A Quarter Active Suspension System Based Ground-Hook Controller

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Abstract. An alternative design technique for active suspension system of vehicle using a developved ground-hook damping system as a reference is proposed. The controller parameters are determined using Lyapunov method and can be tuned to precisely achieve the type of desired response which given by reference model. The simulation result show that the designed active suspension system based ground-hook reference model is able to significantly improve the ride comfort and the road holding compared with semi-active suspension.

1 Introduction

In recent years there has been a significant growth in the design of suspension systems by which should reduce vibration due to the external disturbances to the maximum possible degree [1-16]. Two criteria of good vehicle suspension performance are typically their ability to provide good road handling and increased passenger comfort. It is difficult to meet both requirements, since this involves opposing tendencies in the selection of the main parameters of suspension, i.e. stiffness and damping. Suspension should be rigid enough to effectively carry the static load, and soft enough to ensure the good isolation of vibrations. These two factors are related to the body vertical acceleration and the suspension deflection. If the suspension system is unable to absorb the vibration caused by the unevenness of the road surface, a significant deflection of the suspension will occur resulting unsafe condition of the vehicle. Moreover, the sprung mass of the vehicle will also exhibit a large vertical acceleration which can be the reason of a ride discomfort. In order to achieve a good ride quality, the performance of the suspension system plays an important role.

Since most of the vehicle suspensions do not operate with a single well defined roadway input, real suspensions are rarely optimal for any particular road and speed. On smooth roads the suspension will be stiffer than necessary and on rough roads softer that desirable. This leads to the concept of an adaptable suspension which can change its characteristics bases on measured performance. By using an adaptation algorithm, it is expected that the optimal performance can be maintained over the wide range of input conditions typically encountered by a vehicle. In this paper, an alternative design technique for active suspension system of vehicle using adaptive controller based skyhook damper model is proposed. The reference model and the controller systems are designed and determined using skyhook model and Lyapunov method, respectively. The control design was applied to two different suspension models, namely the body vertical acceleration and the suspension deflection models, each representing comfort and safety factors.

2 Dynamic of suspension system

2.1 A Quarter model of suspension system

In most cases, dynamic systems are stimulated by the movement of anchorages. The suspension system of car is stimulated as a result of moving on the ruggedness of the road. A car is moving on a bumpy road in which some forces are exerted on the passengers as the result of the change of the length of springs and dampers. The schematic diagram of a quarter-vehicle suspension system under investigation can be presented as in Fig. 1, where ms and mu each represents the sprung mass and unsprung mass respectively, c_a represents absorber coefficient, k_a represents spring constants, k_t represents the tire equivalent constants, z_r represents surface roughness of road, z_s represents vehicle body position and z_u represent the tire vehicle position. The variable Q represents an applied actuator force for the active suspension. The values of the parameters in this model, which are taken from a typical passenger car, are collected in Table 1. However, the value of k_a is unknown therefore it can be assumed that there is no limitation in generating the actuator force Q. Using Newton's second law, the equations of motion for the model in Fig. 1 are given by

$$m_s \ddot{z}_s = -k_a (z_s) - z_u - Q \qquad (1)$$

$$m_u \ddot{z}_u = k_a (z_s - z_u) - k_t (z_u - z_r) + Q. \qquad (2)$$

By defining $x_1 = z_s$, $x_2 = \dot{z}_s$, $x_3 = z_s - z_u$, and $x_4 = \dot{z}_s - \dot{z}_u$, then the equations (1) and (2) can be expressed in the state space form as follows:

$$\dot{X} = AX + BQ + Cz_r \tag{3}$$

where, the matrices A, B, and C are given by.

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & \dfrac{k_a}{m_s} & 0 \\ 0 & 0 & 0 & 1 \\ \dfrac{k_t}{m_u} & 0 & -\left(\dfrac{k_a}{m_s} + \dfrac{k_a + k_t}{m_u}\right) & 0 \end{bmatrix},$$

$$B = \begin{bmatrix} 0 & -1/m_s & 0 & -(1/m_s + 1/m_u) \end{bmatrix}^T,$$

$$C = \begin{bmatrix} 0 & 0 & 0 & -k_t/m_u \end{bmatrix}^T.$$
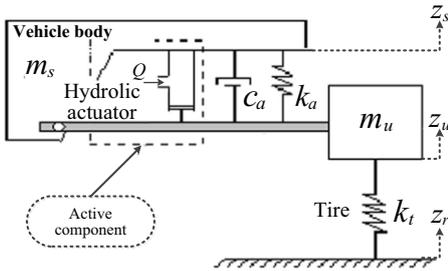
Figure 1. Schematic diagram of a quarter-vehicle model.

Table 1. System parameters.

Parameter	Value
Sprung mass (m_s)	412.5 kg
Unsprung mass (m_u)	51 kg
Tire stiffness (k_t)	210 N/m
Rattle space ($z_s - z_u$)max	91 mm

2.2 A ground-hook damper as reference model

In order to specify the desired performance of the suspension system, the reference model is derived from quarter-vihicle ground-hook model (see Fig. 2). Ground-hook proposed the control method Ground-hook that in which an assumed amortize has been incorporated between the sprung mass and the ground in order to reduce dynamic force of tire and to improve the conductibility of the car. In Fig. 2, The linear equation of the state for reference model Ground-hook from the order of four has been presented as follows:

$$m_s \ddot{z}_s = -k_b(z_s - z_u) - c_b(\dot{z}_s - \dot{z}_u) \tag{4}$$

$$m_u \ddot{z}_u = k_b(z_s - z_u) - c_g \dot{z}_u + c_b(\dot{z}_s - \dot{z}_u) - k_{t1}(z_u - z_r) \tag{5}$$

where, c_b and c_g are indicate a variable damper and a coefficient that connected between the unsprung mass

and the fixed fictitious frame on the ground, respectively. $\dot{z}_s$, $\dot{z}_u$, $\dot{z}_s - \dot{z}_u$ are the velocities of sprung mass, unsprung mass, and relative velocity between sprung mass and unsprung mass. Theoretically, ground-hook damper model will improve the responses of the unsprung system. The ideal ground-hook damper model is given by: if $-\dot{z}_u(\dot{z}_s - \dot{z}_u) \geq 0$ then $f = c_g \dot{z}_u$, if $-\dot{z}_u(\dot{z}_s - \dot{z}_u) < 0$ then $f = 0$, where f is ground-hook damper model force. The generated forces by the ground-hook model are added to the passive vehicle model which used as a reference on active suspension controller design.
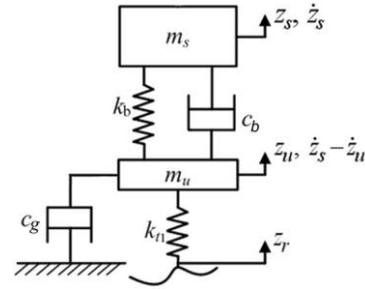

Figure 2. Ground-hook reference model of a quarter-vehicle suspension system.

In the state space form, equations (4) and (5) is written as

$$\dot{X}_d = A_d X_d + C_d z_r \tag{6}$$

where the matrices A_d and C_d are given by

$$A_d = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & -c_g/m_s & -k_b/m_s & -c_b/m_s \\ 0 & 0 & 0 & 1 \\ \dfrac{k_{t1}}{m_u} & -\left(\dfrac{c_g}{m_s} + \dfrac{c_g}{m_u}\right) & -\left(\dfrac{k_b}{m_s} + \dfrac{k_b + k_{t1}}{m_u}\right) & -\left(\dfrac{c_b}{m_s} + \dfrac{c_b}{m_u}\right) \end{bmatrix}$$

$$C_d = \begin{bmatrix} 0 & 0 & 0 & -k_{t1}/m_u \end{bmatrix}^T.$$

3 Controller design

One of the main approach often used in adaptive control design is the model-reference adaptive system. Here, the desired performance is expressed in terms of a reference model, which gives the desired response to a command signal. The desired output of the model is compared to the output of plant, where its error is used to generate an actuating signal. One of most critical point in the design is how to make the error as small as possible. This will depend on the model and the command signal. If it would be possible to make the error equal to zero for all command signals, then perfect model-following would be achieved. Choosing the control law as follows:

$$Q = \theta X, \tag{7}$$

where $\theta = [\theta_1 \quad \theta_2 \quad \theta_3 \quad \theta_4]$ is an adjustable controller gain matrix, then equation (3) becomes

$$\dot{X} = (A + B\theta)X + Cz_r. \qquad (8)$$

Assumed that $C = C_d$, then the error and its dynamic between the states can written as

$$e = X - X_d. \qquad (9)$$

$$\dot{e} = \dot{X} - \dot{X}_d = A_d e + (A - A_d + B\theta)X \qquad (10)$$

In order to make e goes to zero, then there must be exist a set of unknown constant gain θ^* such that

$$A + B\theta^* = A_d. \qquad (11)$$

By substituting (11) into (10), we have

$$\dot{e} = A_m e + B\Phi X \qquad (12)$$

where $\Phi = \theta - \theta^*$. In order to drive the adjustment law of the parameter, the Lyapunov function is introduced as

$$V = \frac{1}{2}\left(\eta e^T P e + \Phi^T \Phi\right) \qquad (13)$$

$$\dot{V} = \frac{1}{2}\left(\eta e^T P e + \eta e^T P\dot{e} + 2\Phi^T \Phi\right) \qquad (14)$$

or

$$\dot{V} = \frac{1}{2}\eta e^T\left(A_d^T P + P A_d\right) + \eta \Phi^T B^T P e X^T + \Phi^T \dot{\Phi}. \qquad (15)$$

By chosing $\dot{\Phi} = -\eta B^T P e X^T$, then equation (15) can be simplify as

$$\dot{V} = -\frac{1}{2}e^T \Psi e. \qquad (16)$$

Then the parameter adjustment law is obtained as

$$\dot{\theta} = -\eta B_p^T B^T P e X^T. \qquad (17)$$

4 Discussions

To investigate and observe the performance of the designed adaptive controller to changing dynamics of the vehicle suspension systems, two different types of input signals (i.e., sinusoid and random signals) representing different types of road to stimulate the dynamics of suspension of the vehicle have been identified. It is desired that when the vehicle runs on the three different road types, the controller will choose the coefficients of controller θ such that the closed-loop output response of each model will be close to the model-reference response for any changes of the dynamic suspension system.

Revisiting the ground-hook scheme in Fig. 2, the variables c_b and c_g define the control gains that are used in computing u in proportion to $\dot{z}_s - \dot{z}_u$ and $\dot{z}_u$.

Fig. 3 shows that, as the damping coefficients c_1 and c_2 (to indicate c_b and c_g, respectively) increases, the peaks of the accelerations of the sprung and the unsprung masses at their natural frequencies decrease, while the acceleration levels in the frequency range 4-8 Hz and in the frequency range 12 Hz and higher increase. Decreasing c_b while increasing c_g results in an increase of the amplitude of the unsprung-mass natural frequency, with the gain of the reduction of the amplitude of other frequency ranges. Another notable observation is that the transmissibility to the sprung-mass acceleration with constant c_b in the higher frequency range of over 12 Hz is constant for any value of c_g. The dynamic response of the system for different values of the damping coefficients is clearly a weighted combination of the damping coefficients of the skyhook control. The controlled skyhook then is used as reference model in the adaptive controller design.
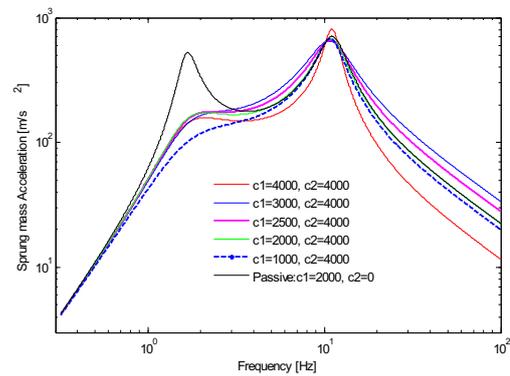

Figure 3. Comparison of the magnitude plots of the frequency responses of $\ddot{z}_s(s) / z_r(s)$ for various damping coefficients.

The magnitudes of the frequency responses of the sprung-mass acceleration with respect to the road disturbance are shown in Fig. 4. The solid line and the dotted line represent the magnitude plots of the frequency responses of the semi-active (with ground-hook controller) and the active. In Fig. 4, the dashed line depicts a large improvement according to the other values. However, the dashed line shows a deterioration of ride comfort in the frequency range 4-8 Hz and the deterioration of noise & harshness in the frequency range of 12 Hz and higher, when compared with the passive suspension.

If the results between semi-active and active suspension systems are compared in Figs. 5-9. It can be seen that, for the same input road disturbance in the form of random, the output response of semi-active suspension will follow the disturbance, whereas for the active suspension using the designed model reference adaptive technique then the output response only slightly react accordingly to the disturbance. The velocities and the accelerations of the sprung and unsprung masses (Figs. 6 – 9), respectively, of the active suspension were highly reduce compate to the only semi-active suspension control design. The performance of the sprung mass displacement and sprung mass veocity were improved about 20% and 35%,

respectively. The response of the sprung mass acceleration with the active one was poor at a certain point. This condition suspected as a accumulation of the sprung and unsprung masses amplitude. However, another performance improvement are achieved in the unsprung mass responses (velocity and acceleration). At the same condition with the sprung mass response, the velocitiy and the acceleration with active one were highly reduces which indicate that the road holding performance is improved. This means that the designed active suspension performs better than the semi-active one when the vehicle is subject to different road disturbance.
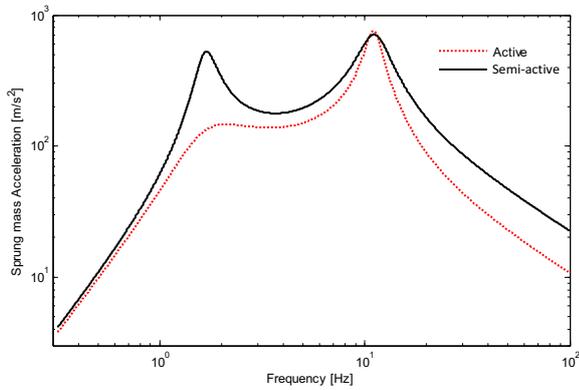
Figure 4. Comparison of the frequency responses of $\ddot{z}_s(s)/z_r(s)$.
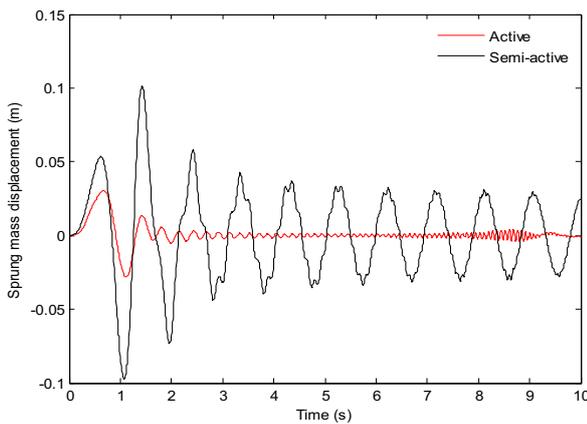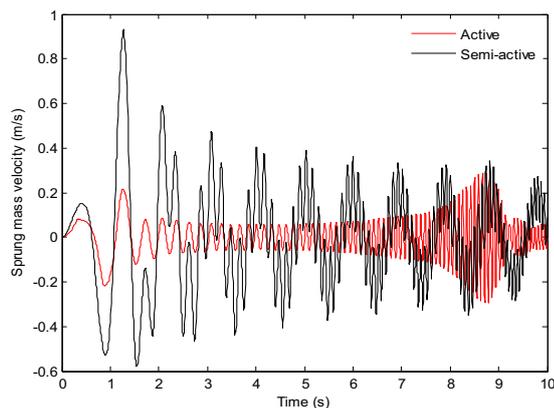

Figure 5. Sprung mass displacement response.
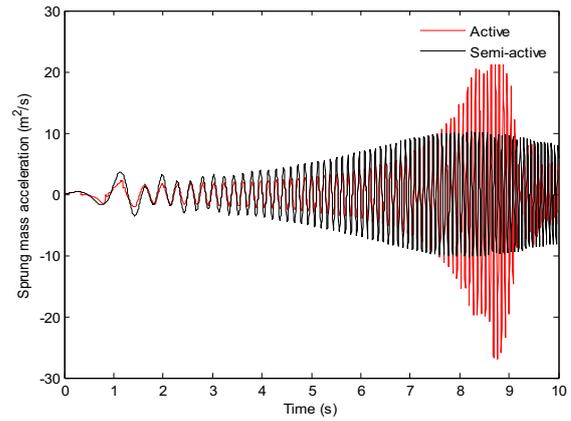

Figure 6. Sprung mass velocity response.

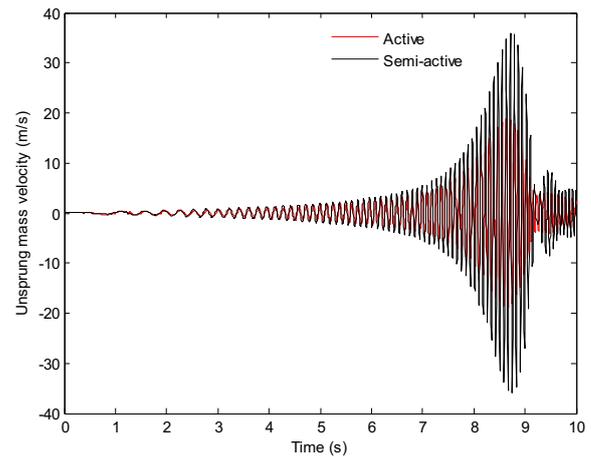**Figure 7.** Sprung mass acceleration response.

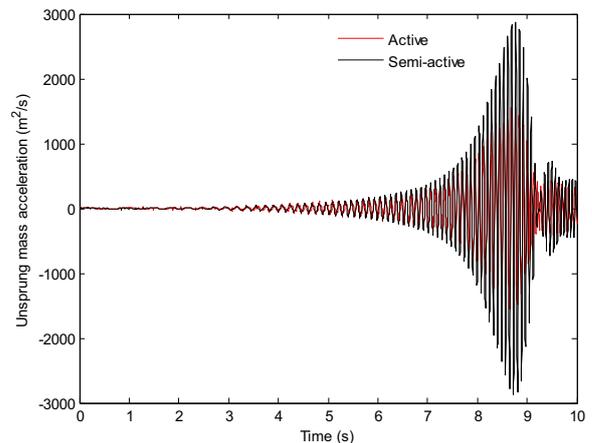**Figure 8.** Unsprung mass velocity response.

Figure 9. Unsprung mass acceleration response.

5 Conclusions

The controller based Lyapunov method and ground-hook reference model for suspension system is designed. The effects of the road excitation frequency and road roughness in association with the ride comfort and the road holding of the vehicle were studied. The simulation result show that an active suspension system with the

proposed control strategy is able to improve the ride comfort and the road holding, significantly compared with the conventional passive suspension systems as well with semi-active suspension.

Acknowledgment

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