Override and Anti-Windup Control Strategy for Active Vehicle Suspension System

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Abstract

Vehicle suspension plays important roles in vehicle performance. The main function of the vehicle suspension is to improve ride and handling performance. The other important requirements need to be considered in the controller design are suspension def ection and actuator saturation. However, these requirements are conf icting. For example, to obtain better ride comfort it is usually required larger control input and larger suspension def ection, but the actuator that deliver the control signal have a limitation which is commonly known as actuator saturation. There is also a structural constraint that limits the suspension def ection. Most of the vehicle active suspension control strategies deal with actuator saturation and limitation of suspension def ection by keeping the control signals small, until the point where constraint is not met at all. The advantage of this approach is allowing one to use the unconstrained design methods and hence a linear analysis of the problem can be carried out. However, this is achieved at the cost of reducing achievable performance since we expect high performance to be associated with acting on or near constraints. In this study, an alternative approach to the vehicle active suspension system is studied. In this approach, some separation in the controller such that one part is devoted to achieve nominal performance and the other part is devoted to constraint handling is performed. This control strategy involves a two-step design procedure. Namely, a state feedback controller is f rst synthesized for a linear system ignoring saturation and state constraint. Then, the anti-windup and override compensator is designed such that, when saturation occurs or state constraints limits is exceeded, this compensator becomes active to recovers as much as possible the performance lost. Local control design technique based on the circle criterion and $\mathcal{L}_2$ gain performance is used for the anti-windup and override compensator synthesis. A quarter car model is considered in this study and the effectiveness of the proposed approach is shown by a numerical example. The application of the override control for active vehicle suspension system is studied. It is shown that the occurrence of the suspension hits it def ection limits and control input reaches the saturation bound can be minimized by override and anti windup control strategy.

Key Words: Feedback Control, Override Control, Anti-Windup Control, Active Vehicle Suspension, Saturation.

Introduction

More and more systems are involved in today cars to guarantee both safety (e.g. ABS, ESP, etc) and comfort (e.g. suspension control, cabin noise control, etc). Concerning the vehicle comfort and road holding ability, the suspension system has the major roles to provide 1) isolating passengers from vibration and shock arising from road roughness (ride comfort); 2) suppressing the hop of the wheels so as to maintain f rm and uninterrupted contact of wheels to road (good handling or good road holding); and 3) keeping suspension strokes within an allowable maximum (Chen and Guo, 2005). The most important objective for the vehicle suspension system is the improvement of the ride comfort (Sun et al., 2011) and a signif cant control input is often necessary to obtain better performance. However, in practice, the actuators which deliver the control signal are always subject to limits in their magnitude which is commonly known as actuator saturation. In addition, there is also a structural constraint that limit the suspension def ection.

There are several ways of dealing with the control and output constraints. One can use the control and output constraints as optimization constraints such as used in (Chen and Guo, 2005; Sun et al., 2011). This leads to a quite signif cant linear programming problem and a controller can be designed such that it ensures that the state always belongs to the maximum output admissible set. The f nite frequency $\mathcal{H}_\infty$ control has been proposed in (Sun et al., 2011) which shows that an improvement in ride comfort is obtained compared with the entire frequency $\mathcal{H}_\infty$ control. Although the aforementioned controllers have been designed so as to maintain the suspension def ection in a certain range based on a regulated road disturbance, the control input and the suspension def ection might reach the limitation due to various road shapes. One way of incorporating output constraints into controller design is using model predictive control (MPC) strategies. However, the MPC approach is generally expensive in terms of computation (Turner and Postlethwaite, 2002). The other way of dealing with the output constraints is override control (Turner and Postlethwaite, 2002, 2004; Park and Youn, 2003), and in this study, we consider the override control to tackle the problem of the output constraints.

In (Wasiwitono and Saeki, 2011) an alternative approach for the vehicle active suspension system is proposed. A separation in the controller such that one part is devoted to achieve nominal performance and the

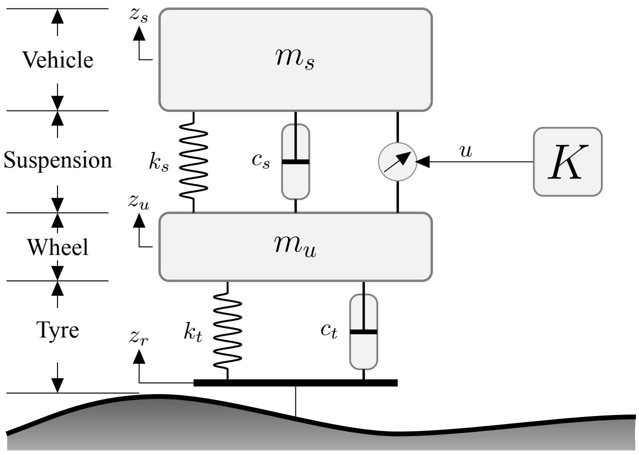

Fig. 1: Quarter car model

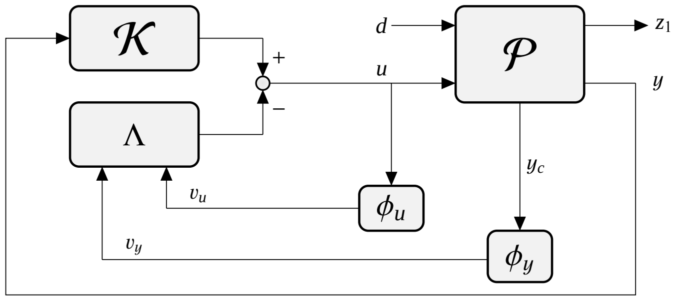

Fig. 2: State feedback control system with deadzone loop and override control strategy

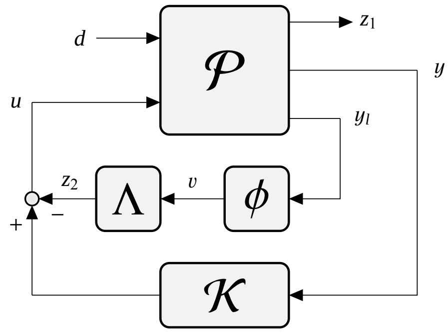

Fig. 3: Override control strategy

other part is devoted to constraint handling is performed. This is the approach taken in anti-windup compensation (Tarbouriech and Turner, 2009). It is shown that the proposed control approach has the potential benef t in achieving the best possible ride comfort. Based on the two-step design procedure, in this study, the combination of override control and anti-windup control strategy are considered to deal with output constraint and control input constraint. Local control design technique based on the circle criterion and $\mathcal{L}_2$ gain performance is used for the anti-windup and override compensator synthesis.

Further, the paper is organized as follows. Section II describes the quarter car system and problem formulation. Section III describes the override and anti-windup compensator synthesis and a numerical example is given in Section IV to show the usefulness of the proposed control strategy.

Problem Formulation

The quarter car model shown in Fig. 1 is considered in this study. The quarter-car model is very often used for the vehicle suspension analysis and design, because of its simplicity yet capture many important characteristics of the full model. In Fig. 1, m_s is the sprung mass, which represents the car chassis; m_u is the unsprung mass, which represents mass of wheel assembly; c_s and k_s are damping and stiffness of the suspension system, respectively; k_t and c_t stand for compressibility and damping of the pneumatic tire, respectively; z_s and z_u are the displacements of the sprung and unsprung masses, respectively; z_r is the road displacement input; and u is the active input of the suspension system.

Def ne the following state variables:

$$x_1 = z_s - z_u, \quad x_2 = z_u - z_r, \quad x_3 = \dot{z}_s, \quad x_4 = \dot{z}_u \qquad (1)$$

where x_1 denotes the suspension def ection, x_2 denotes the tire def ection, x_3 denotes the sprung mass velocity, and x_4 denotes the unsprung mass velocity. Further, def ne the disturbance input as $d = \dot{z}_r$, and $x_p = \left[\ x_1 \quad x_2 \quad x_3 \quad x_4\ \right]^T$, then, by applying Newton's second law of motion and using the static equilibrium position as the origin, the state-space form of the vehicle suspension system can be written as

$$\dot{x}_p \ = \ A_p x_p + B_d d + B_u u \qquad (2)$$

where $x_p \in \mathcal{R}^{n_p}$ is the suspension system state, $d \in \mathcal{R}^{n_d}$ is the disturbance input,$u \in \mathcal{R}^{n_u}$ is the control input, and

$$A_p = \begin{bmatrix} 0 & 0 & 1 & -1 \\ 0 & 0 & 0 & 1 \\ -\frac{k_s}{m_s} & 0 & -\frac{c_s}{m_s} & \frac{c_s}{m_s} \\ \frac{k_s}{m_u} & -\frac{k_t}{m_u} & \frac{c_s}{m_u} & -\frac{c_s+c_t}{m_u} \end{bmatrix}$$
$$B_d = \begin{bmatrix} 0 \\ -1 \\ 0 \\ \frac{c_t}{m_u} \end{bmatrix}, \quad B_u = \begin{bmatrix} 0 \\ 0 \\ \frac{1}{m_s} \\ -\frac{1}{m_u} \end{bmatrix} \qquad (3)$$

In this study the application of the override and anti-windup control strategy for an active vehicle suspension as shown in Fig. 2 is studied. In this Figure y_c is the constrained output, ϕ_u and ϕ_y is deadzone function to control the activation of the compensator Λ. Further, by def ning

$$C_{yl} = \begin{bmatrix} C_{yc} \\ 0 \end{bmatrix}, \quad D_{yu} = \begin{bmatrix} 0 \\ I_{n_u} \end{bmatrix} \qquad (4)$$

the closed-loop system shown in Fig. 2 can be represented as that shown in Fig. 3, with the plant $\mathcal{P}$ is described as

$$\dot{x}_p \ = \ A_p x_p + B_u u + B_d d \qquad (5)$$
$$z_1 \ = \ C_{z1} x_p + D_{zu1} u + D_{zd1} d \qquad (6)$$
$$y_l \ = \ C_{yl} x_p + D_{yu} u \qquad (7)$$

Assume that the linear control $\mathcal{K}$ has been designed, let now focused on the controlled output for the design of the override and anti-windup compensator Λ. The most important objective of the active vehicle suspension is the improvement of the ride comfort, hence, the sprung mass acceleration $\ddot{z}_s$ is choosen as performance output and we have

$$C_{z1} \ = \ \begin{bmatrix} -\frac{k_s}{m_s} & 0 & -\frac{c_s}{m_s} & \frac{c_s}{m_s} \end{bmatrix} \qquad (8)$$
$$D_{zu1} \ = \ \begin{bmatrix} \frac{1}{m_s} \end{bmatrix}, \quad D_{zd1} = \begin{bmatrix} 0 \end{bmatrix} \qquad (9)$$

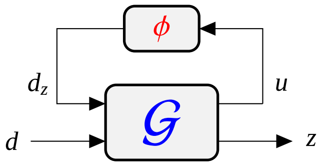

Fig. 4: Nonlinear feedback system with sector condition

Furthermore, because the control objective of the override and anti-windup control is to avoid the suspension reach its stroke limitation and preventing the actuator from saturation, beside the aforementioned performance objective, we want to minimize

$$z_2 = \Lambda v \tag{10}$$

therefore, we will consider the next z

$$z = \begin{bmatrix} W_1 z_1 \\ W_2 z_2 \end{bmatrix} \tag{11}$$

The constant weights W_1 and W_2 are used for the tuning of the balance between the ride comfort requirements and the minimization of z_2. Thus the problem is to minimize γ subject to

$$||z||_2 < \gamma ||d||_2 \tag{12}$$

where z is defned by (11) with internal stability.

Override and Anti-Windup Compensator Synthesis

A state feedback control system is considered is considerd in this study and therefore, the control signal u can be described as

$$u = Kx_p - \Lambda v \tag{13}$$

Defne

$$C_z = \begin{bmatrix} W_1 C_{z1} \\ 0 \end{bmatrix}, \quad D_{zu} = \begin{bmatrix} W_1 D_{zu1} \\ 0 \end{bmatrix}$$
$$D_{zd} = \begin{bmatrix} W_1 D_{zd1} \\ 0 \end{bmatrix}, \quad D_{zv} = \begin{bmatrix} 0 \\ W_2 I_{n_u} \end{bmatrix} \tag{14}$$

then, the system in Fig. 3 can be represented as that of Fig. 4, with the system $\mathcal{G}$ is described by

$$\begin{aligned} \dot{x}_p &= \left(A_p + B_u K\right) x_p - B_u \Lambda v + B_d d & (15) \\ z &= (C_z + D_{zu} K)\, x_p - (D_{zu} - D_{zv})\, \Lambda v + D_{zd} d & (16) \\ y_l &= \left(C_{yl} + D_{yu} K\right) x_p - D_{yu} \Lambda v & (17) \end{aligned}$$

Further, by considering that $\phi(\cdot)$ satisf es the sector condition (Khalil, 1996) in the f nite interval $[-\Xi, \Xi]$ with

$$\Xi = (1/(1-\kappa))\, y_l$$

then the following inequality condition holds

$$v^T X (v - \kappa y_l) \le 0 \tag{18}$$

and the next theorem guarantees the $\mathcal{L}_2$ gain condition (12) for the nonlinearity that satisf es (18).

Table 1: Quarter-car model parameters

Notation	Descreption	Value
m_s	Mass of car chassis	320 kg
m_u	Mass of wheel assembly	40 kg
k_s	Suspension stiffness	18 kN/m
k_t	Tire stiffness	200 kN/m
c_s	Suspension damping	1 kNs/m
c_t	Tire damping	10 Ns/m
z_{max}	Max. suspension def ection	0.1 m
u_{max}	Saturation bound	1.5 kN

Theorem 1. *For a given κ, if there exist a positive-definite symmetric matric $Q \in \mathcal{R}^{n_p \times n_p}$, a diagonal matrix $T = diag\left[t_1, t_2, \cdots, t_{n_u+n_{yc}}\right] > 0$, and a scalar $\gamma > 0$ that satisfies the next matrix inequality*

$$\begin{bmatrix} \mathbf{Q}A_p^T + A_p\mathbf{Q} + \mathbf{Q}K^T B_u^T + B_u K\mathbf{Q} & -B_u\mathbf{N} + \mathbf{Q}C_{yl}^T\kappa + \mathbf{Q}K^T D_{yu}^T\kappa & B_d & \mathbf{Q}C_z^T + \mathbf{Q}K^T D_{zu}^T \\ -\mathbf{N^T} B_u^T + \kappa C_{yl}\mathbf{Q} + \kappa D_{yu} K\mathbf{Q} & -2\mathbf{T} - \mathbf{N^T} D_{yu}^T\kappa - \kappa D_{yu}\mathbf{N} & 0 & \mathbf{N^T} D_{za}^T \\ B_d^T & 0 & -\gamma I & D_{zd}^T \\ C_z\mathbf{Q} + D_{zu}K\mathbf{Q} & D_{za}\mathbf{N} & D_{zd} & -\gamma I \end{bmatrix} < 0 \tag{19}$$

then, the feedback system shown in Fig. 4 with override compensator $\Lambda = NT^{-1}$ is asymptotically stable and the $\mathcal{L}_2$ gain from d to z is less than γ when the condition $|y_l| \leqslant (1/(1-\kappa_i))\, y_{l(i)}(sat)$, $i = 1, 2, \ldots, n_v$ holds.

Proof. See Appendix A □

This design problem is an LMI problem with respect to the variables Q, N, γ, T, and therefore it can be solved easily by a numerical optimization method.

Numerical Example

The quarter-car model parameters are listed in Table 1. We assume that the linear controller $\mathcal{K}$ has been designed, and for the current study we consider the state feedback controller $\mathcal{K}$ reported in (Sun et al., 2011) that use less restrictive constraints compared with that reported in (Chen and Guo, 2005). The corresponding state feedback controller is given by

$$K = 10^4\, [0.5033\ -1.3155\ -0.5329\ -0.0547] \tag{20}$$

Figure 5 shows the frequency response from the disturbance to the body acceleration and suspension def ection for the case of passive suspension and active suspension with state feedback control reported in (Chen and Guo, 2005; Sun et al., 2011). It can be clearly seen that the state feedback controller reported in (Sun et al., 2011) yields the least value of $\mathcal{H}_\infty$ norm over the frequency range 1 ∼ 8 Hz compared with the passive system and the state feedback controller reported in (Chen and Guo, 2005). However, in term of suspension def ection the state feedback controller reported in (Sun et al., 2011) yields larger value especially at lower frequencies.

Let us now consider the case of an isolated bump in a road surface. The corresponding disturbance input is given by (Sun et al., 2011)

$$d = \begin{cases} \frac{A_m}{2}\left(1 - cos(2\pi f t)\right) & if\ 0 \le t \le \frac{1}{f} \\ 0 & if\ \frac{1}{f} < t \end{cases} \tag{21}$$

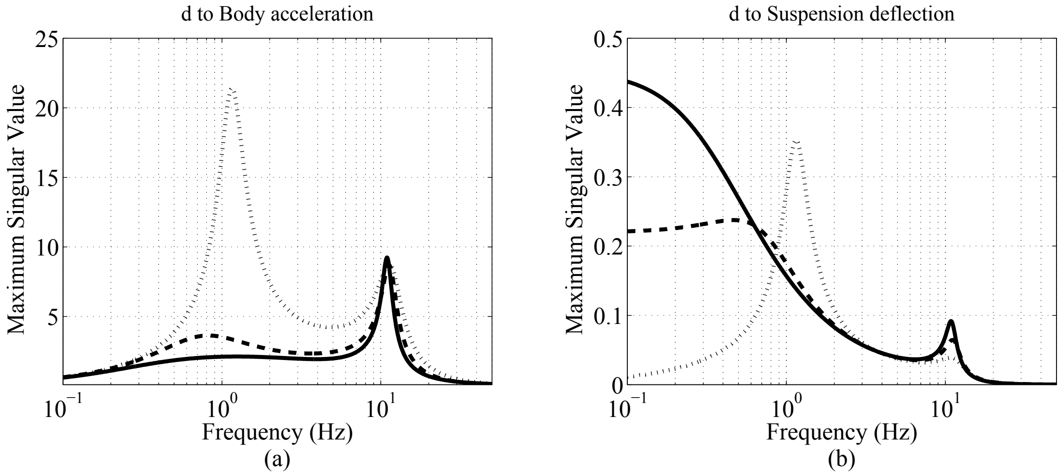

Fig. 5: Frequency response from disturbance to body acceleration and suspension def ection (dotted line: passive suspension; dashed line: state feedback control (Chen and Guo, 2005); solid line: state feedback control (Sun et al., 2011))

where A_m represents the amplitude of the bump and f is disturbance frequency. Fig. 6 shows the plots of body acceleration, suspension def ection and control input for the case of disturbance frequency 1 Hz and bump amplitude 0.1 m. It can be seen that the ride comfort by state feedback control (Sun et al., 2011) is better compared with that by state feedback control (Chen and Guo, 2005). However, the improvement in the ride comfort requires a larger actuator force and larger suspension def ection. Fig. 7 shows the plots of body acceleration, suspension def ection and control input for the case of disturbance frequency 1 Hz and bump amplitude 0.16 m. It can be seen in this Figure that the ride comfort by state feedback control (Sun et al., 2011) is deteriorated caused by the suspension hits it limits, at around 0.5 s.

To overcome this problem, we add override and anti-windup compensator and based on Theorem 1, we synthesize the compensator Λ by setting $\kappa = 1$, $W_1 = 250$ and $W_2 = 50$, and obtain the following result

$$\Lambda = \begin{bmatrix} 2.55657 & -18.72049 \end{bmatrix} \tag{22}$$

Since the control input saturation is ±1.5 kN and suspension def ection limit is ±0.1 m, we choose the deadzone limit $\phi_u = \pm 1$ kN and $\phi_y = \pm 0.05$ m, respectively. Fig. 8 shows the plots of body acceleration, suspension def ection and control input for the case of disturbance frequency 1 Hz and bump amplitude 0.16 m with proposed control strategy. It can be seen that the improvement in ride comfort is obtained.The suspension def ection and control input is reduced to be bellow the allowable bound.

Further, to compare effectiveness of the proposed control strategy with standard state feedback control (Chen and Guo, 2005; Sun et al., 2011) at different disturbance frequencies, Fig. 9 shows the plots of body acceleration, suspension def ection and control input for the case of disturbance frequency 4 Hz with bump amplitude 0.11 m. It can be seen from these Figure that both the state feedback control (Chen and Guo, 2005; Sun et al., 2011) hit the suspension def ection limit at around 0.1 s. Whilst, the active suspension system with proposed control strategy does not.

Conclusion

The application of the override and anti-windup control for active vehicle suspension system is studied. It is shown that the occurrence of the suspension hits it def ection limits and control input reaches the saturation bound can be minimized by the proposed control strategy. Furthermore, by setting $M = KQ$ in (19) it is possible to synthesize the state feedback controller K and compensator Λ simultaneously. It is our future work to show the usefulness of such simultaneous design algorithm.

A Proof of Theorem 1

Consider a quadratic Lyapunov function

$$V\left(x_p\right) = x_p^T P x_p, \quad P = P^T > 0, \quad P \in \mathbb{R}^{n_p \times n_p} \tag{23}$$

In order to show that the closed loop system is asymptotically stable and the $\mathcal{L}_2$ gain from d to z is less than γ, we may show that the Lyapunov function (23) satisf es the next dissipation inequality

$$\frac{dV}{dt} < \gamma d^T d - \frac{1}{\gamma} z^T z \tag{24}$$

By using the sector condition (18) and the S-procedure, we obtain

$$\frac{dV}{dt} + \frac{1}{\gamma} z^T z - \gamma d^T d - 2 v^T X \left(v - \kappa y_l\right) < 0 \tag{25}$$

Def ne $\xi = \begin{bmatrix} x_p^T & v^T & d^T \end{bmatrix}^T$, (25) can be written in the form

$$\xi^T \begin{bmatrix} \left(A_p^T + K^T B_u^T\right) P + P\left(A_p + B_u K\right) + \frac{1}{\gamma}\left(C_z^T + K^T D_{zu}^T\right)\left(C_z + D_{zu} K\right) & -P B_u \Lambda + \left(C_{yl}^T + K^T D_{yu}^T\right)\kappa X + \frac{1}{\gamma}\left(C_z^T + K^T D_{zu}^T\right) D_{za} \Lambda & P B_d + \frac{1}{\gamma}\left(C_z^T + K^T D_{zu}^T\right) D_{zd} \\ -\Lambda^T B_u^T P + X\kappa\left(C_{yl} + D_{yu} K\right) + \frac{1}{\gamma}\Lambda^T D_{za}^T \left(C_z + D_{zu} K\right) & -2X - \Lambda^T D_{yu}^T \kappa X - X\kappa D_{yu} \Lambda + \frac{1}{\gamma}\Lambda^T D_{za}^T D_{za} \Lambda & \frac{1}{\gamma}\Lambda^T D_{za}^T D_{zd} \\ B_d^T P + \frac{1}{\gamma} D_{zd}^T \left(C_z + D_{zu} K\right) & \frac{1}{\gamma} D_{zd}^T D_{za} \Lambda & -\gamma I + \frac{1}{\gamma} D_{zd}^T D_{zd} \end{bmatrix} \xi < 0 \tag{26}$$

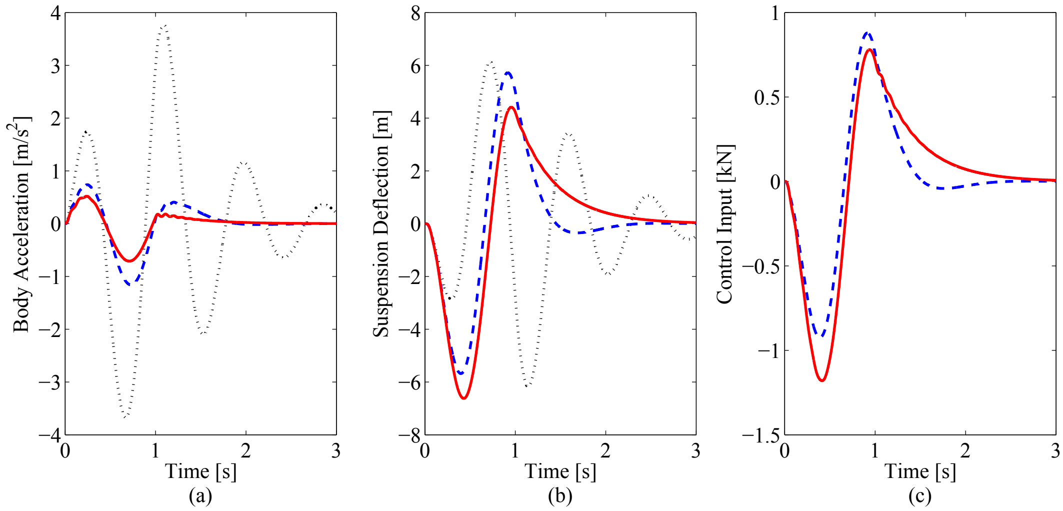

Fig. 6: Bump response for the case of disturbance frequency 1 Hz and amplitude 0.1 m (dotted line: passive suspension; dashed line: state feedback control (Chen and Guo, 2005); solid line: state feedback control (Sun et al., 2011))

Since (26) must hold for all x_p, v, and d, the matrix $P > 0$ must satisfy the next constraint

$$\begin{bmatrix} (A_p^T + K^T B_u^T)P + P(A_p + B_u K) + \frac{1}{\gamma}(C_z^T + K^T D_{zu}^T)(C_z + D_{zu}K) & -PB_u\Lambda + (C_{yl}^T + K^T D_{yu}^T)\kappa X + \frac{1}{\gamma}(C_z^T + K^T D_{zu}^T)D_{za}\Lambda & PB_d + \frac{1}{\gamma}(C_z^T + K^T D_{zu}^T)D_{zd} \\ -\Lambda^T B_u^T P + X\kappa(C_{yl} + D_{yu}K) + \frac{1}{\gamma}\Lambda^T D_{za}^T(C_z + D_{zu}K) & -2X - \Lambda^T D_{yu}^T \kappa X - X\kappa D_{yu}\Lambda + \frac{1}{\gamma}\Lambda^T D_{za}^T D_{za}\Lambda & \frac{1}{\gamma}\Lambda^T D_{za}^T D_{zd} \\ B_d^T P + \frac{1}{\gamma} D_{zd}^T (C_z + D_{zu}K) & \frac{1}{\gamma} D_{zd}^T D_{za}\Lambda & -\gamma I + \frac{1}{\gamma} D_{zd}^T D_{zd} \end{bmatrix} < 0 \quad (27)$$

Applying the Schur complement to (27), we obtain

$$\begin{bmatrix} A_p^T P + PA_p + K^T B_u^T P + PB_u K & -PB_u\Lambda + C_{yl}^T \kappa X + K^T D_{yu}^T \kappa X & PB_d & C_z^T + K^T D_{zu}^T \\ -\Lambda^T B_u^T P + X\kappa C_{yl} + X\kappa D_{yu}K & -2X - \Lambda^T D_{yu}^T \kappa X - X\kappa D_{yu}\Lambda & 0 & \Lambda^T D_{za}^T \\ B_d^T P & 0 & -\gamma I & D_{zd}^T \\ C_z + D_{zu}K & D_{za}\Lambda & D_{zd} & -\gamma I \end{bmatrix} < 0 \quad (28)$$

Applying a simple congruence transformation block-diag$\left(P^{-1}, X^{-1}, I, I\right)$ to (28) and define $Q = P^{-1}$, $T = X^{-1}$, we have

$$\begin{bmatrix} QA_p^T + A_p Q + QK^T B_u^T + B_u KQ & -B_u\Lambda T + QC_{yl}^T \kappa + QK^T D_{yu}^T \kappa & B_d & QC_z^T + QK^T D_{zu}^T \\ -T\Lambda^T B_u^T + \kappa C_{yl}Q + \kappa D_{yu}KQ & -2T - T\Lambda^T D_{yu}^T \kappa - \kappa D_{yu}\Lambda T & 0 & T\Lambda^T D_{za}^T \\ B_d^T & 0 & -\gamma I & D_{zd}^T \\ C_z Q + D_{zu}KQ & D_{za}\Lambda T & D_{zd} & -\gamma I \end{bmatrix} < 0 \quad (29)$$

Define $N = \Lambda T$ then (29) can be written as (19)

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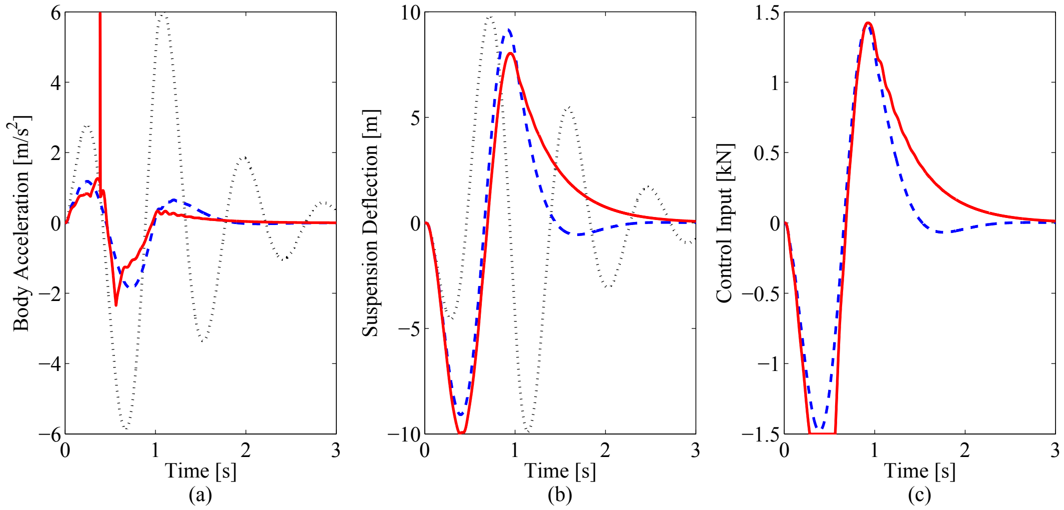

Fig. 7: Bump response for the case of disturbance frequency 1 Hz and amplitude 0.16 m (dotted line: passive suspension; dashed line: state feedback control (Chen and Guo, 2005); solid line: state feedback control (Sun et al., 2011))

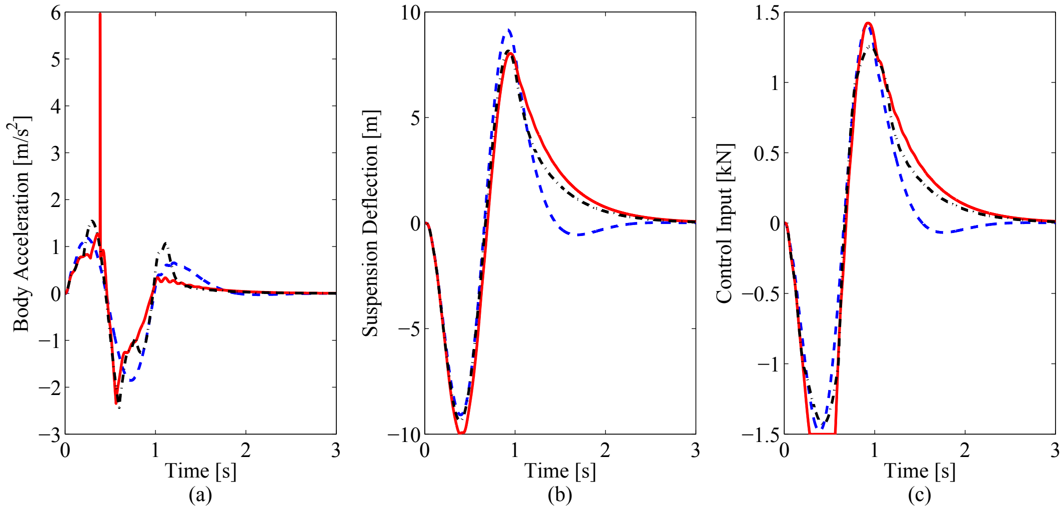

Fig. 8: Bump response for the case of disturbance frequency 1 Hz and amplitude 0.16 m (dashed line: state feedback control (Chen and Guo, 2005); solid line: state feedback control (Sun et al., 2011)); dash-dotted line: proposed control strategy)

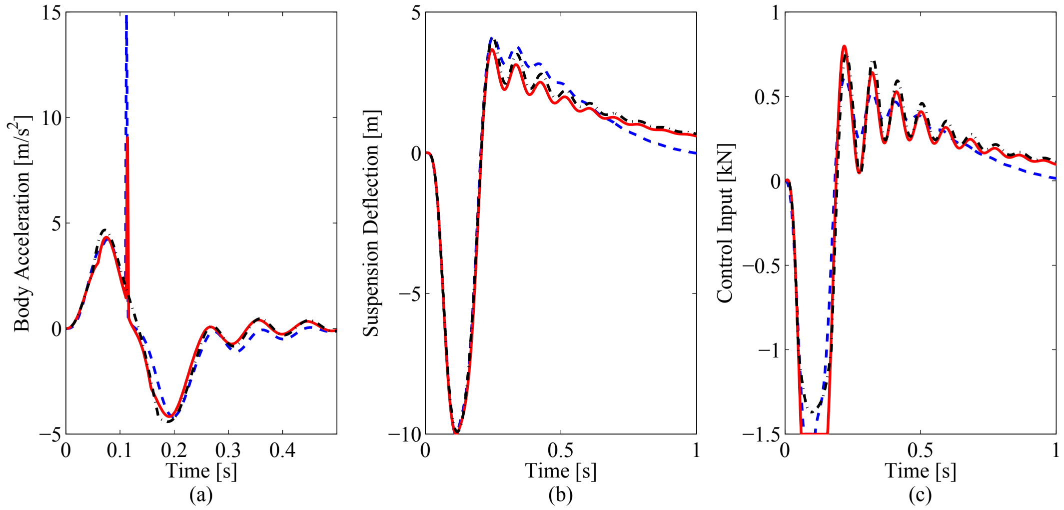

Fig. 9: Bump response for the case of disturbance frequency 4 Hz and amplitude 0.11 m (dashed line: state feedback control (Chen and Guo, 2005); solid line: state feedback control (Sun et al., 2011)); dash-dotted line: proposed control strategy