

Estimator Design Using Four Relative Displacement Sensors for Semi-Active Suspension System

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Abstract— Estimator designs only using relative displacement between sprung mass and unsprung mass for semi-active suspension system are investigated. Modified sensitivity method controls semi-active suspension directly using relative velocity and MR damper model without calculating absolute velocity and parameter estimator. The comfort evaluation parameters about the driver are presented and the mathematical models of skyhook and modified skyhook control suspension are simulated in Simulink. The comparison results of suggests that modified skyhook control could significantly improve riding comfort and provide more comfortable working conditions for passenger, which can be widely applied on car in the future.

Index Terms— Semi-Active Suspension, Skyhook Control, Estimator, MR damper, Full-vehicle Model, Relative Displacement Sensor

I. INTRODUCTION

In recent years there has been a significant growth in the design of suspension systems [1-16]. This is associated with new solutions for innovative suspensions applied in vehicles, transport platforms, agricultural tractors, operator cabins and seats, etc. A suspension system performs two main functions. The first is to give support to the structure at the accepted static deflection. In addition, suspension systems in a car have to ensure good contact between the wheels and the surface of the road [4-6]. Thus the conventional suspensions are usually very stiff. The second function is associated with the vibroisolation of car bodies, operator seats, and platforms used to transport equipment sensitive to vibration. In this case, suspension should reduce vibration due to the external disturbances to the maximum possible degree. 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. The design of suspension is always a compromise between handling and comfort. Several studies have shown that this conflict can be eased by using controlled suspension systems instead of passive ones. Especially semi-active dampers are wide spread in industrial application because of their performance increase compared to passive suspensions and their lower energy consumption compared to active systems [1].

Methods for controlling semi-active suspension systems reach back to the works of Karnopp who has proposed the skyhook algorithm in order to improve the ride comfort [3-5]. Several extensions of skyhook control have been suggested in the past decadesdecades and especially the combination of a skyhook-based controller with a groundhook algorithm promises additional improvements on ride safety [7-8].

The skyhook control is simple and provides a relatively good vibration suppression capability compared to other algorithms. The implementation of the skyhook control requires information on the relative velocity of the sprung and unsprung masses as well as the absolute velocity of the sprung mass [8]. To achieve this, at least seven acceleration sensors, if a full vehicle model is used, are needed. Because only four relative displacement sensors are used in this paper, three ways estimation schemes for the absolute velocity of the sprung mass is pursued. The skyhook control itself is good at suppressing the vibration of the sprung mass, however it deteriorates the vibration of the unsprung mass [9]. From this point of view, a weighted skyhook control for the purpose of improving the vibration suppression capability of the unsprung mass near 10 Hz is investigated. The contributions of this paper are the following. A new control strategy using four relative displacement sensors is pursued. The conventional skyhook control is modified in such a way that the control characteristics at 10Hz is improved.

This paper is structured as follows. In Section 2, a full-vehicle model for the semi-active suspension system using MR damper is introduced. In Section 3, three ways estimator and weighted skyhook control algorithms are introduced. In Section 4, the performances of three ways estimator and a controlled semi-active suspension system are evaluated with Simulink. Conclusions are given in Section 5.

II. MODELING OF A FULL-VEHICLE MODEL

When we desire to improve the ride quality in various roads, not sacrificing the handling performance of the vehicle with semi-active dampers, the damping force range needs to be widened and the response time of the dampers needs to become small. Hence, for most advanced vehicles, the use of a continuous damping control (CDC) damper is the current trend. A discrete

damping control damper equipped with a step motor (or even a hydraulic damper equipped with a solenoid valve) is capable of controlling 1-4 Hz road disturbances. Hence, on a road with high frequency components, the vibration suppression capability of the traditional hydraulic CDC damper is limited. One of the notable features in this paper is the use of relative displacement sensors, in contrast with the conventional method which uses two acceleration sensors on the sprung and unsprung masses, respectively. The measurement principle of relative displacement is to detect the phase difference between the input and output sinusoidal signals, which changes with the movement of the piston. With this approach, a relative displacement up to the size of micrometers can be measured. Most of all, the sensing mechanism is resistant to external impacts and noises.

In this paper, however, because the absolute velocities at four corner points of the sprung mass (vehicle body) have to be estimated, a full-vehicle model is considered. If the vertical velocity at the center of the vehicle body (\dot{v}_s) and the roll and pitch rates ($\dot{\phi}$, $\dot{\theta}$) are known, the velocities at the four corners points can be easily calculated using the geometry of the vehicle body. The accuracy of models can be enhanced by extending from a quarter-vehicle model to a full-vehicle model as given below. The improvement comes from the fact that the roll and pitch motions of the vehicle can be incorporated and the mass moment of inertia of the vehicle can be included.

$$\begin{aligned}
m_s \ddot{z}_s &= -k_{sf}(z_{sfl} - z_{ufl}) - k_{sf}(z_{sfr} - z_{ufr}) \\
&\quad - k_{sr}(z_{srl} - z_{url}) - k_{sr}(z_{srr} - z_{urr}) \\
&\quad - f_{fl} - f_{fr} - f_{rl} - f_{rr}, \\
I_\theta \ddot{\theta} &= k_{sf} l_f (z_{sfl} - z_{ufl}) + k_{sf} l_f (z_{sfr} - z_{ufr}) \\
&\quad - k_{sr} l_r (z_{srl} - z_{url}) - k_{sr} l_r (z_{srr} - z_{urr}) \\
&\quad + l_f f_{fl} + l_f f_{fr} - l_r f_{rl} - l_r f_{rr}, \\
I_\phi \ddot{\phi} &= -k_{sf} t_f (z_{sfl} - z_{ufl}) + k_{sf} t_f (z_{sfr} - z_{ufr}) \\
&\quad - k_{sr} t_r (z_{srl} - z_{url}) + k_{sr} t_r (z_{srr} - z_{urr}) \\
&\quad - t_f f_{fl} + t_f f_{fr} - t_r f_{rl} + t_r f_{rr}.
\end{aligned} \tag{1}$$

The subscript appearing in (1): s and us indicate sprung and unsprung mass; fl, fr, rl, and rr are front left, front right, rear left, and rear right, respectively. A full-vehicle model depicts the schematic of a full-vehicle model and the control strategy using the relative displacement of the sprung and unsprung masses. Let f be the force generated at the damper, which is the control force. Then the equations of motion are

$$\begin{aligned}
m_{uf} \ddot{z}_{ufl} &= k_{sf}(z_{sfl} - z_{ufl}) - k_{uf}(z_{ufl} - w_{fl}) + f_{fl}, \\
m_{uf} \ddot{z}_{ufr} &= k_{sf}(z_{sfr} - z_{ufr}) - k_{uf}(z_{ufr} - w_{fr}) + f_{fr}, \\
m_{ur} \ddot{z}_{url} &= k_{sr}(z_{srl} - z_{url}) - k_{ur}(z_{url} - w_{rl}) + f_{rl}, \\
m_{ur} \ddot{z}_{urr} &= k_{sr}(z_{srr} - z_{urr}) - k_{ur}(z_{urr} - w_{rr}) + f_{rr}.
\end{aligned} \tag{2}$$

$$\begin{aligned}
z_{sfl} &= z_s + l_f \theta + t_f \phi, \\
z_{sfr} &= z_s + l_f \theta - t_f \phi, \\
z_{srl} &= z_s - l_r \theta + t_r \phi, \\
z_{srr} &= z_s - l_r \theta - t_r \phi.
\end{aligned} \tag{3}$$

III. MODIFIED SKYHOOK CONTROL

In this section, upon the use of relative displacement sensors, the skyhook control approach introduced by Karnopp et al. (1974) for a quarter model is applied to a full-vehicle model by a modification of the law to improve the handling performance. The skyhook approach means inserting a fictitious damper between the sprung mass and an imaginary sky (a fixed frame) so that the vibration of the sprung mass is damped out through the imaginary damper. The skyhook control is used to reduce the vertical acceleration of the sprung mass for the ride quality improvement. The skyhook control law is concise and is known adequate for a 8-bit ECU due to its computational simplicity, while the performance is almost equivalent to a state feedback control. However, the implementation of a skyhook control requires the velocity of the sprung mass as well as the relative velocity between the sprung and unsprung masses. For completeness of the paper, the skyhook control law is reviewed first as follows.

$$\begin{aligned}
f_j &= -c_{s,j} \cdot (\dot{z}_{s,j} - \dot{z}_{u,j}) \quad , \quad \text{if } \dot{z}_{s,j}(\dot{z}_{s,j} - \dot{z}_{u,j}) > 0 \quad , \\
f_j &= \min_I(c_{s,j}(I)) \cdot (\dot{z}_{s,j} - \dot{z}_{u,j}) \quad , \quad \text{if } \dot{z}_{s,j}(\dot{z}_{s,j} - \dot{z}_{u,j}) < 0 \quad , \tag{4}
\end{aligned}$$

where $c_{sky,j} = 2\zeta_{critical} \sqrt{(m_s/4) k_{s,j}}$, $j \in \{fl, fr, rl, rr\}$ (i.e., j refers to the four dampers), $c_{sky,j}$ is the desired skyhook damping coefficient at the j -th damper, $\zeta_{critical}$ is the critical damping ratio, and I is the current input to the damper.

The above skyhook control law is very effective in reducing the resonance peak of the sprung mass, but it deteriorates the handling performance by reducing the contacting force between the tire and the road surface at the resonance frequency of the unsprung mass. In order to overcome this, the skyhook control is modified as follows: We first filter the signals via a band-pass filter of 8-12 Hz, which is the resonance frequency band of the unsprung mass, and changes the skyhook damping coefficient $c_{sky,j}$ in relation to the magnitude of the filtered signal, and then the damping force of the MR dampers is increased when the road disturbance is given in the range of the unsprung mass resonance frequency. The relative velocities between the sprung mass and four unsprung masses are obtained by differentiating the relative displacement signals that were processed with a band pass filter as follows.

$$\dot{z}_{s,j} - \dot{z}_{u,j} = s \cdot \frac{40\omega_b s}{s^2 + 40\omega_b + \omega_b^2} (z_{s,j} - z_{u,j}) \quad , \tag{5}$$

where $j \in \{fl, fr, rl, rr\}$ and ω_b is a design parameter which characterizes the band (in our case, $\omega_b = 10$ Hz is used).

$$c_{\text{mod},j} = 2\zeta_{\text{critical}}\sqrt{(m_s/4)k_{s,j} + \beta \cdot (\dot{z}_{s,j} - \dot{z}_{u,j})} \quad (6)$$

where β is a controller parameter, which needs to be tuned for a given vehicle (in our case, $\beta = 1.2$ is used). The idea in this modification is to make the MR damper harder by adding the second term in (6) that focuses on the resonance frequency of the unsprung mass. Because $(\dot{z}_{s,j} - \dot{z}_{u,j})$ is obtained through the band pass filter in (5), the road disturbance at 10 Hz affects the second term in (6). The role of β is to provide a freedom to the driver to choose the ride quality or the handling performance of the given suspension system. Finally, the following modified skyhook law is proposed.

$$f_j = -c_{\text{mod},j} \cdot \dot{z}_{s,j} \quad , \quad \text{if} \quad \dot{z}_{s,j}(\dot{z}_{s,j} - \dot{z}_{u,j}) > 0 \quad , \\ f_j = \min(c_{s,j}(I)) \cdot \dot{z}_{s,j} \quad , \quad \text{if} \quad \dot{z}_{s,j}(\dot{z}_{s,j} - \dot{z}_{u,j}) < 0 \quad .$$

Since $\dot{z}_{s,j}$ in (8) can not be measured, two estimation schemes for it are discussed in the next section.

IV. FULL-VEHICLE MODEL ESTIMATOR

The first approach is to use the equations of motion of the full-vehicle model developed in the previous section.

$\dot{z}_{s,j}$ can be calculated using the geometry of the vehicle body (i.e., t_f, t_r, l_f, l_r), if \dot{z}_s , $\dot{\phi}$, and $\dot{\theta}$ are known. Therefore, motivated by (1), the use of the following equations is proposed.

$$\ddot{z}_s \cong \frac{1}{m_s} [-k_{sf}(z_{sfl} - z_{ufl}) - k_{sf}(z_{sfr} - z_{ufr}) \\ - k_{sr}(z_{srl} - z_{url}) - k_{sr}(z_{srr} - z_{urr}) \\ - f_{fl} - f_{fr} - f_{rl} - f_{rr}] \\ \ddot{\theta} \cong \frac{1}{I_\theta} [k_{sf}l_f(z_{sfl} - z_{ufl}) + k_{sf}l_f(z_{sfr} - z_{ufr}) \\ - k_{sr}l_r(z_{srl} - z_{url}) - k_{sr}l_r(z_{srr} - z_{urr}) \\ + l_f f_{fl} + l_f f_{fr} - l_r f_{rl} - l_r f_{rr}] \quad (8) \\ \ddot{\phi} \cong \frac{1}{I_\phi} [-k_{sf}t_f(z_{sfl} - z_{ufl}) + k_{sf}t_f(z_{sfr} - z_{ufr}) \\ - k_{sr}t_r(z_{srl} - z_{url}) + k_{sr}t_r(z_{srr} - z_{urr}) \\ - t_f f_{fl} + t_f f_{fr} - t_r f_{rl} + t_r f_{rr}]$$

where \ddot{z} , $\ddot{\theta}$, $\ddot{\phi}$ denote the estimates of \ddot{z} , $\ddot{\theta}$, $\ddot{\phi}$ respectively. In (8), because the real damping force f_j of the MR dampers can not be measured, the command signal f_j in (7) is replaced under the assumption that the command

signal f_j is well generated through the real dampers.

With (8), the estimates of the velocities (\dot{z}_s , $\dot{\phi}$, and $\dot{\theta}$) are now obtained by integrating (8) and passing through a high pass filter as follows.

$$\dot{z}_s = \frac{s}{s+1.256} \cdot \frac{1}{s} \cdot \ddot{z}_s \quad , \\ \dot{\theta} = \frac{s}{s+1.256} \cdot \frac{1}{s} \cdot \ddot{\theta} \quad , \\ \dot{\phi} = \frac{s}{s+1.256} \cdot \frac{1}{s} \cdot \ddot{\phi} \quad , \quad (9)$$

where a high pass filter of 0.2 Hz cutoff frequency is used. This high pass filter removes the DC offset component (integration constant). Fig. 1 shows the comparison of the unsprung mass displacements. Finally, using (3), the absolute velocities of the four corner points are obtained as follows. Fig. 2 shows the comparison of the sprung mass velocities. The Simulink simulation diagram is given in Fig. 3.

$$\dot{z}_{s,fl} = \dot{z}_s + l_f \dot{\theta} + t_f \dot{\phi} \quad , \\ \dot{z}_{s,fr} = \dot{z}_s + l_f \dot{\theta} - t_f \dot{\phi} \quad , \\ \dot{z}_{s,rl} = \dot{z}_s - l_f \dot{\theta} + t_f \dot{\phi} \quad , \\ \dot{z}_{s,rr} = \dot{z}_s - l_f \dot{\theta} - t_f \dot{\phi} \quad . \quad (11)$$

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. To find the best compromise for the trade-off between the ride comfort and road holding effects, a modified skyhook control was developed (i.e., the two independent variables to be specified by designers can be adaptively tuned to precisely achieve the type of desired response). The simulation result show that a semi-active suspension system with the proposed control strategy is able to improve the ride comfort as shown in Fig. 4 and the road holding as shown in Fig. 5, significantly compared with the conventional skyhook control. Around 6 Hz frequency ranges, the performance of conventional skyhook is slightly better than the modified one. It is predicted as one limitation of the estimator design.

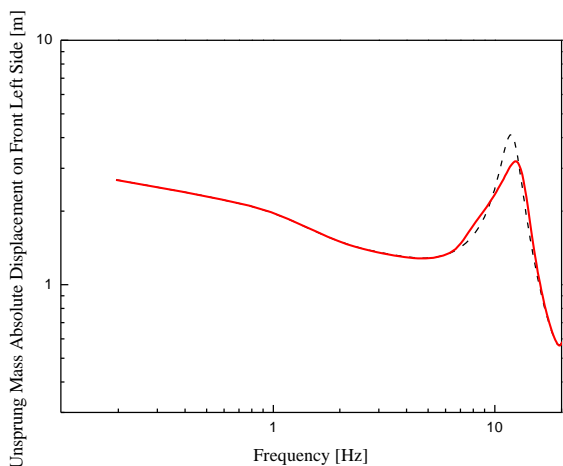


Fig. 1 Comparison of the unsprung mass displacements: Skyhook control (dashed line) vs. modified skyhook control (solid line)

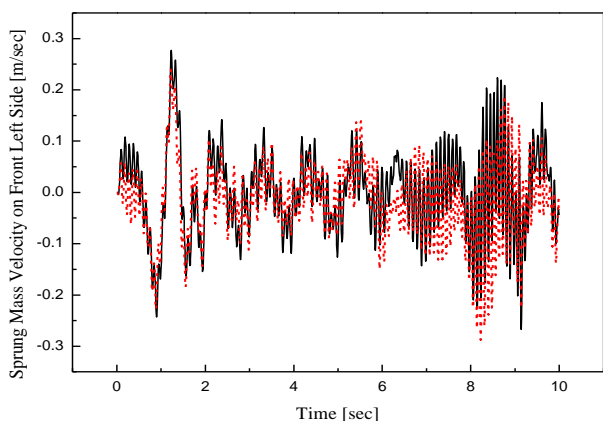


Fig. 2 Comparison of the sprung mass velocities: The estimated velocity using the equations of motion (dot line) vs. the real one (solid line)

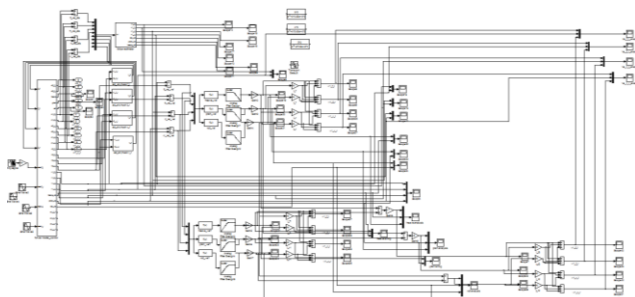


Fig. 3 Simulink Diagram for Simulation

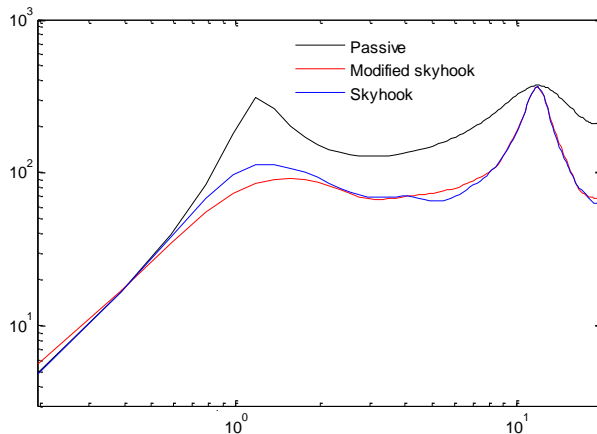


Fig. 4 Frequency response of sprung mass absolute acceleration (front left side): Passive (black line), skyhook control (blue line), and modified skyhook control (red line)

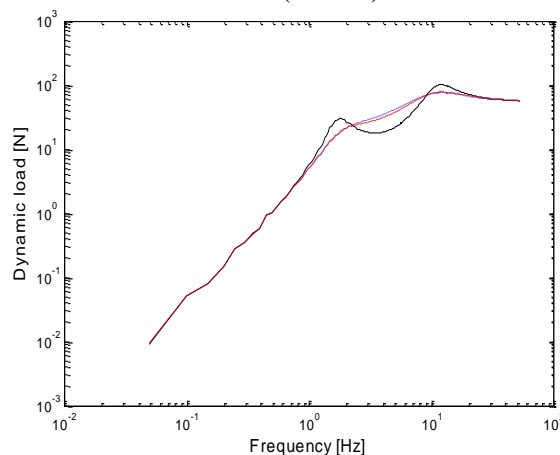


Fig. 5 Frequency responses of $dynamic\ load(s) / z_r(s)$ (front left side): passive (black line), skyhook control (blue line), and modified skyhook control (red line)

V. CONCLUSION

For improving the handling performance of the vehicle near the resonance frequency of the unsprung mass, the traditional skyhook coefficient was modified resulting in the addition of a relative velocity term to the critical damping ratio. For estimating the absolute velocities of the four corner points of the sprung mass, a full-vehicle model that describes the heave, roll, and pitch motions of the vehicle body was utilized. The simulation result show that the sprung mass acceleration (i.e. indicate the ride comfort) and the dynamic load (i.e. indicate the road holding) especially at the resonance frequencies were significantly improve.

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