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Robust fault-tolerant H_∞ control of active suspension systems with finite-frequency constraint

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ABSTRACT

In this paper, the robust fault-tolerant (FT) H_∞ control problem of active suspension systems with finite-frequency constraint is investigated. A full-car model is employed in the controller design such that the heave, pitch and roll motions can be simultaneously controlled. Both the actuator faults and external disturbances are considered in the controller synthesis. As the human body is more sensitive to the vertical vibration in 4–8 Hz, robust H_∞ control with this finite-frequency constraint is designed. Other performances such as suspension deflection and actuator saturation are also considered. As some of the states such as the sprung mass pitch and roll angles are hard to measure, a robust H_∞ dynamic output-feedback controller with fault tolerant ability is proposed. Simulation results show the performance of the proposed controller.

1. Introduction

Vehicle suspension plays a critical role in improving car's ride comfort and handling ability. Active suspension can potentially improve the driving maneuverability and passenger's comfort simultaneously, thanks to the advanced control algorithm and improved actuators. Numerous control strategies for active suspension control have been designed to improve the ride quality and vehicle handling performance based on quarter-car suspension model or the half-car model. For example, an adaptive backstepping control for active suspension systems with hard constraints is discussed in [1]. A linear quadratic Gaussian control based on quarter-car suspension model is presented in [2]. A linear parameter-varying (LPV) gain-scheduling controller design for a non-linear active suspension system is given in [3]. To further analyze heave and pitch motions of the vehicle, a robust H_∞ controller with input delay is given in [4], a saturated adaptive robust control for active suspension systems in [5], and a constrained H_∞ control of active suspension with time delay is presented in [6], both of which a half-car model is employed.

Whenever a car is cruising on roads or in an emergency, i.e. significant rolling or pitching dynamics are generated, the performance demands significantly both of passenger comfort, suspension deflections and tire deflections simultaneously. In these cases the controllers not only focus on one of the performance specifications, but also focus on all the above performances [7]. Consequently, for overall consideration, a full-car model is the most practical model for analyzing the suspension performance. In [8] an adaptive robust control method for a full-car model with electrohydraulic actuators is proposed to improve the ride comfort. A full vertical car observer design methodology is given for active suspension control

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applications in [9]. In [10,11], a novel motion-mode energy detection method is presented for vehicle suspension control, by which the power consumption can be possibly reduced. In addition, integrated vehicle ride and steady-state handling control is proposed to improving ride comfort and handling performance in [12].

However, the researches discussed above mainly investigated the state-feedback based control schemes for active suspensions with the assumption that all the states can be measured. The fact is, when some of the required state variables of the suspension systems are not measurable, the aforementioned methods are not feasible. Consequently, the output-feedback based control schemes have been developed, see for instance [13–16]. For active suspension systems, there exist some results on controller design for the suspension systems by using output-feedback control approach e.g. [17–19]. However, there are no evidence that full-car model was used in the above active suspension systems. In addition, as the performance of active suspension heavily depends on the proper operations on the actuator, the proposed control method should be robust to the actuator faults. A fault-tolerant control for electric ground vehicles with independently actuated in-wheel motors is presented in [20]. A data-driven fault diagnosis design for active suspension using clustering method is proposed in [21]. A fault-tolerant H_∞ control design for active suspension vehicle systems with actuator faults is given in [22], and a quarter car model is employed in the research. Nevertheless, few of reported fault-tolerant control designs were developed for the full-car model or with output-feedback control methodologies.

For vehicle passengers, another critical problem is that the human body is much more sensitive to the vibration between 4 and 8 Hz in vertical direction [23]. This should not be neglected in controller design of active suspension systems. In this paper, the output-feedback control problem of active suspension systems with finite-frequency constraint is investigated. The main contributions of this paper lie in the following aspects. First, in order to control the suspension in heave, pitch and roll motions, a full-car model is employed as the object of research. Second, as the fact that human body is more sensitive to vibrations between 4 and 8 Hz in vertical direction, a finite-frequency constraint is considered in the controller design. The generalized Kalman–Yakubovich–Popov (KYP) lemma [24] is adopted such that the targeted disturbance attenuation in the specific frequency range can be achieved. Third, as the measurement of some states is impractical or too expensive, a dynamic output-feedback controller is designed. In addition, some undesirable performances such as actuator faults, suspension deflection and actuator saturation are all considered in the controller design. The rest of the paper is organized as follows. The full-car model and the control objective is briefly described in Section 2. The proposed robust fault-tolerant controller with finite-frequency constraint is designed in Section 3. Simulation results and conclusive remarks are presented in Sections 4 and 5, respectively.

2. System model and problem formulation

A full-car model of active suspension is shown in Fig. 1. In Fig. 1, m_s is the sprung mass of vehicle body, m_{ui} denote the vehicle unsprung mass, respectively; x_{gi} are the road input to the four wheels, x_{ui} are the displacement of unsprung mass, x_{si} are the displacement of sprung mass, k_i and c_i are the stiffness and damping of active suspension, respectively; f_i are the control efforts of the actuators in the active suspension, k_{ti} denote the stiffness of tire. Note that as the damping of tire is small, the damping is omitted in the model.

Based on Newton's Second Law, and the assumption that the pitch and roll angles are small, the motion equations of the sprung mass can be modelled as

$$m_s\ddot{x}_s = F_1 + F_2 + F_3 + F_4$$
$$I_p\ddot{\theta} = -a(F_1+F_2)+b(F_3+F_4)$$

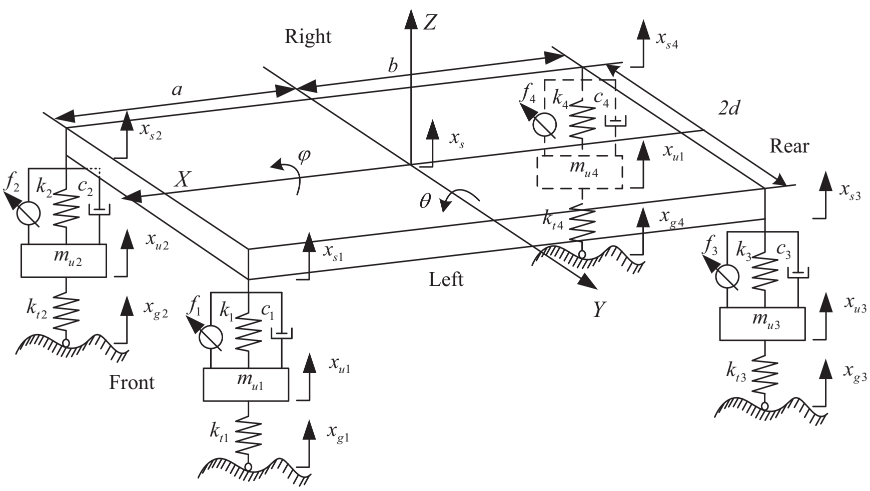

Fig. 1. Full car active suspension model.