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Reducing Displacement of Spring Mass in Active Vehicle Suspension System Using Sliding Mode Controller Based on Disturbance Observer

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Abstract

The main cause of oscillation during the movement of the vehicle is the unevenness of the road. Therefore, in order to maintain the stability of the car in swing states, the suspension system plays an essential role. Therefore, the active suspension system is used to replace the conventional passive suspension system, to improve comfort and smoothness. To reduce the displacement of the spring mass in the active vehicle suspension system, a high-order sliding mode controller is proposed in this paper. Uncertainty of system parameters, nonlinear characteristic of damping and spring, load changes and unknown path disturbance are estimated by disturbance observer. The controller only needs the information of the spring mass state variables and therefore does not need separate sensors to measure the suspension mass state variables. Particle swarm optimization algorithm has been used to determine the control parameters. The efficiency of the proposed method has been shown using simulation in MATLAB software and the results have been compared with the passive suspension system.

Keywords: active suspension, disturbance observer, higher order sliding mode controller, particle swarm optimization
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1. Introduction

Active vehicle suspension systems have attracted a lot of attention due to the significant improvement in passenger ride comfort from the point of view of eliminating the effect of road irregularities [1,2]. Vehicle suspension systems, like most physical systems, have nonlinear characteristics and are subject to uncertainty and unknown disturbances [3,4]. Active suspension design is a challenging study, as ride comfort must be improved despite model uncertainties, and track disturbance input, while maintaining stability and road adhesion due to the limitations of suspension component motion space [5,6].

So far, many control strategies such as sliding mode control [7], adaptive finite-time control [8], saturated feedback control [9], driving state adaptive control [10] and virtual reference feedforward controller [11] have been implemented in active suspension systems [12,13].

An output feedback finite-time control method for stabilizing a chaotic vehicle active suspension is proposed to improve the suspension performance in [14], which is continuous and can completely eliminate the matching disorder.

The SMC (Sliding Mode Control) algorithm for active suspension performance control is presented in [15], and five mode variables are considered to optimize the algorithm. Also, the output signal of the model is considered by the fifth derivative, and the error signal is also considered by the fourth derivative. The simulation results using the Matlab-Simulink environment show that the displacement and acceleration of the spring mass are significantly reduced when the vehicle uses the active suspension system controlled by the SMC algorithm.

A discrete switching-free network sliding mode control design for active suspension system, using an event-triggered approach and a focus activation network delay compensation scheme is proposed in [16]. The delay of random excitation network in forward and feedback channel is modeled using Poisson distribution and compensated using Thiran's approximation technique. The stability of the closed-loop system ensures the finite-time convergence of the state variables of the system in the specified sliding band.

The design of an adaptive sliding mode control, based on nonlinear disturbance observer, to obtain passenger comfort and maintain driving safety of pneumatic active suspension system is presented in[17].

An integral terminal sliding mode control (TSMC) is designed for an active nonlinear vehicle suspension system under external disturbances and uncertainties in [18], where the integral TSMC is designed to deal with uncertainties and external disturbances in the system when the upper bound is known, and then an adaptation law is recommended to estimate the upper bound of uncertainties and external disturbances.

A switching logic based saturation tracking control scheme for active suspension control is proposed in [19], where the designed controller changes its structure to remove or retain the disturbance based on a well-designed new disturbance effect indicator.

Spring mass deviations are largely caused by the unknown profile of the road, and in the absence of sensors to detect road disturbances, a control plan with the ability to estimate their effect seems necessary. The presence of position sensors to measure the position and speed of spring mass and non-spring mass will lead to an increase in hardware cost and a decrease in system reliability. Therefore, the design of the controller with the minimum need for position sensors helps to solve the mentioned problems.

The problem of improving the comfort of car passengers has been investigated in this study, which depends on the displacement of the spring mass and its rate of change.

An active suspension system is considered using a high-order sliding mode controller along with a disturbance viewer. In the standard sliding mode controller, if the limits of disturbances and uncertainties are known, their effect on the system response can be removed. Road turbulence can vary significantly even over a short distance, thus making uncertainty bounds difficult. In order to solve this problem, a method to estimate the effect of path uncertainty and disturbance has been considered. Among the highlights of this research, the following can be mentioned:

- The effect of factors such as unknown path disturbance, nonlinear characteristic and uncertainty in the suspension system is estimated using the viewer;
- The controller does not need non-spring mass state variables, and as a result, the reliability of the system is improved and the cost of the system hardware is reduced;
- Unlike the standard sliding mode control, there is no need to determine the limits of disturbance and uncertainty;
- Controller parameters are determined using particle group optimization algorithm.

The organization of this article is as follows: in the second part, a one-fourth vehicle model and the corresponding dynamic relations are obtained in the majority of the state space equations. The high-order sliding mode controller is described in the third section and the perturbation viewer is given in the fourth section. The necessary conditions for the stability of the proposed controller are discussed in the fifth section. The sixth part is dedicated to the particle group optimization algorithm. The efficiency of the proposed controller is shown by the simulation results in the seventh section, and at the end of the eighth section, it includes the conclusion and suggestions for future works.

2. System Modeling

The spring mass $m_s(t)$ consists of the total mass of the car body, passengers and internal components, and it may change significantly according to the load conditions of the car passengers. This mass is held by a suspension system including a spring with stiffness k_s and a damper with coefficient c_s. The spring is modelled by a linear stiffness coefficient k_{1s} and a nonlinear stiffness coefficient k_{2s}, and the damper is modelled by a linear damping coefficient c1 and a nonlinear damping coefficient c2. The mass of the wheel, tire, brake, and suspension interface is known as the un-sprung mass mu, which is held on the tire. The tire is modelled by a combination of spring and linear damper with k_t and c_t coefficients, respectively. In an active suspension system, in addition to the aforementioned passive components, an actuator is placed between the spring mass and the non-spring mass. This actuator produces a control force u, which is responsible for eliminating oscillations.

It is assumed that no initial information about the unknown path profile is available. The vertical disturbance of the path that affects the non-spring mass is denoted by z and the vertical displacement of the spring mass and the non-spring mass with respect to the corresponding static position is specified by X_s and Xu, respectively. The dynamic equations of the suspension system are mostly expressed in the state space equations.

State space is one of the methods of representing system equations to study and simulate system behavior [20-23]. State variables are defined as follows:

$$\begin{cases} x_1 = x_s \\ x_2 = \dot{x}_s \\ x_3 = x_u \\ x_4 = \dot{x}_u \end{cases} \tag{1}$$

The purpose of the control is to reduce the fluctuations of the spring mass for riding comfort by measuring the position and speed variables of the spring mass. The motivation for using spring mass measurements is to simplify the control algorithm and hardware system by avoiding the use of sensors installed on the wheel and tire assembly. To achieve the considered goals, the high-order sliding mode controller strategy along with the disturbance viewer has been used.

The equations of motion of the studied system model can be expressed as follows in the representation of the state space equations:

$$\dot{x}_1 = x_2 \tag{2}$$

$$\dot{x}_2 = \frac{1}{m_s(t)}(-f_s - f_d + u) \tag{3}$$

$$\dot{x}_3 = x_4 \tag{4}$$

$$\dot{x}_4 = \frac{1}{m_u}(f_s + f_d - f_t - u) \tag{5}$$

where f_a and f_d are symbols of spring force and damping, respectively. Tire force f_t is obtained from the sum of tire spring force and tire damping force.

3. High Order Sliding Mode Control

In sliding mode control, a suitable sliding surface with desired dynamics is selected and the control signal is chosen so that the dynamics of the system is maintained on the sliding surface [24,25]. Therefore, the system becomes insensitive to uncertainties and its behavior is determined based on the definition of the slip surface [26,27].

The sliding surface is determined according to the following relationship [28]:

$$\sigma = Sx_1 + x_2 \tag{6}$$

where S is a user-defined constant. The selection of the mentioned level clearly shows that the control goal is achieved only by applying the desired behavior to the dynamics of the spring mass.

The control signal is divided into two parts u_{eq} and u_n. The u_{eq} section is intended to compensate for the known sentences and un to compensate for the integrated uncertainty e in the model dynamics. The main idea of the scheme is to estimate the internal and external uncertainties in the system using the disturbance viewer and then use the estimated value in un to remove their effect.

4. Disturbance Viewer

Here, the disturbance viewer is considered a very important module so that by accurately estimating the amount of uncertainty, it can meet the conditions of the sliding mode. Also, the stability of the whole proposed controller system depends on the dynamic stability of the viewer. We assume that the estimate of integrated uncertainty e is described by the following relation:

$$\hat{e} = \hat{d}(t) + p(\sigma) \tag{7}$$

where $p(\sigma)$ is a linear or non-linear function of the slip surface variable σ. Here, the function $\hat{d}(t)$ should be updated so that the estimation error $\tilde{e}$ converges to zero. By deriving, we arrive at the dynamics of uncertainty estimation:

$$\hat{e} = \hat{d}(t) + \frac{\partial p}{\partial \sigma}\delta \tag{8}$$

We insert the sliding surface dynamics into the above equation:

$$\hat{e} = \hat{d}(t) + \frac{\partial p}{\partial \sigma}(-\nu + e + G_m u_n) \tag{9}$$

Therefore, the dynamics $\hat{d}(t)$ is suggested according to the following relationship:

$$\hat{d}(t) = -\frac{\partial p}{\partial \sigma}(-\nu + e + G_m u_n) \tag{10}$$

5. Particle Swarm Optimization

Particle swarm optimization (PSO) is a robust optimization technique based on group motion and intelligence and is generally used to find answers to complex optimization problems [29,30]. The creation of this algorithm was inspired by the group behavior of creatures such as birds in nature [31]. In the particle swarm optimization algorithm, a set of particles move in the search space to achieve the all-encompassing optimal point, and at the same time, the particles interact with each other directly or indirectly using the search direction. In each iteration of the algorithm, each particle in the group updates its position according to the velocity of the particle, the previous best position of the particle and the best overall position of the particles [32,33].

It should be noted that determining the best individual position of each particle and the best overall position of particles is done according to the merit function, which is created according to the integral of the absolute magnitude of the deviation of the position of the spring mass from its static position [34,35]. In this research, PSO technique has been applied to find the optimal value of K_1, K_2, K_3 and K_4 parameters.

6. Simulation Results

In this section, the efficiency of the proposed controller has been investigated using suspension model simulation. The route profile is considered according to fig. 1. The simulation results are compared with the performance of the passive suspension system. The initial conditions of all state variables are assumed to be zero.

The control parameters are: S=0.5, $\partial p/\partial \sigma = 1$, $C_1=C_2=2$ and W=0.9.

Also, for the system with the nominal parameters of the suspension system, after running the optimization algorithm with the number of particles 30 and the maximum repetition 20 times, the control parameters K_1, K_2, K_3 and K_4 were obtained as follows: $K_1=1.8898$, $K_2=2.0193$, $K_3=3.6694$ and $K_4=4.1035$.

The response of the system to the track disturbance with passive suspension system and with active suspension system with nominal parameters is shown in figs. 2 and. 3, respectively. For the strength of the proposed design, another simulation has been performed with a 20% increase in the actual parameters against the corresponding nominal values and a 60 kg increase in the spring mass, the results of which are shown in fig. 4.

Fig. (2-a) shows that in the passive suspension system, the amplitude of spring mass fluctuations may even exceed the amplitude of track disturbances, which greatly affects the ride comfort. Simulation results with nominal and non-nominal active suspension systems show that the proposed controller almost completely eliminates the effect of track disturbance and parametric uncertainties.

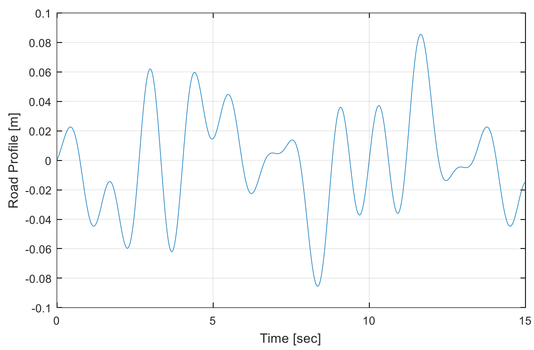
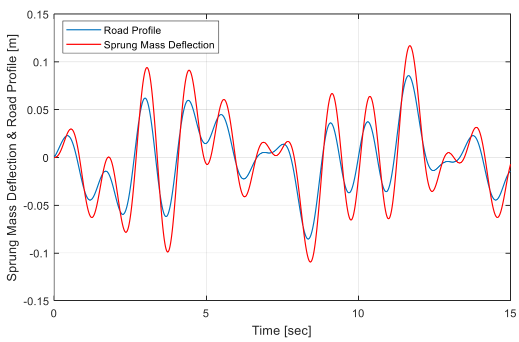
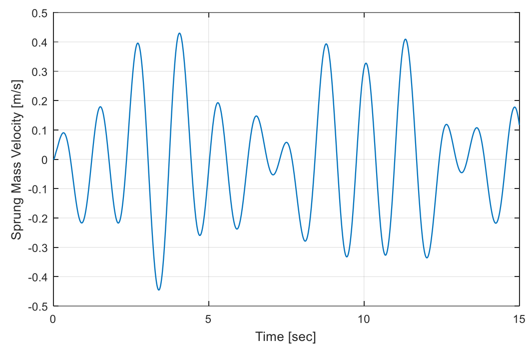


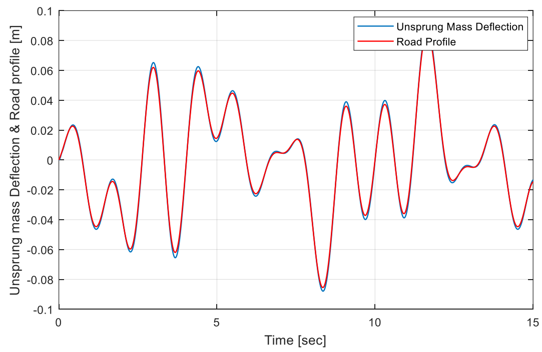
Fig. 1. Path disturbance profile



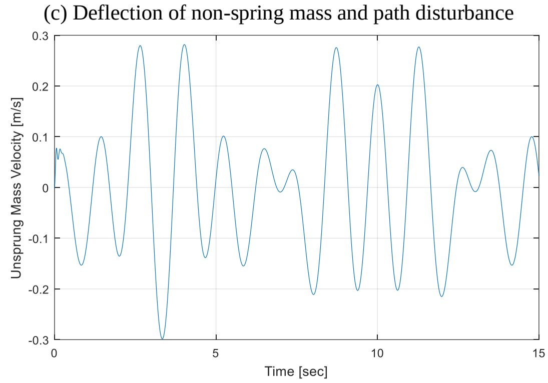
(a) Sprung mass deflection and track disturbance



(b) Sprung mass velocity

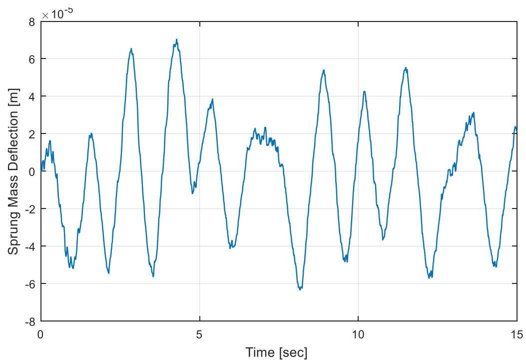


(c) Deflection of non-spring mass and path disturbance

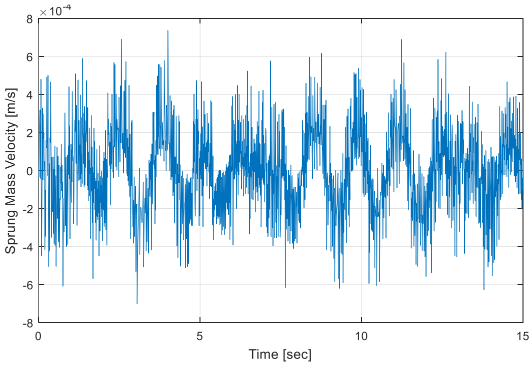


(d) Velocity of non-spring mass

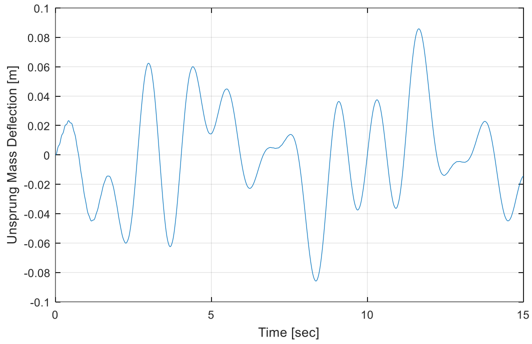
Fig. 2. Passive suspension response to track disturbances



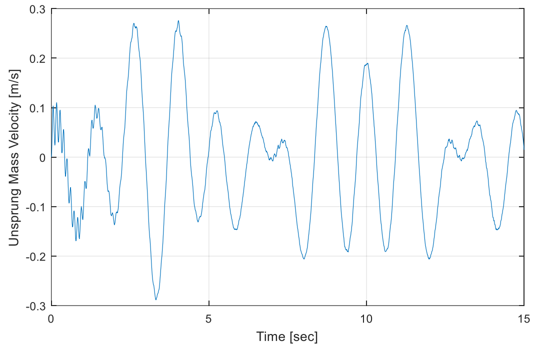
(a) Sprung mass deflection and track disturbance



(b) Sprung mass velocity

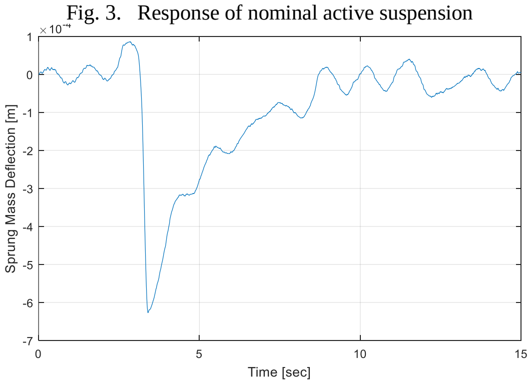


(c) Deflection of non-spring mass and path disturbance

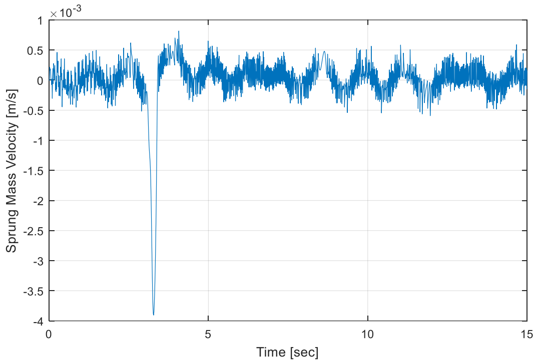


(d) Velocity of non-spring mass

Fig. 3. Response of nominal active suspension



(a) Spring mass deflection



(b) Spring mass velocity

Fig. 4. Response of non-nominal active suspension system to track perturbations

7. Conclusion

An active suspension system makes the road vehicle move with more stability, safety and comfort by keeping the wheels in contact with the road surface. In this paper, a sliding mode control scheme based on disturbance viewer was designed for active suspension system and stability conditions were investigated. In the design, a part of the control signal is considered according to the high-order sliding mode control. The proposed scheme works with high accuracy in estimating the uncertainties and path disturbance input. Some control parameters were determined by particle swarm optimization algorithm. The simulation results for the track disturbance profile show that with a series of control parameters, the ride comfort characteristic is improved in the nominal suspension state and with changed parameters according to the amount of spring mass displacement. In the design done, the dynamics of the drive is not considered. Therefore, for the future study, it is possible to study the effect of dynamic as well as the delay of various actuators such as pneumatic or hydraulic on the system response and the methods of reducing their adverse effect.

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