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Research paper

Model-free fractional-order sliding mode control for an active vehicle suspension system

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ABSTRACT

This paper presents a new model-free fractional-order sliding mode control (MFFOSMC) based on an Extended State Observer (ESO) for a quarter car active suspension systems. The main goal is to increase the ride comfort while the dynamic wheel load and the suspension deflection remain within safety critical bounds. The model with nonlinearity, parameter variation and/or external disturbance which includes the friction force effect are simultaneously considered to provide a realistic framework. Moreover, modeling was performed using the software LMS AMESim, while the control part was configured on Matlab/Simulink. Lyapunov stability theory is used to analyze the closed-loop system and finite-time convergence stability. Finally, to demonstrate the effectiveness of the proposed controller, a comparison with classical PID, time delay estimation control, and intelligent PID controller has been performed.

1. Introduction

The main tasks of the suspension systems are to improve the three conflicting indices which are ride comfort, suspension deflection, and road holding, (i.e., to improve the ride comfort, often, the suspension deflection increased, and the road holding decreased [1]). Moreover, the active suspension systems can adjust the system energy to control the vibration of the vehicle body, keep the tire road holding, decrease the influences of road disturbances and improper operations in braking and steering. Thus, ride comfort and ride safety can be improved. In recent years, active suspension control technologies have become a hot research topic. Moreover, the suspension systems have significant influence of the subjective impression of the vehicle [2,3].

The active suspension systems are widely used to improve the performance of the suspension system over the conventional passive and semi-active suspension systems [4]. To improve ride comfort and ride safety, great deals of attention of many researchers and considerable efforts are devoted to study and design the various types of control strategies and physical modeling such as linear or nonlinear active suspension systems. Linear quadratic Gaussian control [5,6] and the adaptive backstepping control scheme is used for the uncertain mathematical model of suspension systems with nonlinear spring and piecewise linear damper [3]. In contrast, for nonlinear active suspension systems, many control methods have been proposed to improve the performance [7–10].

The sliding mode control has been widely used to control the active suspension system, because of its robustness and disturbance reduction of a control system regardless of the nonlinearity and parameters variation and/or external disturbance. The feedback linearization method has been employed as one of the main techniques [11,12]. As a result in [13], the control method which depending on the linearized active suspension system cannot provide accurate optimal results.

The terminal sliding mode (TSM) control scheme was proposed to resolve the chattering and singularity problems [14]. However, the nonlinear terms for the suspension system are not analyzed in the above mentioned paper. Another type of controller which is using the fuzzy sliding mode control for active suspension systems are introduced in [15]. unfortunately, the effect of actuator dynamics are not analyzed in this model. Thus, it is necessary to design a new controller for nonlinear active suspension model with uncertainties and actuator dynamics.

In order to improve the performance tracking of the active suspension systems, a novel model free controller depending on an intelligent proportional-integral-derivative (iPID) control has been proposed in [16]. It is proved to control a variety of systems such as a quadrotor vehicle in [17] and to ensure the robustness and the stability of the control synthesis [18,19]. This approach needs to estimate the unknown dynamics, therefore, an observer based on time delay estimation techniques is used [20]. Nevertheless, this observer based model-free control cannot ensure the trajectory tracking error tends to zero rapidly. Besides, the measurement noise degrades the performance of closed loop control system significantly [21].

To solve this problem, an extended state observer (ESO) is presented

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to estimate the unknown dynamics of the active suspension system. The proper performance can be ensured when the unknown dynamics are bounded, and correctly selected of the ESO parameters [22]. Perversely, there always remains a non-null estimation error if the ESO observer is not appropriately selected.

According to this estimation error, an auxiliary continuous fractional-order fast terminal sliding mode controller is added to the ESO based model-free control. Moreover, the continuous fractional-order fast terminal sliding mode controller (FOFTSMC) has great results which ensure fast convergence and high tracking accuracy under substantial lumped uncertainties due to the fractional-order non-singular fast terminal sliding mode (FONTSM) surface and fast-TSM-type reaching law, see [23]. In this paper, the full control strategy that we propose will be referred to ESO based on MFFOSMC.

The proposed controller mainly has three components. First, the ESO term used to estimate the unknown uncertain dynamics via the knowledge of the control input and output variables for an active suspension system. Second, the model-free term is used to reduce the existing controller's complexity, insert required performances, and decrease the high order derivative output. The last one, the continuous fractional-order fast terminal sliding mode control term is used to overcome the ESO estimation error and ensure finite-time convergence. Working with virtual program technology has shown potential to improve the product development process [24–26]. Using simulations generated by the Co-Simulation platform, modeling was performed using the software LMS AMESim and the control part was configured by Matlab/Simulink. The results demonstrate that MFFOSMC outperforms iPID, PID and TDEC controllers in ride comfort and ride safety.

The remainder of this paper is organized as follows. In Section 2, the dynamic model of the nonlinear quarter vehicle active suspension system and the system requirements are described. In Section 3 detailed design procedure of the model-free fractional-order sliding mode controller is presented. Section 4 gives the simulation results to prove the performance of MFFOSMC controller. The MFFOSMC controller is implemented in Co-Simulink platform based on Matlab/Simulink and LMS-AMESim environment compared with classic PID, TDEC, and iPID controllers. Finally, in Section 5, the conclusion and future work are summarized.

2. A quarter car active suspension model and system requirements

A quarter car active suspension model which considered in this work, is taken from [27] as shown in Fig. 1. A detailed nonlinear model and an actuator dynamics are deemed to make a model that accurately matches the quarter car active suspension system dynamic behavior which is defined below:

2.1. Active suspension system

The principal task of the suspension system is to reduce the resonance peak at the natural frequency of the unsprung mass without transferring the reaction force directly to the chassis. The nonlinear active suspension constructed based on the Newton law which is presented in Eq. (1), where m_c is the sprung mass, and it represents the vehicle chassis, m_w is the unsprung mass, and it represents mass of the vehicle wheel, $F(t)$, $F_c(t)$, $F_d(x,t)$, $F_{wc}(x,t)$ and $F_{wd}(t)$ are denote the active control force generated by the actuator, the force produced by the spring and damper of suspension system and the force produced by the springs and dampers of the wheel, respectively.

x_c and x_w denote the displacements of the body and the axle masses, respectively, where x_g denotes the road disturbance input.

The dynamic model of the sprung and unsprung masses are given as [27]:

$$\begin{cases} m_c\ddot{x}_c(t) = -F_c(t) - F_d(x,\,t) - F_{f,i}(x,\,t) + F(t) \\ m_w\ddot{x}_w(t) = F_c(t) + F_d(x,\,t) + F_{wc}(x,\,t) + F_{wd}(t) + F_{f,i}(x,\,t) - F(t) \end{cases} \tag{1}$$

where $F_{f,i}(x,\,t)$ is the Coulomb friction forces which expressed as $F_{f,1}(x,\,t)$ in the suspension and $F_{f,2}(x,\,t)$ in the vertical linear guides of the chassis mass as in [28], for smooth zero crossings, tanh-function is used to approximate the Coulomb friction forces, see [29].

$$F_{f,i} = \widehat{F}_{f,i}\tanh(\Delta v_i k_{f,i}) \tag{2}$$

For the quarter-vehicle model of the test rig's suspension configuration, the state vector is chosen as:

$$X = [x_c \quad \dot{x}_c \quad x_w \quad \dot{x}_w \quad x_{lm,1} \quad x_{lm,2}]^T \tag{3}$$

where $x_{lm,1,\ 2}$ is the actuator state and the output vector can be represented as:

$$Y = [\ddot{x}_c \quad F_{dyn} \quad x_c - x_w]^T \tag{4}$$

which contains the measurement signals representing a realistic sensor configuration of modern production vehicles.

Where $\ddot{x}_c$ is the sprung mass acceleration, $x_c - x_w$ is the displacement between the sprung and unsprung masses and F_{dyn} is the dynamic wheel load which denoted as below:

$$F_{dyn} = F_{wc}(x,\,t) + F_{wd}(t) \tag{5}$$

2.2. The actuator subsystem

An actuator is inserted into the vehicle suspension system to apply forces between chassis and unsprung mass depending on the relative

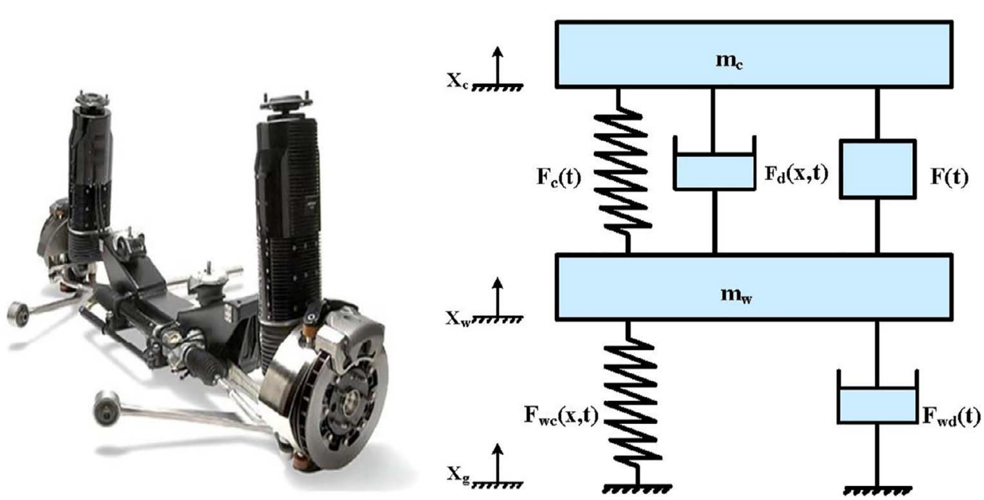

Fig. 1. The nonlinear quarter-vehicle model.