frac{p}{q} - 1 < 1$. When $\dot{e} \neq 0$,

$$\dot{e}^{p/q-1} > 0 \quad (48)$$

$$\dot{s}s = \Psi(-\eta|s| - \omega_1 s + \ddot{\alpha}_d(I_{m1} + m_1 h_b^2)s) < \Psi(-\eta|s| + (\xi_1 + \xi_2)s) < 0 \quad (49)$$

thus, the sliding condition is satisfied.

When $\dot{e} = 0$, from equations (40) and (45), we get:

$$(I_{m1} + m_1 h_b^2) \ddot{\alpha} = \eta \text{sgn}(\alpha_d - \alpha) + \omega_1 \quad (50)$$

When $\alpha_d > \alpha$, $\ddot{\alpha} > 0$; when $\alpha_d < \alpha$, $\ddot{\alpha} < 0$; when $\alpha_d = \alpha$, $s = 0$ can be obtained in a finite time.

4.2 Vertical vibration controller

The controller for the bottom-layer suspension developed in [26] is proven to be robust with friction disturbance. This controller applies the acceleration and the relative displacement of the vertical seat suspension as feedback signals. In practice, the roll angle of the cab floor is limited by the vehicle suspension in a small range in order to keep the stability of the vehicle, thus the relative rotary angle of the roll joint is also bounded within a certain range. We assume that $c\beta = c\theta = 1$, $s\beta = 0$, equation (37) can be simplified as:

$$M(\ddot{h} + \ddot{z}_s) = -k_s h - F_{rt} \text{sgn}(\dot{h}) + u_t + F_c \quad (51)$$

$$F_c = -m_1[-\dot{\theta}^2(h + h_0) - \dot{\alpha}^2 h_b] - m_2[-\dot{\theta}^2(h + h_0 - h_s)] \quad (52)$$

where $M = m_2 + m_1$. This model is similar to a single-DOF vertical active seat suspension except the centripetal forces F_c .

Now, we can control it with the same method for a single-DOF vertical active seat suspension as shown in Figure 7. Assume that $a_z = \ddot{h} + \ddot{z}_s$ is the measured vertical acceleration of the plate between top and bottom layers; due to the roll vibration, there is a small different between a_z and $\ddot{h} + \ddot{z}_s$. The state variables are chosen as $x_1 = h$, $x_2 = \dot{h}$. The measurement variables are $\mathbf{Y}_1 = [x_1 \ x_2]^T$ and $Y_2 = \ddot{z}_s$. Thus, the system model is defined as:

$$\dot{\mathbf{X}} = \mathbf{A}\mathbf{X} + \mathbf{B}_1(u + F_d) + \mathbf{B}_2 d \quad (53)$$

$$\mathbf{Y}_1 = \mathbf{C}_1 \mathbf{X} \quad (54)$$

$$Y_2 = \mathbf{C}_2 \mathbf{X} + D_2(u + F_d) \quad (55)$$

where $F_d = F_c - F_{rt} \text{sgn}(\dot{h})$, $d = \ddot{z}_s$, $\mathbf{X} = [x_1 \ x_2]^T$, $\mathbf{A} = \begin{bmatrix} 0 & 1 \\ -\frac{k_s}{M} & 0 \end{bmatrix}$, $\mathbf{B}_1 = \begin{bmatrix} 0 \\ \frac{1}{M} \end{bmatrix}$, $\mathbf{B}_2 = \begin{bmatrix} 0 \\ -1 \end{bmatrix}$, $\mathbf{C}_1 = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}$, $\mathbf{C}_2 = \begin{bmatrix} -\frac{k_s}{M} & 0 \end{bmatrix}$, $D_2 = \frac{1}{M}$.

The vertical acceleration a_z is the main optimization objective in the controller design process. In addition, the vehicle's operation devices, such as the steering wheel and the gear shift lever, are always fixed on the vehicle cab floor, thus, the suspension deflection, x_1 , should be also controlled to make sure the driver can easily handle those devices and to avoid bottoming. Therefore, the controlled output is defined as:

$$\mathbf{Z} = \begin{bmatrix} Z_1 \\ Z_2 \end{bmatrix} = \mathbf{C}_3 \mathbf{X} + D_3(u + F_d) \quad (56)$$

where $Z_1 = a_z$, $Z_2 = x_1$, $\mathbf{C}_3 = \boldsymbol{\varphi} \begin{bmatrix} -\frac{k_s}{M} & 0 \\ 1 & 0 \end{bmatrix}$, $D_3 = \boldsymbol{\varphi} \begin{bmatrix} \frac{1}{M} \\ 0 \end{bmatrix}$, $\boldsymbol{\varphi} = \begin{bmatrix} \varphi_1 & 0 \\ 0 & \varphi_2 \end{bmatrix}$ is a weighting matrix.

Figure 7. Simplified vertical vibration mode

The disturbance observer can be designed as:

$$\dot{\hat{F}}_d = L(Y_2 - \mathbf{C}_2\mathbf{X} + D_2(u + F_d)) \quad (57)$$

where $L > 0$ is the observer gain, $\hat{F}_d$ is the observed disturbance.

Then the observed disturbance is compensated into a H_∞ controller:

$$u_t = \mathbf{K}\mathbf{X} - \hat{F}_d \quad (58)$$

where $\mathbf{X} = [x_1 \ x_2]^T$, $\mathbf{K} = [k_1 \ k_2]$ is a controller gain vector.

The design of observer and controller can be found from [26]; the detailed design procedure of the controller, is not presented here for saving space.

5. Evaluation of the proposed method

In this section, the proposed control method is validated with both simulations and experiments.

5.1 Simulation

For comparison, a conventional single-DOF passive seat suspension with same dimension is applied in the simulation. Unlike the active seat suspension, the passive seat suspension has a damper c_s instead of friction to dissipate vibration energy. Based on the kinematic model, the acceleration of the driver body in its own frame is:

$$\mathbf{p}_4^{\ddot{\mathbf{m}}1} = \begin{bmatrix} 0 \\ -[\ddot{\theta}(h + h_0 + h_b) + 2\dot{\theta}\dot{h}] + s\theta\ddot{z}_s \\ -[\dot{\theta}^2(h + h_0 + h_b) - \ddot{h}] + c\theta\ddot{z}_s \end{bmatrix} \quad (59)$$

Because the passive seat suspension does not have a DOF in the roll direction, the roll acceleration of the vehicle body will totally be transferred to the driver body. Thus, $\ddot{\alpha} = \ddot{\theta}$.

The dynamic of the seat suspension can be defined as:

$$(m_2 + m_1)\ddot{h} = -k_s h - c_s \dot{h} + u_t - m_1[-\dot{\theta}^2(h + h_0 + h_b) + c\theta\ddot{z}_s + c\theta g - g] - m_2[-\dot{\theta}^2(h + h_0 - h_s) + c\theta\ddot{z}_s - g + c\theta g] \quad (60)$$

Table 2 shows the parameters of the multiple-DOF active seat suspension and the single-DOF passive seat suspension. And the designed controller parameters are listed in

Table 3.

Table 2. Model parameters

m_1	80 kg	h_b	0.3 m
m_2	8 kg	F_{rr}	8 Nm
k_s	5000 N/m	F_{rt}	80 N
h_0	0.2 m	k_r	700 Nm/rad
h_s	0.05 m	I_{m1}	2 kg*m ²
c_s	2000 Ns/m		

Table 3. Controller parameters

ε	0.5	k_1	4500
p	5	k_2	-200
q	3	L	3500
β	10	η	20

The bump vibration is always applied to evaluate the transient response of a system. For testing the roll vibration controller, a roll vibration is defined as:

$$\theta = \begin{cases} a + a \sin\left(2\pi f t + \frac{3\pi}{2}\right) & (t \leq \frac{1}{f}) \\ 0 & (t > \frac{1}{f}) \end{cases} \quad (61)$$

where $f = 1\text{Hz}$ and $a = 0.01745 \text{ rad } (1^\circ)$.

In the simulation, the lateral and roll accelerations of the multiple-DOF active seat are greatly reduced when compared with the passive one, which are shown in Figure 8 and Figure 9. The top-layer suspension deflection and the actuator torque output are shown in Figure 10 and Figure 11, respectively.

The results indicate that by applying the proposed control method, the system can have a quick response to the “bump” vibration in roll direction.

Figure 8. Lateral acceleration.

Figure 9. Roll acceleration.

Figure 10. Deflection of the top layer suspension.

Figure 11. Control torque.

For further evaluating the system performance, a random vertical vibration and a random roll vibration are exerted on the multiple-DOF active seat suspension models (27) and (37) and the single-DOF passive seat suspension model (60), simultaneously.

The roll acceleration (Figure 12), roll velocity (Figure 13), vertical acceleration (Figure 14) and lateral acceleration (Figure 15) of the driver's body with multiple-DOF active seat suspension and single-DOF passive seat suspension are presented. Their RMS values are shown in Table 4 which indicates that the vertical and lateral accelerations can be controlled simultaneously with the proposed Multiple-DOF seat suspension and control method.

Figure 12. Roll acceleration

Figure 13. Roll velocity

Figure 14. Vertical acceleration

Figure 15. Lateral acceleration

Table 4. Simulation result

RMS value	Passive	Active	Reduction (%)
Roll acceleration $\ddot{\alpha}$	1.972 rad/s ²	0.8197 rad/s ²	58.4
Roll velocity $\dot{\alpha}$	0.1429 rad/s	0.0468 rad/s	67.2
Vertical acceleration	1.26 m/s ²	0.3137 m/s ²	75.1
Lateral acceleration	0.99 m/s ²	0.563 m/s ²	43.1

The coupling effect between the rotational vibration and vertical vibration can be seen in (51) where the centripetal force F_c is the coupling term. Figure 16 shows the centripetal force in the simulation which indicates that the rotational vibration has a relatively small effect to the vertical dynamic, considering that the friction force can reach 80 N.

Figure 16. Centripetal force.

5.2 Experiments

The proposed controller is implemented on the multiple-DOF active seat suspension prototype in the lab. The experimental setup is shown in Figure 17 where a multiple-DOF active seat, a single-DOF active seat and a conventional passive seat are fixed on a 6-DOF vibration platform, respectively, and the same load is applied in the comparison experiments. The schematic diagram of the whole system is shown in Figure 18. The 6-DOF vibration platform, also called the Stewart platform, can generate vibration in six DOFs based the predesigned vibration profile which is sent to the controller (NI CompactRio 9076 with two NI 9401 modules) from a computer. The multiple-DOF active seat suspension is controlled by an NI CompactRio 9074 with three modules, namely TB 3501, NI 9205 and NI 9264; and a computer is applied to set the controller parameters and record experimental data. With a Serial Peripheral Interface (SPI) bus, the TB 3501 module can read the data of MPU 9250. The analog input module NI 9205 can get the feedback from an acceleration sensor ADXL 203EB and a displacement sensor optoNCDT 1302. Based on the feedback, the desired active force and torque are calculated, and the corresponding commands are sent to their motor drives. The 6 DOFs vibration of the seat surface is measured by a XSSENS sensor which is one of the best performing MENS based sensor on the market. Figure 19 shows the location of sensors where the optoNCDT 1302 is used to measure the relative displacement of the bottom layer; the ADXL 203EB is placed under the plate between top and bottom layers; two MPU 9250s are set on the top and base of the Multiple-DOF seat suspension, respectively.

Figure 17. Experimental setup

Figure 18. Schematic diagram of experimental setup

Figure 19. Location of sensors

MPU 9250 is a low cost IMU which contains a 3-axis gyroscope, a 3-axis accelerometer and a 3-axis digital compass. In this research, the accelerometer and gyroscope data are applied to estimate the roll angle. First, the calibration techniques in [30] have been implemented to deal with the sensor errors like bias, scale factor and nonorthogonality. Then, with the calibrated data, the rotary angles of the seat suspension's base and top platform are estimated based on an orientation filter [31].

In the experiment, the random vertical vibration is generated by inputting a random road profile into a quarter-car model, then the vibration of the sprung mass is applied to the seat bottom; and the random

roll vibration is composed of several sinusoidal vibrations with different frequencies and amplitudes. The vibration in the two DOFs is generated by a 6-DOF vibration platform, simultaneously. The XSENS sensor can obtain the vibration information of the three seat surfaces which include the 3-axis rotary velocities, 3-axis accelerations and estimated 3-axis rotary angles; all the measured data are referred to the fixed reference coordinate. Figure 20 shows the tested roll velocity of the multiple-DOF active, single-DOF active, and conventional passive seat surfaces; the RMS value of the multiple-DOF seat is 0.126 rad/s, the passive one is 0.289 rad/s, and the single-DOF active one is 0.298; there is a 57.7% reduction when the multiple-DOF active one is compared with the single-DOF active one. The roll velocity can indicate the roll vibration magnitude of the seat surface. In Figures 21 and 22, the vertical and lateral accelerations are presented. The results indicate that the multiple-DOF active seat has similar performance with the single-DOF active one in the vertical vibration isolation, but it has better performance in the lateral vibration control. In Figure 23, the power spectral density (PSD) of the lateral acceleration shows the multiple-DOF active seat suspension has the best performance. The torque output which has been amplified by the gear box is shown in Figure 24. **The vertical control force shows in Figure 25 where the multiple-DOF active seat suspension requires a bigger vertical control force; it is partly caused by the mass increase of the two-layer seat suspension.** By referring to the value of the seat surface roll velocity in Figure 18, we can assume that the maximum relative roll velocity is 1 rad/s, thus the maximum power is about 40 W. For evaluating the improvement of the multiple-DOF active seat suspension in health and ride comfort, the FW-RMS and VDV are calculated based on ISO 2631. Table 5 shows the comparison of the vibrations in the three seat surfaces. The FW-RMS vertical accelerations of the multiple-DOF and the single-DOF active seats are very close. This indicates the applied H_∞ controller with disturbance compensation is also effective under multiple-DOF vibration conditions. When compared with the conventional passive seat, the reduction is 32.1%. The FW-RMS lateral acceleration has reductions of 49.4% and 51.3%, when compared with the conventional passive seat and single-DOF active seat, respectively. **The lateral accelerations of the conventional passive seat and the single-DOF active one have a small difference which is mainly caused by the prototype manufacture; the single-DOF active seat suspension has a higher degree of wear than the new passive one, thus it is easier to sway with roll vibration.** By assuming that there is no longitudinal acceleration (a_{wx}), in Table 6, the vibration total value of FW-RMS acceleration is calculated with the suggested weighting multiplying factors for health and comfort. The results indicate that with the multiple-DOF active seat suspension there is a 29.8% improvement in driver health, and a 23.6% improvement in ride comfort, when compared with the single-DOF active seat suspension.

Figure 20. Roll velocity of seat surface

Figure 21. Vertical acceleration of seat surface

Figure 22. Lateral acceleration of seat surface

Figure 23. PSD of lateral acceleration.

Figure 24. Torque output for roll vibration control.

Figure 25. Vertical control force.

Table 5. Seat surface vibration comparison

		Conventional passive	Single-DOF active	Multile- DOF active	Reduction (%)	
					With conventional passive	With single- DOF active
Vertical acceleration	RMS (m/s^2)	1.113	0.6718	0.647	41.9	3.7
	FW-RMS (m/s^2)	0.895	0.597	0.608	32.1	-1.8
	VDV ($\text{m/s}^{1.75}$)	2.601	1.710	1.748	32.8	-2.2
Lateral acceleration	RMS (m/s^2)	1.298	1.310	0.579	55.4	55.8
	FW-RMS (m/s^2)	0.928	0.966	0.47	49.4	51.3
	VDV ($\text{m/s}^{1.75}$)	2.392	2.507	1.144	52.2	54.4

Table 6. Vibration total value of FW-RMS acceleration

	Conventional passive (m/s^2)	Single-DOF active	Multiple-DOF active (m/s^2)	Reduction (%)	
				With conventional passive	With single- DOF active
Health	2.131	1.478	1.037	51.3	29.8
Comfort	1.710	1.136	0.868	49.2	23.6

6. Conclusion

In this paper, a method for controlling roll vibration and vertical vibration of heavy duty vehicle seat suspension has been proposed and validated. The single-DOF active seat suspension can only suppress the vertical vibration but high magnitude lateral acceleration of the driver's body may be introduced by the roll vibration of the vehicle caused by the uneven road under both sides' tyres. A two-layer multiple-DOF seat suspension prototype has been applied in this study; its roll movement and vertical movement can be independently controlled. By assuming that there is only movement in the y-z coordinate, the model of the seat suspension prototype has been built. An H_∞ controller with disturbance compensation is applied to control the bottom layer for vertical vibration isolation. A non-singular terminal sliding controller is designed to control the top layer by tracking a desired roll angle which will reduce the lateral acceleration and roll acceleration. Two IMU sensors MPU9250 have been carefully calibrated and their algorithm parameters have been tuned for accurately estimating the roll angles of the top and base platforms of the seat prototype. The z-axis DOF and roll DOF of the Multiple-DOF seat suspension are controlled independently with practical measurable feedback. This kind of decoupled control strategy will benefit the controller design and implementation. The simulation and experiment have both validated the effectiveness of the proposed method. A well-tuned conventional passive seat suspension and a single-DOF active seat suspension have been applied in the experiment for comparison. The roll vibration and lateral vibration of the seat surface can be greatly reduced by controlling the roll movement of the top layer. Based on ISO 2631, the FW-RMS accelerations of the multiple-DOF active seat surface has a 51.3% reduction in lateral direction when compared with the single-DOF active seat suspension; and the two seat suspensions have similar performances in vertical vibration control. The vibration total value of FW-RMS acceleration of seat surface has 29.8% and 23.6% reductions for evaluating its influence on health and ride comfort, respectively. The proposed method can greatly improve the work conditions for heavy duty vehicle drivers.

In future study, the design parameters of the multiple-DOF active seat suspension prototype will be optimised. The roll and pitch movement of top-layer of the suspension will be controlled together for considering more complex work scenarios.

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