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Optimization of the exponential stabilization problem in active suspension system using PSO

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ABSTRACT

This paper proposes an active suspension mechanism for three-degree-of-freedom (3-DOF) twin-shaft vehicles of front axle suspension with bounded uncertainties using exponential decay control and particle swarm optimization (PSO) techniques. A new exponential stabilization criterion of the system via the dynamic state feedback control is derived and the optimization problem of exponential stabilization is discussed. The optimization problem is solved by using PSO method to guarantee all the states of vehicles in an optimal exponential decay in nearly real-time. Simulation results in both frequency and time domains show that the vibration characteristics of vehicles for the proposed active suspension system achieve significant improvements over the passive and linear quadratic (LQ) active suspension systems.

1. Introduction

The vibration of vehicle leads to fatigue of driver and decreases drive safety and operation stability of vehicle. Hence, developing improved suspension systems to achieve high ride quality is one of the important challenges in automotive industry. Passive suspension system built of springs and dampers with serious limitations only provide fixed rates for spring and damping, which cannot efficiently compromise between ride comfort and handling performance. In addition, there is no way to eliminate resonance vibration around natural frequencies, which is the result of vehicle body dynamics (Sakman, Guclu, & Yagiz, 2005). Therefore, the goals of vehicle suspension systems are to decrease the acceleration of vehicle body, determining ride comfort, as well as to provide adequate suspension deflection, determining handling performance, within an allowable maximum (Hrovat, 1997). In reality, some of vehicle parameters are with uncertainties, so that it is an important issue to deal with vehicle vibration subject to uncertain parameters in engineering applications. In the literature, many approaches (Cao, Liu, Li, & Brown, 2008; Fialho & Balas, 2002; Gao, Zhang, & Du, 2007; Hrovat, 1997; MacLeod & Goodall, 1996; Rajamani & Hedrick, 1995; Sunwoo, Cheok, & Huang, 1991; Yagiz, Hacioglu, & Taskin, 2008) based on adaptive control, linear quadratic (LQ) optimal control, fuzzy control and sliding mode control for the active suspension systems have been widely studied over the past decades.

Since the suspension systems are needed to reduce the vibrations of vehicles in a short period of time and PSO performs fast convergence to the optimal or near-optimal solutions over a small number of iterations which is widely used to solve varieties of optimization problems (Abido, 2002; Parsopoulos & Vrhatis, 2004; Ourique, Biscaia, & Pinto, 2002), PSO is employed to solve the optimization problem of suspension systems. In this paper, an exponential decay control, namely active suspension system, of a 3-DOF twin-shaft vehicle with bounded uncertainties is presented. The contributions of this paper are summarized as follows. First, a new control strategy to stabilize the vibrations of a vehicle in an exponential decay is proposed. Second, the proposed condition with a suitable dynamic feedback control law combining the PSO algorithm can yield an optimal exponential decay controller for the automotive application. Third, the exponential stabilization in active suspension system is detailedly discussed. In the experiments, the characteristics among all states of the vertical acceleration of vehicle body, the pitch vibration angle acceleration of vehicle body and the dynamic deflection of front axle suspension for active suspension system are analyzed and compared with the passive as well as linear quadratic (LQ) active suspensions. The results show that the proposed active suspension system is able to minimize the vertical and pitch vibration angle accelerations of vehicle body to significantly improve ride comfort. It is also seen that all magnitudes of states are rapidly and exponentially decaying during the settling time, which indicates that the stabilization can be achieved in a very short period of time.

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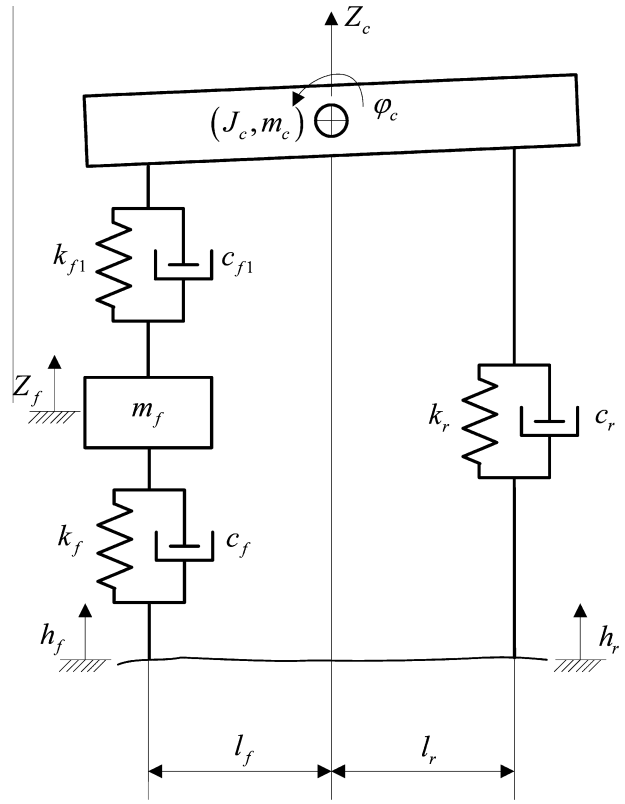

Fig. 1. 3-DOF model of vehicle with front axle suspension.

The paper is organized as follows. A plant model of a 3-DOF twin-shaft tractor with the front axle suspension is introduced in Section 2. Section 3 formulates an exponential stabilization criterion of active suspension system by optimizing decay rate. The procedure of PSO controller algorithm in solving the optimization problem is discussed in Section 4. Then Section 5 presents the simulation results and discussions both in the frequency and time domains. Finally, conclusions are given in Section 6.

2. Plant model of a tractor with the front axle suspension

Consider a passive suspension system for a 3-DOF twin-shaft vehicle, such as road tractors, engineering tractors and agricultural tractors, with front axle suspension modeling as shown in Fig. 1, the equations of motion for the vehicle body and the wheels are given as follows (Lu, Zhu, & Zhang, 2007):

$$\ddot{m}_c\ddot{Z}_c = c_{f1}(\dot{Z}_f - \dot{Z}_c) + k_{f1}(Z_f - Z_c) + c_r(\dot{h}_r - \dot{Z}_c) + k_r(h_r - Z_c) - (k_r\phi_c l_r + c_r\dot{\phi}_c l_r) + (k_{f1}\phi_c l_f + c_{f1}\dot{\phi}_c l_f), \tag{1a}$$

$$J_c\ddot{\phi}_c = l_r[c_r(\dot{h}_r - \dot{Z}_c) + k_r(h_r - Z_c)] - l_f[c_{f1}(\dot{Z}_f - \dot{Z}_c) + k_{f1}(Z_f - Z_c)] - l_r(k_r\phi_c l_r + c_r\dot{\phi}_c l_r) - l_f(k_{f1}\phi_c l_f + c_{f1}\dot{\phi}_c l_f), \tag{1b}$$

$$m_f\ddot{Z}_f = k_f(h_f - Z_f) + c_f(\dot{h}_f - \dot{Z}_f) - c_{f1}(\dot{Z}_f - \dot{Z}_c) - k_{f1}(Z_f - Z_c) - (k_{f1}\phi_c l_f + c_{f1}\dot{\phi}_c l_f), \tag{1c}$$

where m_c is the whole mass of vehicle, J_c is the inertia of vehicle body, $\ddot{Z}_c$ is the vertical acceleration of vehicle body, $\ddot{\phi}_c$ is the pitch vibration angle acceleration of vehicle body, m_f is the unsprung mass of front axle suspension, Z_f is the dynamic deflection of front axle suspension, k_{f1} is the stiffness of front axle suspension system, c_{f1} is the damping of front axle suspension system, k_f is the front tire stiffness, c_f is the front tire damping, k_r is the rear tire stiffness, c_r is the rear tire damping, l_f is the horizontal distance of front shaft to vehicle center of mass, l_r is the horizontal distance of rear shaft to vehicle center of mass, h_f is the road disturbance input of front wheel, h_r is the road disturbance input of rear wheel.

Let $x = [Z_c \quad \phi_c \quad Z_f]^T$, the dynamic model of the tractor with the front axle suspension can be expressed in a form of second-order vector differential equation as

$$\begin{bmatrix} m_c & 0 & 0 \\ 0 & J_c & 0 \\ 0 & 0 & m_f \end{bmatrix}\ddot{x} + \begin{bmatrix} c_{f1}+c_r & c_r l_r - c_{f1} l_f & -c_{f1} \\ c_r l_r - c_{f1} l & c_{f1} l_f^2 + c_r l_r^2 & c_{f1} l_f \\ -c_{f1} & c_{f1} l_f & c_{f1}+c_f \end{bmatrix}\dot{x} + \begin{bmatrix} k_{f1}+k_r & k_r l_r - k_{f1} l_f & -k_{f1} \\ k_r l_r - k_{f1} l & k_{f1} l_f^2 + k_r l_r^2 & k_{f1} l_f \\ -k_{f1} & k_{f1} l_f & k_{f1}+k_f \end{bmatrix} x = \begin{bmatrix} k_r h_r + c_r\dot{h}_r \\ k_r h_r l_r + c_r\dot{h}_r l_r \\ k_f h_f + c_f\dot{h}_f \end{bmatrix}. \tag{2}$$

The road roughness characterized by consistent excitations is typically specified as a random process of a ground displacement power spectral density and the road disturbance velocity inputs $\dot{h}_f = [\dot{h}_f \quad \dot{h}_r]^T$ are independent of frequency. Then the equation of road disturbance velocity inputs is (Sun & Chen, 2003)

$$\dot{h}(t) = 2\pi\sqrt{G_0 V}\omega(t), \tag{3}$$

where G_0 stands for the road roughness coefficient, V is the vehicle forward velocity, and, $\omega = [\omega_f \quad \omega_r]^T$, ω_f and ω_r are the road inputs from front and rear wheels, respectively.

3. Derivation of the optimization for exponential stabilization

In order to stabilize the vibration characteristic of the 3-DOF twin-shaft tractor described as (1) in an exponential decay, the system represented in (4) with a dynamic state feedback control to be exponentially stable is developed, which is extended from the authors' previous work (Tung, Juang, Wu, & Shieh, in press) for the stabilization of the system without bounded uncertainties. Consider the system described by the following second-order vector differential equation

$$N(t)\ddot{x}(t) + A(t)\dot{x}(t) + B(t)x(t) + f(x(t),t) = C(t)u(t), \quad x(t_0) = x_0,\ t \geqslant t_0 \tag{4}$$

with uncertainties bounded by

$$\|f(x(t),t)\| \leqslant \alpha\|x(t)\| \tag{5}$$

and the dynamic state feedback control

$$u(t) = E(t)\dot{x}(t) + F(t)x(t), \tag{6}$$

where the initial condition x_0 and $x(t)$ are the n-dimensional state vectors, $||x(t)||$ denotes the vector norm of a vector $x(t)$, $f(x(t), t)$ represents the n-dimensional vector non-linear uncertainties, α is known as the Lipschitz constant at zero (Zevin & Pinsky, 2003), $u(t)$ is the n'-dimensional control vector, $N(t) \in R^{n\times n}$ is an invertible matrix, $A(t), B(t) \in R^{n\times n}$ and $C(t) \in R^{n\times n'}$ are continuous matrices, and $E(t)$, $F(t) \in R^{n'\times n}$ are controller matrices. Then the closed-loop system is

$$N(t)\ddot{x}(t) + \overline{A}(t)\dot{x}(t) + \overline{B}(t)x(t) + f(x(t),t) = 0, \tag{7}$$

where

$$\overline{A}(t) = A(t) - C(t)E(t), \quad \overline{B}(t) = B(t) - C(t)F(t). \tag{8}$$

Let

$$q(t) = e^{mt}x(t) \tag{9}$$

and