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Vibration control of seat considering vehicle suspension and human-body models

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Abstract

Vehicle seat suspension is one of very important components to provide ride comfort, in particular, commercial vehicles, to reduce driver fatigue due to long hours driving. This paper presents a study on active control of seat suspension to reduce vertical vibration transmitted from uneven road profile to driver body. The control problem will be firstly studied by proposing an integrated seat suspension model which includes vehicle chassis suspension, seat suspension, and driver body model. This is a new concept in the field of study because most of the current active and semi-active seat suspension studies only consider seat suspension or seat suspension with human body model, and road disturbance is generally assumed to be applied to the cabin floor directly. Controller design based an integrated model will enable the seat suspension to perform in a scenario where vibration caused by road disturbance is transmitted from wheel to seat frame and ride comfort performance is evaluated in terms of human body instead of seat frame acceleration. A static output feedback controller is then designed for the seat suspension with using measurement available signals. Driver mass variation and actuator saturation are also considered in the controller design process. The conditions for designing such a controller are derived in terms of linear matrix inequalities (LMIs). Finally, numerical simulations are used to validate the effectiveness of the proposed control strategy. It is shown from the driver body acceleration responses under both bump and random road disturbances that the newly designed seat suspension can improve vehicle ride comfort regardless of driver body mass variation.

Keywords

suspension, human, body, models, vibration, seat, control, considering, vehicle

Disciplines

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Vibration Control of Seat Considering Vehicle Suspension and Human-body Models

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Abstract

Vehicle seat suspension is one of very important components for providing ride comfort to passengers, in particular, for reducing fatigue of professional drivers with long hours driving. In this paper, active control strategy will be applied to reduce the vertical vibration of driver body with focusing on the following aspects. Firstly, the control problem will be studied by considering an integrated vehicle suspension, seat suspension, and human body model. This is a new idea in the field of study as most current active and semi-active seat suspension studies only consider seat suspension or seat suspension with human body model, and road disturbance is generally assumed to be applied to the cab directly. Secondly, as a controller needs to use available information as feedback signals, state feedback may not be able to do this when some signals are not available for measurement, for example, head displacement, cushion displacement, etc. This paper will present static output feedback control design for seat suspension. Thirdly, driver load variation will be considered as different drivers may have different weights. Taking weight variation into account will make the controller have similar performance for different drivers. At last, actuator saturation is considered. This is a practical constraint that needs to be dealt with when implementing a controller in practice.

Key words: Seat suspension control, integrated model, static output feedback, actuation saturation, load variation

1. Introduction

Seat suspension has been commonly accepted in commercial vehicles for industrial, agricultural and other transport purposes [1] to provide driver ride comfort, to reduce driver fatigue due to long hour driving or exposure to severe working environment such as rough road condition, and to improve driver safety and health [2]. Study on optimisation and control of seat suspensions for reducing vertical vibration has been an active topic for decades. Three main types of seat suspensions, i.e., passive seat suspension, semi-active seat suspension, and active seat suspension, have been presented so far. The study on passive seat suspension mainly focuses on parameter optimisation for the spring stiffness and the damping coefficient. With the development of magnetorheological (MR) or electrorheological (ER) dampers, semi-active control of seat suspension has been proposed to provide variable damping force with less power consumption [1, 3]. The study on active seat suspension mainly focuses on developing advanced control strategies or applying different types of actuators to improve seat suspension performance with taking account of issues like actuator saturation, load variation, time delay, and reliability, etc. [4-7]. Among these three types of seat suspensions, active seat suspension is able to provide the best ride comfort performance, and therefore, receives much more attention in recent years.

In addition to seat suspension, vehicle suspension has been extensively studied for a long time [8]. Vehicle suspension is, in fact, designed as a primary suspension for all the vehicles to provide ride comfort, road holding, and other dynamic functions. However, it is noticed that most of the current active/semi-active seat suspension and active/semiactive vehicle suspension are designed/studied separately though their common function is to improve vehicle ride comfort performance.

In this paper, active control strategy will be applied to reduce the vertical vibration of driver body when sitting in a running vehicle. The main contributions are given in the following. Firstly, the control problem will be studied by considering an integrated vehicle suspension, seat suspension, and human body model. As vehicle suspension itself is designed to isolate vibration from road disturbance to vehicle body, considering vehicle suspension functions when designing a seat suspension system will obtain more accurate information on vibration sources and therefore reduce control effort for seat suspension in terms of attenuated excitation input. In addition, human body model is necessary to be included as acceleration reduction should be evaluated on human body not on the cabin floor. Secondly, as a controller needs to use available information as feedback signals, state feedback may not be able to do this when some signals are not available for measurement, for example, head displacement, cushion displacement, etc. The shortcoming of dynamic output feedback control is that its order will be higher, in particular, when high DOF of human-body model will be considered, which results in either higher order controller, which is hard to be implemented, or no feasible solution. From this point of view, this paper will present static output feedback control design for seat suspension. Some possible configurations in terms of available measurements will be further studied. Simulation results show that some signals are not helpful in vibration reduction even they are measurement available. Thirdly, driver load variation will be considered as different drivers may have different weights. Taking weight variation into account will make the controller have similar performance for different drivers. At last, actuator saturation is considered. This is a practical constraint that needs to be dealt with when implementing a controller in practice.

The notation used throughout the paper is fairly standard. For a real symmetric matrix W, the notation of $W > 0$ ($W < 0$) is used to denote its positive- (negative-) definiteness. $\|\cdot\|$ refers to either the Euclidean vector norm or the induced matrix 2-norm. I is used to denote the identity matrix of appropriate dimensions. To simplify notation, $*$ is used to represent a block matrix which is readily inferred by symmetry.

2. Integrated Vehicle Seat and Suspension Model

The integrated vehicle seat and suspension model includes a quarter-car suspension model, a seat suspension model, and a driver body model as shown in Figure 1, where m_s is the sprung mass, which represents the car chassis; m_u is the unsprung mass, which represents the wheel assembly; m_f is the seat frame mass; m_c is the seat cushion mass; and m_b is the driver body mass. z_u, z_s, z_f, z_c and z_b are the displacements of the corresponding masses, respectively; z_r is the road displacement input. c_s and k_s are damping and stiffness of the car suspension system, respectively; k_t and c_t stand for compressibility and damping of the pneumatic tyre, respectively; c_{ss}, c_{cc}, k_{ss} and k_c are damping and stiffness of seat suspension and seat cushion, respectively. u represents the active control force applied to the seat suspension.

The dynamic vertical motion of equations for the quarter-car suspension, seat suspension, and driver body are given by

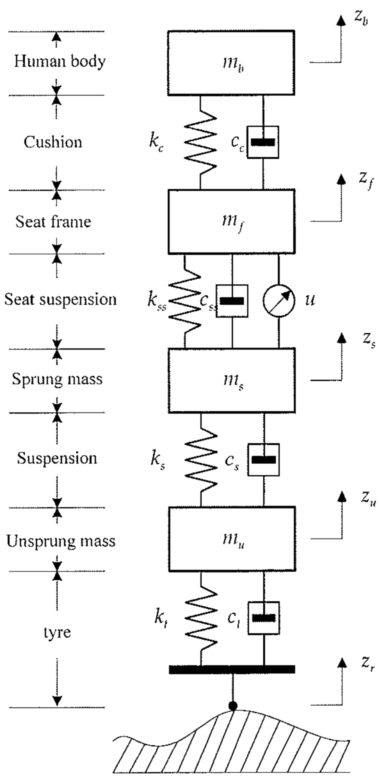

Fig. 1 Integrated car suspension, seat suspension, and driver body model.

$$
\begin{aligned}
m_u\ddot{z}_u &= -k_t(z_u - z_r) - c_t(\dot{z}_u - \dot{z}_r) + k_s(z_s - z_u) + c_s(\dot{z}_s - \dot{z}_u) \\
m_s\ddot{z}_s &= -k_s(z_s - z_u) - c_s(\dot{z}_s - \dot{z}_u) + k_{ss}(z_f - z_s) + c_{ss}(\dot{z}_f - \dot{z}_s) + u \\
m_f\ddot{z}_f &= -k_{ss}(z_f - z_s) - c_{ss}(\dot{z}_f - \dot{z}_s) + k_c(z_c - z_f) + c_c(\dot{z}_c - \dot{z}_f) - u \\
m_b\ddot{z}_b &= -k_c(z_c - z_f) - c_c(\dot{z}_c - \dot{z}_f)
\end{aligned} \tag{1}
$$

By defining the following set of state variables

$$x_1 = z_u - z_r,\ x_2 = \dot{z}_u,\ x_3 = z_s - z_u,\ x_4 = \dot{z}_s,\ x_5 = z_f - z_s,\ x_6 = \dot{z}_f,\ x_7 = z_c - z_f,\ x_8 = \dot{z}_b$$

and state vector $x = [x_1\ x_2\ x_3\ x_4\ x_5\ x_6\ x_7\ x_8]^{\mathrm{T}}$, we can write the dynamic equation (1) into a state-space form as

$$\dot{x} = Ax + B_w w + Bu \tag{2}$$

where

$$
A = \begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
-\frac{k_t}{m_u} & -\frac{c_t + c_s}{m_u} & \frac{k_s}{m_u} & \frac{c_s}{m_u} & 0 & 0 & 0 & 0 \\
0 & -1 & 0 & 1 & 0 & 0 & 0 & 0 \\
0 & \frac{c_s}{m_s} & -\frac{k_s}{m_s} & -\frac{c_s + c_{ss}}{m_s} & \frac{k_{ss}}{m_s} & \frac{c_{ss}}{m_s} & 0 & 0 \\
0 & 0 & 0 & -1 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & \frac{c_{ss}}{m_f} & -\frac{k_{ss}}{m_f} & -\frac{c_{ss} + c_c}{m_f} & \frac{k_c}{m_f} & \frac{c_c}{m_f} \\
0 & 0 & 0 & 0 & 0 & -1 & 0 & 1 \\
0 & 0 & 0 & 0 & 0 & \frac{c_c}{m_b} & -\frac{k_c}{m_b} & -\frac{c_c}{m_b}
\end{bmatrix}
$$

$$B=\begin{bmatrix}0 & 0 & 0 & \frac{1}{m_s} & 0 & -\frac{1}{m_f} & 0 & 0\end{bmatrix}^T, \quad B_w=\begin{bmatrix}-1 & \frac{c_t}{m_u} & 0 & 0 & 0 & 0 & 0 & 0\end{bmatrix}^T.$$

and the road disturbance $w=\dot{z}_r$.

In practice, all the actuators are limited by their physical capabilities, and hence, actuator saturation needs to be considered for active control of seat suspension [10]. Taking actuator saturation into account, equation (2) is modified as

$$\dot{x}=Ax+B_w w+B\bar{u} \tag{3}$$

where $\bar{u}=sat(u)$ and $sat(u)$ is a saturation function of control input u defined as

$$sat(u)=\begin{cases}-u_{\lim} & \text{if } u<-u_{\lim}\\ -u_{\lim} & \text{if } -u_{\lim}\le < u\le u_{\lim}\\ u_{\lim} & \text{if } u>u_{\lim}\end{cases} \tag{4}$$

As the varying driver body mass m is actually bounded by its minimum value mmin and its maximum value mmax in a real operation, it is not difficult to represent the uncertain vehicle mass by

$$1/m_b=M_1(\xi)m_{s\max}+M_2(\xi)m_{s\min},$$

where $\xi=1/m_b, m_{s\max}=1/m_{b\min}, m_{s\min}=1/m_{b\max}$, $M_1(\xi)$ and $M_2(\xi)$ are defined as

$$M_1(\xi)=\frac{1/m_b-m_{s\min}}{m_{s\max}-m_{s\min}}, M_2(\xi)=\frac{m_{s\max}-1/m_b}{m_{s\max}-m_{s\min}}$$

Hence, the parameter varying system for the integrated model can be expressed as

$$\dot{x}=\sum_{i=1}^{2}M_iA_ix+B_w w+B\bar{u}=A_m x+B_w w+B\bar{u} \tag{5}$$

where $A_m=\sum_{i=1}^{2}M_iA_i$.

3. Static Output Feedback Controller Design

For seat suspension design, the performance on ride comfort is mainly described by the driver body acceleration [5, 7], and therefore, the driver body acceleration,

$$z=\ddot{z}_b=C_1x \tag{6}$$

where C_1 is composed of the last row of A in (2), is defined as the control output.

To achieve good ride comfort and make the controller performing adequately for a wide range of road disturbances, the L2 gain between the road disturbance input w and the control output z, which is defined as

$$\|T_{zw}\|_\infty=\sup_{\|w\|_2\neq 0}\frac{\|z\|_2}{\|w\|_2} \tag{7}$$

where $\|z\|_2^2=\int_0^\infty z^Tzdt$ and $\|w\|_2^2=\int_0^\infty w^Twdt$, is chosen as the performance measure. A small value of $\|T_{zw}\|_\infty$ generally means a small value of driver body acceleration under the energy limited road disturbances. The control objective is to design a controller

$$u=\sum_{i=1}^{2}M_iK_iCx=K_mCx \tag{8}$$

where $K_m=\sum_{i=1}^{2}M_iK_i$, K_i is a constant state feedback gain matrix to be designed such that the closed-loop system, which is composed by substituting (8) into (5), is asymptotically stable, and the performance measure (7) is minimised.

For designing the controller (8), we define a Lyapunov function for the system (5) as

$$V(x)=x^TPx, \tag{9}$$

where P is a positive definite matrix. By differentiating (9), we obtain

$$\dot{V}(x) = \dot{x}^T Px + x^T P\dot{x} = [A_m x + B\frac{1+\varepsilon}{2}u + B(\overline{u} - \frac{1+\varepsilon}{2}u) + B_w w]^T Px$$
$$+x^T P[A_m x + B\frac{1+\varepsilon}{2}u + B(\overline{u} - \frac{1+\varepsilon}{2}u) + B_w w] \tag{10}$$

By using the following inequalities

$$\left[\overline{u} - \frac{1+\varepsilon}{2}u\right]^T \left[\overline{u} - \frac{1+\varepsilon}{2}u\right] \le \left(\frac{1-\varepsilon}{2}\right)^2 u^T u, \text{ for } |u| \le \frac{u_{\lim}}{\varepsilon} \text{ and } \varepsilon > 0 \tag{11}$$

$$X^T Y + Y^T X \le \sigma X^T X + \sigma^{-1} Y^T Y \tag{12}$$

and the definition (8), we have

$$\dot{V}(x) = x^T [A_m^T P + PA_m + (B\frac{1+\varepsilon}{2}K_m C)^T P + PB\frac{1+\varepsilon}{2}K_m C]x$$
$$+w^T B_w^T Px + x^T PB_w w + \sigma(\overline{u} - \frac{1+\varepsilon}{2}u)^T (\overline{u} - \frac{1+\varepsilon}{2}u) + \sigma^{-1} x^T PBB^T Px \tag{13}$$
$$\le x^T \Theta x + w^T B_w^T Px + x^T PB_w w$$

where

$$\Theta = A_m^T P + PA_m + (B\frac{1+\varepsilon}{2}K_m C)^T P + PB\frac{1+\varepsilon}{2}K_m C + \sigma(\frac{1+\varepsilon}{2})^2 C^T K_m^T K_m C + \sigma^{-1} PBB^T P$$

Adding $z^T z - \gamma^2 w^T w$ $\gamma > 0$ to both sides of (13 yields

$$\dot{V}(x) + z^T z - \gamma^2 w^T w = \begin{bmatrix} x^T & w^T \end{bmatrix} \begin{bmatrix} \Theta + C_1^T C_1 & PB_1 \\ B_w^T P & -\gamma^2 I \end{bmatrix} \begin{bmatrix} x \\ w \end{bmatrix} \tag{14}$$

Let us consider

$$\Pi = \begin{bmatrix} \Theta + C_1^T C_1 & PB_w \\ B_w^T P & -\gamma^2 I \end{bmatrix} < 0 \tag{15}$$

then $\dot{V}(x) + z^T z - \gamma^2 w^T w \le 0$, and the L_2 gain defined in (7) is less than $\gamma > 0$ with the initial condition $x(0) = 0$. When the disturbance is zero, i.e., $w = 0$, it can be inferred from (14) that if $\Pi < 0$, then $\dot{V}(x) < 0$, and the closed-loop system is quadratically stable.

Pre- and post-multiplying (15) by $\mathrm{diag}(P^{-1}, \ I)$ and its transpose, respectively, and defining $Q = P^{-1}$, $WC = CQ$, $Y_m = K_m Q$, the condition $\Pi < 0$ is equivalent to

$$\Sigma = \begin{bmatrix} QA_m^T + A_m Q + \frac{1+\varepsilon}{2}C^T Y_m^T B^T + \frac{1+\varepsilon}{2}BY_m C \\ +\sigma(\frac{1-\varepsilon}{2})^2 C^T Y_m^T Y_m C + \sigma^{-1} B\, B^T + QC_1^T C_1 Q & B_w \\ * & -\gamma^2 I \end{bmatrix} < 0, \tag{16}$$

Using the Schur complement, $\Sigma < 0$ is equivalent to

$$\Lambda = \begin{bmatrix} QA_m^T + A_m Q + \frac{1+\varepsilon}{2}[C^T Y_m^T B^T + BY_m C] + \sigma^{-1} BB^T & C^T Y_m^T & QC_1^T & B_w \\ * & -\sigma^{-1}(\frac{1-\varepsilon}{2})^2 I & 0 & 0 \\ * & * & -I & 0 \\ * & * & * & -\gamma^2 I \end{bmatrix} < 0 \tag{17}$$

By the definition $A_m = \sum_{i=1}^{2} M_i A_i$, and the fact that $M_i \ge 0$ and $\sum_{i=1}^{2} M_1 = 1$, $\Lambda < 0$ is equivalent to

$$\begin{bmatrix} QA_i^T + A_iQ + \frac{1+\varepsilon}{2}[C^TY_i^TB^T + BY_iC] + \sigma^{-1}BB^T & C^TY_i^T & QC_1^T & B_w \\ * & -\sigma^{-1}(\frac{1-\varepsilon}{2})^2 I & 0 & 0 \\ * & * & -I & 0 \\ * & * & * & -\gamma^2 I \end{bmatrix} < 0, \quad i = 1,2$$

(18)

On the other hand, from (8), the constraint $|u| \le \frac{u_{\lim}}{\varepsilon}$ can be expressed as

$$\left|\sum_{i=1}^{2} M_i(\xi)K_iCx\right| \le \frac{u_{\lim}}{\varepsilon} \tag{19}$$

It is induced that if $|K_iCx| \le \frac{u_{\lim}}{\varepsilon}$, then (19) holds. Let define a set of $\Omega(K) = \left\{x \,||\, x^TC^TK_i^TK_iCx| \le (\frac{u_{\lim}}{\varepsilon})^2\right\}$, then the equivalent condition for an ellipsoid $\Omega(P,\rho) = \{x \,|\, x^TPx \le \rho\}$ being a subset of $\Omega(K)$, i.e., $\Omega(P,\rho) \subset \Omega(K)$, is

$$K_iC\left(\frac{P}{\rho}\right)^{-1}C^TK_i^T \le \left(\frac{u_{\lim}}{\varepsilon}\right)^2 \tag{20}$$

By the Schur complement, inequality (20) can be written as

$$\begin{bmatrix} \left(\frac{u_{\lim}}{\varepsilon}\right)^2 I & K_iC\left(\frac{P}{\rho}\right)^{-1} \\ * & \left(\frac{P}{\rho}\right)^{-1} I \end{bmatrix} \ge 0 \tag{21}$$

Using definitions $Q = P^{-1}$, $WC = CQ$, and $Y_m = K_mQ$, inequality (21) is equivalent to

$$\begin{bmatrix} \left(\frac{u_{\lim}}{\varepsilon}\right)^2 I & Y_iC \\ * & \rho^{-1}Q \end{bmatrix} \ge 0 \tag{22}$$

Now, we consider the computational algorithm for implementing the proposed controller. It is observed from above that the controller design is the feasibility problem of linear matrix inequalities (LMIs) (18) and (22) with equality constraint $WC = CQ$. We now convert the equality constraint problem to the LMI problem [9]. Consider the equality constraint $WC = CQ$, it can be equivalently converted to

$$tr\left[(WC - CQ)^T(WC - CQ)\right] = 0 \tag{23}$$

By introducing the condition

$$(WC - CQ)^T(WC - CQ) \le \mu I \tag{24}$$

which is equivalent to

$$\begin{bmatrix} -\mu I & (WC - CQ)^T \\ * & -I \end{bmatrix} \le 0 \tag{25}$$

by means of the Schur complement, the design of the static output feedback controller can be changed to a problem of finding a global solution to the minimisation problem of:

minimise μ subject to LMIs (18), (22), and (25) for i = 1, 2. (26)

This minimisation problem can be solved by using the LMI Toolbox in Matlab. If μ equals zero, then the solutions will satisfy the LMIs (18) and (22), and the equality $WC = CQ$, and then, the static output feedback controller can be obtained as $K_i = Y_iW^{-1}$.

4. Numerical Simulations

Numerical simulations are conducted in this section to show the effectiveness of the proposed integrated seat and suspension model and control strategy for improving driver ride comfort. The parameters used in the simulations are listed in Table 1, where the quarter-car suspension parameters have been optimised in terms of driver body acceleration in [10] and the seat suspension and driver body model parameters are referred to [3].

Table 1. Parameter values of the proposed suspension model

Mass	kg	Damping	Ns/m	Spring	N/m
m_u	20	c_t	0	k_t	180000
m_s	300	c_s	2000	k_s	10000
m_f	20	c_{ss}	1080	k_{ss}	7414.86
m_b	70	c_c	152.8	k_c	8228.78

In the simulation, the actuator force limitation for the seat suspension is considered as u_{lim}=1000 N. Driver body mass is assumed to be in the range of 40 kg to 100kg, and the nominal mass is 70 kg as listed in Table 1.

To validate the suspension performance in time-domain, a typical road disturbance, i.e., bump road disturbance, will be considered in the simulation and applied to the vehicle wheel. The ground displacement for an isolated bump in an otherwise smooth road surface is given by

$$z_r = \begin{cases} \frac{a}{2}\left(1-\cos\left(\frac{2\pi v_0}{l}t\right)\right), & 0 \le t \le \frac{l}{v_0} \\ 0, & t > \frac{l}{v_0} \end{cases}$$

where a and l are the height and the length of the bump, v_0 is vehicle forward speed. We choose a= 0.1 m, l= 2 m, and v_0=30 km/h in the simulation.

To show the effectiveness and advance of the proposed control strategy, several different controllers will be designed and compared. At first, we design a state feedback controller with choosing the scalars ε= 0.9 and ρ= 10^{-3} as K $=10^5 \times$ [0.2221 0.0005 0.2341 0.0191 0.0910 0.0484 -2.4017 0.3043]. The state feedback controller is designed by setting C_1 as an identity matrix, and solving LMIs (18) and (22) for i=1. This controller does not consider the driver body mass variation. Then, we design different static output feedback controllers with assuming different measurable state variables. When assuming both seat suspension displacement and velocity are measurable, we define output matrix as C_1=[0 0 0 0 1 0 0 0;0 0 0 0 0 1 0 0], and the static output feedback controller is obtained with choosing the scalars ε= 0.993 and ρ= 1 as K_1 $=10^4 \times$ [0 0 0 0 -0.5008 1.7497 0 0] and K_2 $=10^4 \times$ [0 0 0 0 -0.4326 2.2141 0 0].

The bump responses of the driver body acceleration for the integrated seat and suspension system with the above-designed two controllers are compared in Fig. 2, where Passive means no controller has been used. It can be seen from Fig. 2 that the state feedback control achieves the best performance on ride comfort in terms of the peak value of driver body acceleration, and the static output feedback controller achieves similar performance to the state feedback controller in spite of its simple structure.

We now consider some other possible cases for designing a static output feedback controller.

Case 1: Car suspension displacement and velocity are measurable, output matrix is defined as C_1=[0 0 1 0 0 0 0 0;0 0 0 1 0 0 0 0], and the static output feedback controller is obtained as K_1 $=10^3 \times$ [0 0 -2.2272 0.6235 0 0 0 0]; K_2 $=10^3 \times$ [0 0 -0.2027 0.6273 0 0 0 0].

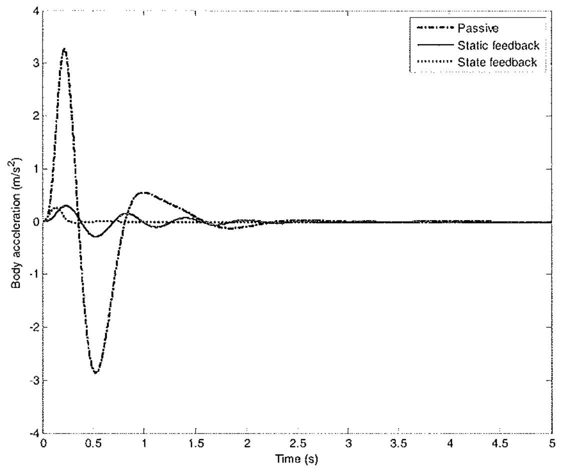

Fig. 2 Bump responses on driver body acceleration for different control systems.

Case 2: Car suspension displacement and seat suspension displacement are measurable, output matrix is defined as C_1=[0 0 1 0 0 0 0 0;0 0 0 1 0 0 0 0], and the static output feedback controller is obtained as $K_1 = 10^3 \times$ [0 0 0.6746 0 -4.7360 0 0 0]; $K_2 = 10^3 \times$ [0 0 4.6527 0 -2.8969 0 0 0].

Case 3: Car suspension displacement and seat suspension displacement are measurable, output matrix is defined as C_1=[0 0 0 1 0 0 0 0;0 0 0 0 0 1 0 0], and the static output feedback controller is obtained as $K_1 = 10^5 \times$ [0 0 0 -0.1265 0 4.2378 0 0]; $K_1 = 10^5 \times$ [0 0 0 -0.1066 0 3.5590 0 0].

The comparison of the designed static output feedback controllers with passive seat suspension on bump responses is shown in Figs. 3, 4, 5, respectively, for three cases. It can be seen that both Case 1 and Case 2 do not achieve too much improved ride comfort performance compared the passive seat suspension, however, Case 3 achieves an improved ride comfort performance compared to the passive seat suspension. These results may indicate that we could to choose the most appropriate measurement available signals to provide the best control action so that the ride comfort performance can be improved. This, however, may need further investigation.

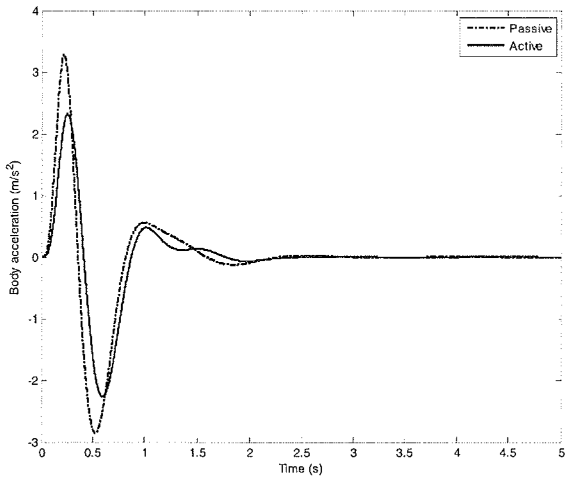

Fig. 3 Bump responses on driver body acceleration for Case 1.

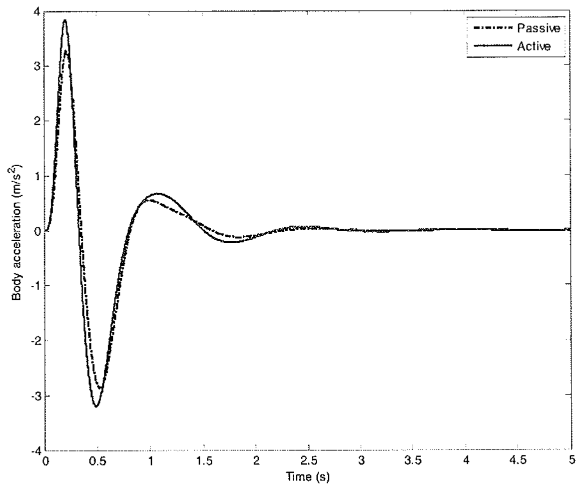

Fig. 4 Bump responses on driver body acceleration for Case 2.

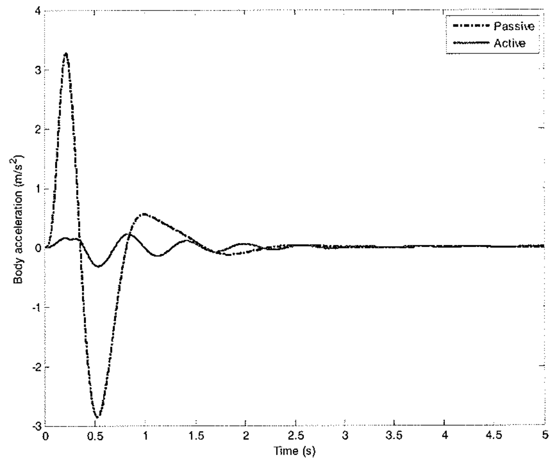

Fig. 5 Bump responses on driver body acceleration for Case 3.

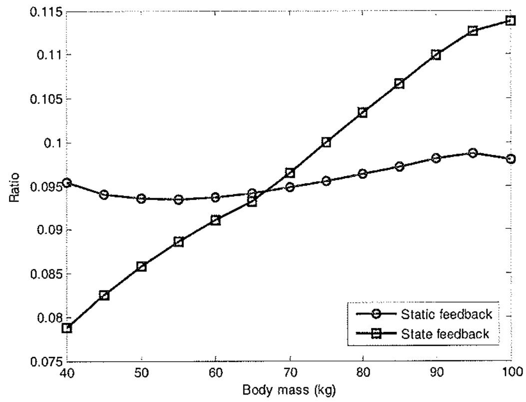

Fig. 6 Bump responses on driver body acceleration with varying driver body mass.

At last, we check the robustness of the designed controller by changing the driver body mass. The comparison result between the state feedback controller and the static output feedback controller, which uses seat suspension displacement and velocity as feedback signals, is shown in Fig. 6. The driver body mass is varied from 40 kg to 100kg, and the ratio is defined as the peak acceleration ratio between the seat suspension with the indicated controller and the passive seat suspension. It can be seen that the static feedback controller can keep the ride comfort performance unchanged in spite of the changes of driver body mass. The ride comfort performance realized by the state feedback controller, which does not consider the driver body mass variation in the controller design process, changes with the variation of driver body mass. The proposed controller shows its better robustness on driver body mass.

5. Conclusions

In this paper, an integrated seat and suspension has been developed and used for active seat suspension design. Static output feedback controller design method is presented with considering driver body mass variation and the limited capability of actuators. Numerical simulations are used to validate the performance of the designed controllers. The results show that an appropriately designed static output controller can provide better ride comfort performance compared to the passive seat suspension. Its performance is compatible to the state feedback control with a realizable structure. Further study on the optimal configuration of the static output feedback controller considering parameter uncertainties will be investigated.

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