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Automotive Active Suspension System Design for a Quarter Bus: An Acceleration Feedback Approach

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Abstract

Although most of the control designs are based on position and velocity feedback, in general acceleration is easier to measure than position or velocity. In contrast to position and velocity sensors, accelerometers are low-cost and efficient measurement devices. As a natural result of these, they are widely used in mechanical systems. Thus, these situations increase the importance of acceleration feedback control designs for mechanical systems. Passing the acceleration filter through a second order filter with substantial damping and generate a force feedback proportional to the output of the filter is one of the popular approaches for the acceleration feedback control design. Since this control approach is based on adjusting two control parameters properly, it is easy to use and generally gives satisfactory results. In this study, an automative active suspension system design for a quarter bus is addressed by utilizing the mentioned acceleration feedback approach. Modeling approach of an active suspension system of a quarter bus, acceleration feedback control design and effect of the change of control parameters are examined in a detailed manner. Performance of the designed controller is demonstrated via numerical simulation results realized in Matlab.

Keywords: Active suspension control, Acceleration feedback control, Quarter bus

INTRODUCTION

Controlling the vertical movement of the wheels relative to the chassis or vehicle body with an onboard system is called as an active suspension and active suspension is a type of automotive suspension. A higher degree of ride quality and car handling by keeping the tires perpendicular to the road in corners and allowing better traction and control are achieved in assistance of this technology. Body movement is detected by an onboard computer from sensors throughout the vehicle and the action of the active suspension is controlled by using data calculated by opportune control techniques. Body roll and pitch variation in many driving situations including cornering, accelerating and braking is virtually eliminated by this system. The suspension system of the vehicle is modeled as a mass-spring-damper system and the mentioned variations are observed at an acceleration level. Therefore, active suspension system design is basically considered as a vibration control [1]. Eliminating this vibration as quick as possible by considering the ride quality, car handling and the comfort of driver and passengers is the main purpose of this control design.

In closed-loop control systems include vibration control systems, a feedback information about the system is needed during the control process for providing the persistence of the control. The necessary control input is calculated according to this information. In the most of the control designs position and velocitys feedback have been used for this purpose. Examples of this usage for an active suspension design of a quarter car can be seen in [2,6].

In [2], a linear quadratic Gaussian (LQG) optimization based control design was realized for an active vehicle suspension. The connections between LGQ-optimal 1 degree-of-freedom (DOF) and 2 DOF models were explored in this study. In [3], Rao and Prahlad suggested a novel fuzzy-logic-based control for vehicle-active suspension. Thanks to this design vibration of the vehicle and disturbance were considerably reduced. A vertical acceleration was reduced by controlling the active suspension system of a hypothetical reference model of a quarter car. Yoshimura et al. concerned with the construction of an active suspension system for a quarter car model using the concept of sliding mode control in [4]. In this study, the active control is derived by the equivalent control while the switching function where the sliding surface is obtained by using linear quadratic control theory. The road profile was estimated by using the minimum order observer based on a linear system transformed from the exact non-linear system and the active control was generated with non-negligible time lag by using a pneumatic actuator. In [5], a robust linear controller design for an active suspension mounted in a quarter car test-rig was presented. This approach does not require a physical model of either the car or the shock-absorber. Linear black box models of a quarter car test-rig was identified via frequency domain identification techniques and H_∞ and μ-synthesis control design techniques account for the model uncertainties introduced by the linear model approximation of the nonlinear dynamics. A proportional-integral-derivative type controller was developed for active suspension system of light passenger vehicle to improve the comfort of the passengers in [6]. In this study, system was subjected to bumpy road and its performance was assessed and compare with a passive suspension system.

According to authors' best knowledge, in contrast to position and velocity feedback designs, there is no available acceleration feedback control approach for an active suspension design of a quarter bus. An acceleration feedback approach for an active suspension design of a quarter bus is aimed in this study. To realize this purpose, acceleration is passed through a second order filter with substantial damping and a force feedback proportional to the output of the filter is generated. Modeling approach of an active suspension system of a quarter bus and an acceleration feedback control design are examined in a detailed manner. Performance of the designed controller is demonstrated via numerical simulation results realized in Matlab.

MATHEMATICAL MODEL of the SYSTEM

The active suspension system of a quarter bus is represented with the mass-spring-damper system as shown in Fig. 1 [7]. In this system, mass of the quarter bus and suspension are denoted by M_1 and M_2, respectively. Spring and damping constants of the suspension system are denoted by K_1 and b_1 and spring and damping constants of wheel and tire are denoted by K_2 and b_2, respectively. U denotes the control force while the disturbance at the ground level is denoted by W. Finally, X_1 and X_2 denote the movement of body and the suspension. The mathematical model of the open-loop system is established via Langrange equations as

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{C}\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{u} \tag{1}$$

where **M, C** and **K** are the mass, damping and rigidity square matrices, respectively. The vector of the system states is denoted by $\mathbf{q} = [\,X_1\,X_2\,]^T$ where T stands for the transpose while the input column vector is denoted by **u**. Structures of the system matrices and input column vector are found as

$$\mathbf{M} = \begin{bmatrix} M_1 & 0 \\ 0 & M_2 \end{bmatrix},\quad \mathbf{C} = \begin{bmatrix} b_1 & -b_1 \\ -b_1 & b_1+b_2 \end{bmatrix}$$
$$\mathbf{K} = \begin{bmatrix} K_1 & -K_1 \\ -K_1 & K_1+K_2 \end{bmatrix},\quad \mathbf{u} = \begin{bmatrix} U \\ b_2\dot{W}+K_2W-U \end{bmatrix}. \tag{2}$$

Undamped natural frequencies, ω, are found by solving the following eigenvalue equation [8]

$$\left| -\omega^2 \mathbf{M} + \mathbf{K} \right| = 0. \tag{3}$$

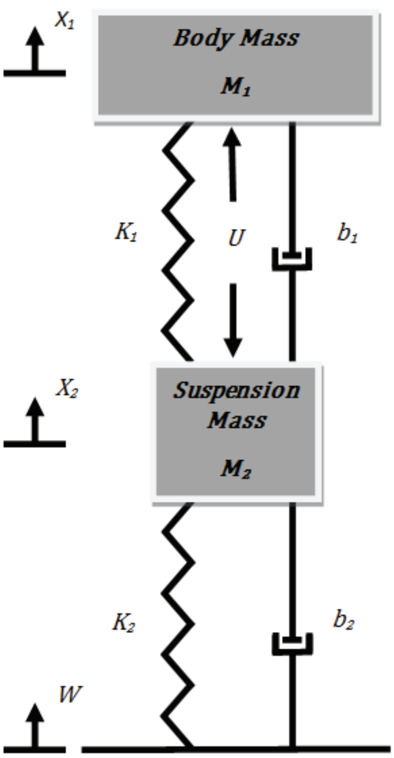

Figure 1. Model of a quarter bus active suspension system

CONTROL APPROACH

The block diagram of the closed-loop control system is shown in Fig. 2. In this figure s^2 represents the accelerometer dynamics where s stands for Laplace variable. The controller dynamics is denoted by $D(s)$ while $r(t)$ represents the reference input. Transfer functions from control force and disturbance to system output are denoted by $G_U(s)$ and $G_W(s)$, respectively. Passing the acceleration signal through a second order filter with substantial damping and generating a force feedback proportional to the output of the filter is the basic idea of this approach. As it can be seen from Fig. 2, a second order filter structure must be used for each modes that will be controlled [9].

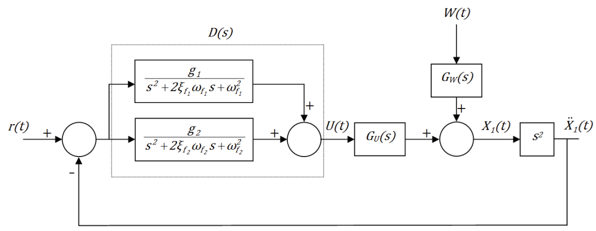

Figure 2. Closed-loop block diagram

The structure of the controller dynamics is a parallel combination of these second order filter structures and expressed as

$$D(s) = \sum_{i=1}^{2} \frac{g_i}{s^2 + 2\zeta_{f_i}\omega_{f_i}s + \omega_{f_i}^2} \tag{4}$$

where g_1 and g_2 denote the constant control gains. One can check that the combined system is always stable for positive selections of these control gains. The other filter dynamics are denoted by ζ_f and ω_f. All of these parameters can be considered as the adjustable parameters of the controller. According to the comparison given in [9], for the same values of ζ_f the mode with the natural frequency close to ω_f is more heavily damped. Thus the controller's performance relies on the tuning of the filter on the mode that is damped. In [9], it was also observed that the maximum achievable damping ratio increases with ζ_f but the interval between 0.5 and 0.7 is recommended for ζ_f for guaranteeing the stability. Finally, the gains g_1 and g_2 can be adjusted via trial-and-error method by considering these selection conditions.

SIMULATION RESULTS

For the simulation studies, numerical values of the system parameters are used as given in Table 1 [7]. At this point it should be noted that these values are selected by considering the reality of the simulations. It is considered that the vehicle encountered with a bump with 10 cm height during its travel. Simulation studies are realized in Matlab environment.

Table 1. System Parameters [7]

Description	Symbol	Value	Unit
Quarter bus body mass	M_1	2500	kg
Suspension Mass	M_2	320	kg
Spring constant of suspension system	K_1	80000	N/m
Spring constant of wheel and tire	K_2	500000	N/m
Damping constant of suspension system	b_1	350	Ns/m
Damping constant of wheel and tire	b_2	15020	Ns/m

The open-loop response of the displacement of the body according to this scenario is obtained as it can be seen in Fig. 3. From this figure it can be seen that the vibration at the body level started with 100% overshoot. It means that the body rises up to 20 cm when the vehicle encounter with a 10 cm height bump. This situation is not only very disturbing for driver and passengers but also decreases the ride quality and handling of the bus. Moreover, the vibration of the body get to desired level approximately in 40 seconds. This value is an unacceptable for standard driving conditions.

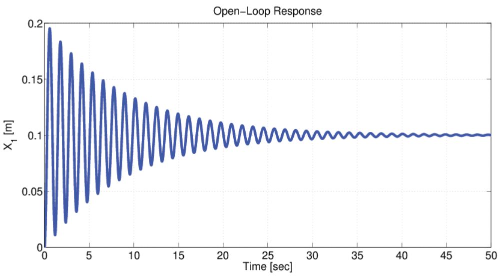

Figure 3. Open-loop response

For the controlled case, the undamped natural frequencies of the system are calculated by utilizing Eq. (3) to adjust the ω_f values of the filters for both modes. The undamped natural frequencies are found as 5.25 and 42.58 rad/s. The other parameters of the filters are selected as follows via trial-and-error method by considering aforementioned selection recommendations

$$\begin{aligned} \zeta_{f1} &= \zeta_{f2} = 0.7 \\ g_1 &= g_2 = 100. \end{aligned} \tag{5}$$

The closed-loop response of the displacement of the body is obtained as in Fig. 4. From this figure it can be seen that the overshoot decreased to approximately 45% while the vibration of the body get to desired level approximately in 5 seconds. From these results it can be clearly seen that the controller is performed efficiently.

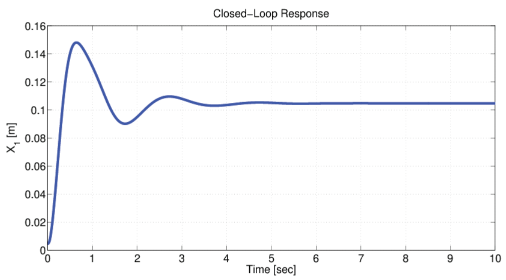

Figure 4. Closed-loop response

CONCLUSIONS

In this study, active suspension system design for a quarter bus is addressed. In contrast to position and/or velocity feedback designs that are available in the literature, an acceleration feedback control approach, that has been used in little studies, is used to reach this purpose. This situation can be considered as the main novelty of this study. According to the given approach, the acceleration signal passed through second order filter structures and force feedbacks proportional to the output of the filters are generated to control the system. These filters are specified according to the modes of the system that are wanted to be damped. The control design process is examined in a detailed manner. Efficiency of the designed controller is demonstrated via numerical simulation studies realized in Matlab environment. It is observed that the open-loop response of the controller is considerably developed by utilizing the proposed approach.

Although the proposed method is a feasible solution, trial-and-error method that must be used to adjust the control gains can be seen as the disadvantage of this method. To overcome this disadvantage, proposing an appropriate method to adjust the control gains and realizing the experimental verification of the proposed method are considered as possible future works.

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