

Improvement of Ride Comfort by Unsprung Negative Skyhook Damper Control Using In-Wheel Motors

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ABSTRACT

Vehicles equipped with in-wheel motors (IWMs) are capable of independent control of the driving force at each wheel. These vehicles can also control the motion of the sprung mass by driving force distribution using the suspension reaction force generated by IWM drive. However, one disadvantage of IWMs is an increase in unsprung mass. This has the effect of increasing vibrations in the 4 to 8 Hz range, which is reported to be uncomfortable to vehicle occupants, thereby reducing ride comfort. This research aimed to improve ride comfort through driving force control. Skyhook damper control is a typical ride comfort control method. Although this control is generally capable of reducing vibration around the resonance frequency of the sprung mass, it also has the trade-off effect of worsening vibration in the targeted mid-frequency 4 to 8 Hz range. This research aimed to improve mid-frequency vibration by identifying the cause of this adverse effect through the equations of motion. As a result, a method was derived by analysis that reduced vibration over a wide mid-frequency range by a control that applies unsprung vertical velocity in the direction that enhances that velocity (i.e., a negatively signed skyhook damper control called unsprung negative skyhook damper control). This control was then incorporated into a vehicle equipped with IWMs and the improvement effect on ride comfort was verified.

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INTRODUCTION

Vehicles equipped with in-wheel motors (IWMs) are capable of independently controlling the driving force at each wheel. Therefore, in addition to yaw control, IWMs can also control the roll and pitch of the sprung mass [1]. However, IWMs may also have an adverse effect on ride comfort due to the increased unsprung mass. Figure 1 compares simulated power spectral density (PSD) results for vertical acceleration of a vehicle body driven at 60 km/h (37 mph) over a rough road with and without a 25% increase in unsprung mass (same total vehicle weight) using a 4-wheel full-vehicle model. The increase in the unsprung mass caused vibration to increase, particularly around the 3 to 9 Hz range. Since vibration in the 4 to 8 Hz range is reported to be uncomfortable to vehicle occupants [2], this confirms that increasing the unsprung mass is not desirable for maintaining ride comfort. Therefore, this research examined a method of reducing uncomfortable vibrations using IWM driving force control.

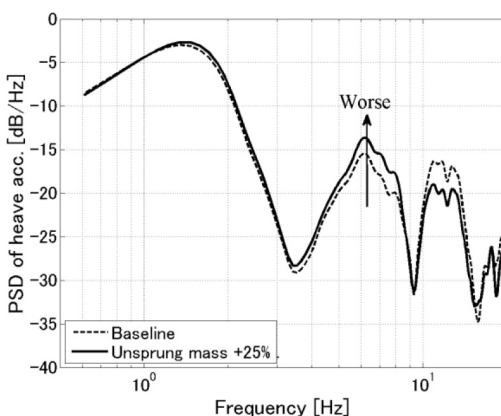


Figure 1. Effect of increased unsprung mass on vertical acceleration of vehicle body (simulation).

RIDE COMFORT CONTROLS TARGETING ROAD SURFACE DISTURBANCES

Conventional Controls

Before being applied to vehicles, active suspension research originally began as a branch of rail vehicle engineering [3]. Skyhook damper control is typical well-established ride comfort control method [4] that has been applied to vehicle suspension controls [5], although various suspension control concepts have been proposed [6]. First, to confirm the effect of skyhook damper control, a simulation was carried out using a full-vehicle model equipped with IWMs, which was driven over a rough road at 60 km/h. Skyhook damper control was used to distribute the driving force. Figure 2 shows the PSD results for the vertical acceleration of the vehicle. Although the control was clearly effective in reducing vibration around the sprung mass resonance frequency, it had a slightly adverse effect on the 4 to 8 Hz range targeted in this research (vibration in this frequency range is referred to as “mid-frequency vibration” in this paper). When adopted in an actual vehicle, although the control suppressed major movements, fine hard vibrations were noticeable.

The reasons for the adverse effect of the skyhook damper control on mid-frequency vibration were examined theoretically. Figure 3 defines the quarter-car 2-degree-of-freedom (DOF) model used for this purpose. Equations (1) and (2) show the equations of motion for the sprung and unsprung mass, respectively. Here, m_1 and m_2 are the unsprung and sprung mass, k_s and c_s are the spring constant and damping factor of the suspension, and k_t is the spring constant of the tire. F_c is the acting force of the suspension due to the control, and reaction force is a positive value. z_0 , z_1 , and z_2 are the vertical displacements of the road surface, unsprung mass, and sprung mass, respectively. s is the Laplace operator, e^{-cs} in Equation (3) is delayed

by the damping factor of the skyhook damper and acts in accordance with the transfer function $D(s)$. Table 1 shows the values used for calculation with this model.

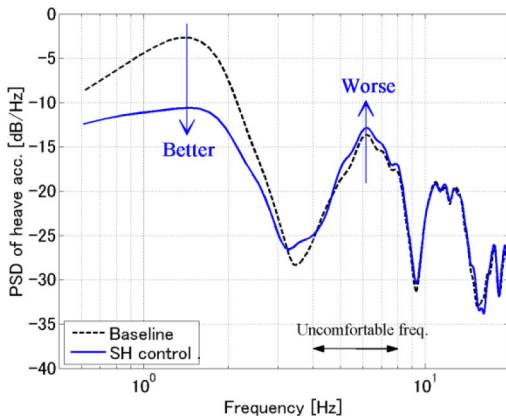


Figure 2. Effect of skyhook damper control on vertical acceleration of vehicle body (simulation).

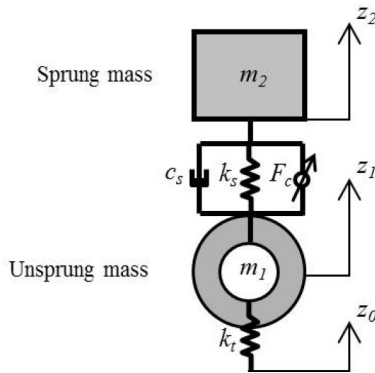


Figure 3. Quarter-car 2-DOF model.

$$m_2 z_2 s^2 = c_s(z_1 - z_2)s + k_s(z_1 - z_2) + F_c \quad (1)$$

$$m_1 z_1 s^2 = c_s(z_2 - z_1)s + k_s(z_2 - z_1) + k_t(z_0 - z_1) - F_c \quad (2)$$

$$F_c = -c_{sh} z_2 s D(s) \quad (3)$$

Table 1. Parameters for calculation.

Parameters	Symbol	Unit	Value
Sprung mass	m_2	kg	500
Unsprung mass	m_1	kg	80
Spring stiffness	k_s	N/m	30e3
Damping coefficient	c_s	N/(m/s)	2000
Tire stiffness	k_t	N/m	300e3

Figure 4 shows a Bode plot of the sprung mass acceleration that occurs in response to road surface displacement input, as calculated by the model. The first order lag shown in Equation (4) was substituted for $D(s)$, 5.3 Hz was used as the cutoff frequency f_c , and

40% of c_s was substituted into the control damping factor c_{sh} . In addition, the input amplitude was multiplied by $1/f$, which is the reciprocal of the input frequency f . As a result, applying the skyhook damper control reduced vibration around the sprung mass resonance frequency and slightly worsened mid-frequency vibration. These results reproduce the same trends calculated using the full-vehicle model. However, when there was no lag, mid-frequency vibration did not worsen. In other words, the worsening of mid-frequency vibration is not an inherent characteristic of IWMs, but a characteristic of skyhook damper control containing lag.

$$D(s) = \frac{2\pi f_c}{s + 2\pi f_c} \quad (4)$$

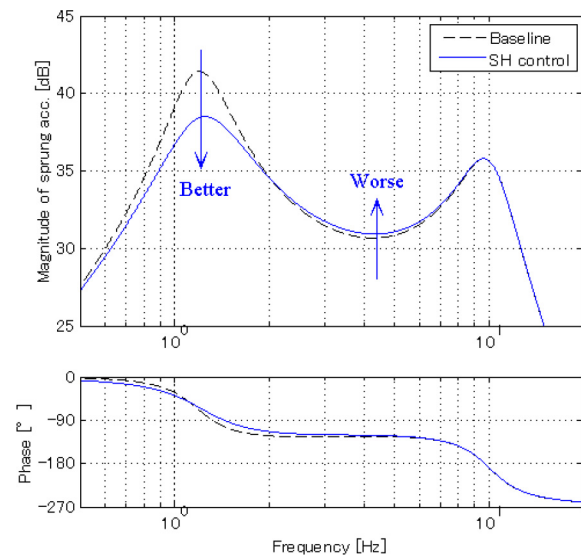


Figure 4. Effect of skyhook damper control on vertical acceleration of sprung mass.

The mechanism by which the control worsened mid-frequency vibration was analyzed by calculating sprung mass displacement z_2 with respect to unsprung mass displacement z_1 rather than road surface displacement z_0 . Equation (5) was used for this purpose.

$$\frac{z_2}{z_1} = \frac{c_s s + k_s}{m_2 s^2 + (c_s + c_{sh} D(s))s + k_s} \quad (5)$$

This equation can be used to identify the following points. The suspension damping factor c_s is contained in the velocity terms of both the numerator and denominator of the transfer function shown in Equation (5). However, the skyhook coefficient c_{sh} is only included in the denominator. In other words, although convergence improves as the suspension damping factor increases, it also has the trade-off effect of worsening input from the unsprung mass. In contrast, it also explains that a skyhook damper enhances the suppression of the sprung mass without worsening the input from the unsprung mass. However, since a control lag is applied to the c_{sh} term, it is not impossible to increase the equivalent damping factor of the denominator with respect to fast frequency inputs, thereby affecting the sprung mass control.

Proposed Control

First, the research examined ways of eliminating the control lag $D(s)$ in the denominator. The first order term of s in the numerator and denominator both contain the suspension damping factor c_s . In skyhook damper control theory, the coefficient of the first order term of s in the denominator is larger than the coefficient of the first order term in the numerator. However, to eliminate the lag in the denominator while maintaining this magnitude relationship, c_s can be reduced by software control rather than through the hardware. In other words, as shown in Equation (6), in addition to the sprung mass skyhook control force shown in the first term on the right side, a control force should be applied that is proportional to the suspension stroke velocity of the second term on the right side. It should be noted that the sign of the control force F_c is positive in the direction of the rebound of the suspension control actuator (i.e., the extension direction).

$$F_c = [-c_{sh}z_2s - c_{sc}(z_1 - z_2)s]D(s) \quad (6)$$

In this equation, although c_{sc} is the amount of decrease in suspension damping caused by the control, Equation (7) can be obtained from Equation (6) by applying a value equal to the sprung mass skyhook damper damping factor c_{sh} . By substituting this into Equation (5), the sprung mass displacement with respect to the unsprung mass displacement can be expressed using Equation (8)

$$F_c = -c_{sh}z_1sD(s) \quad (7)$$

$$\frac{z_2}{z_1} = \frac{(c_s - c_{sh}D(s))s + k_s}{m_2s^2 + c_s s + k_s} \quad (8)$$

Based on Equation (8), the control lag of the denominator is eliminated and the first order term of s in the denominator satisfies the condition that it should be larger than the first order term of the numerator. In addition, as shown in Equation (7), one distinguishing feature of the proposed control is that the command value does not require information about the sprung motion and that the control is performed using only the vertical velocity of the unsprung mass. The direction of this control force is opposite to the skyhook damper and enhances the absolute vertical velocity of the unsprung mass. Therefore, this control is called the unsprung negative skyhook damper control. This control can be explained in physical terms as follows. When a wheel travels over a bump, the unsprung mass possesses velocity in the upward direction. This control raises the unsprung mass and, simultaneously, the reaction force lowers the sprung mass. This has the effect of holding down the sprung mass, which should inherently rise up, while restricting an increase in the contact load. When the wheel moves down after the bump, the opposite action occurs and the same sprung mass motion suppression effect occurs. Ultimately, this control acts to actively move the tires to follow the road surface profile.

The effect of the conventional and proposed controls was confirmed and compared using a 2-DOF model simulation. The control lag $D(s)$ was set as the same first order lag as described above. Figure 5 shows

the results. Although the conventional sprung skyhook damper control increased the mid-frequency vibration that is a source of discomfort for vehicle occupants, the proposed control enabled a substantial improvement. Figures 6 and 7 compare the unsprung vertical acceleration and contact load of the controls. Unsprung vertical acceleration appears to worsen around the unsprung resonance frequency. Although fluctuations in contact load also worsen around the unsprung resonance frequency, a substantial reduction is achieved around 3 Hz.

These simulation results suggested a major improvement effect. However, Equation (8) contains a lag term in the numerator. Since the reason why this does not have an adverse impact on sprung acceleration could not be explained, this research then focused on analyzing this phenomenon.

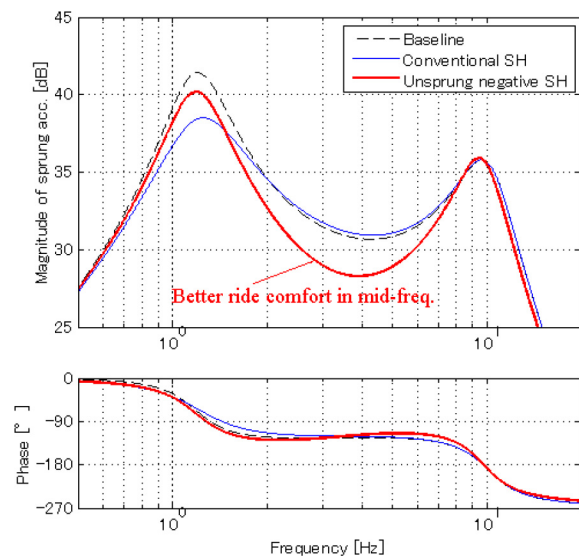


Figure 5. Effect of unsprung negative skyhook damper control on sprung mass vertical acceleration.

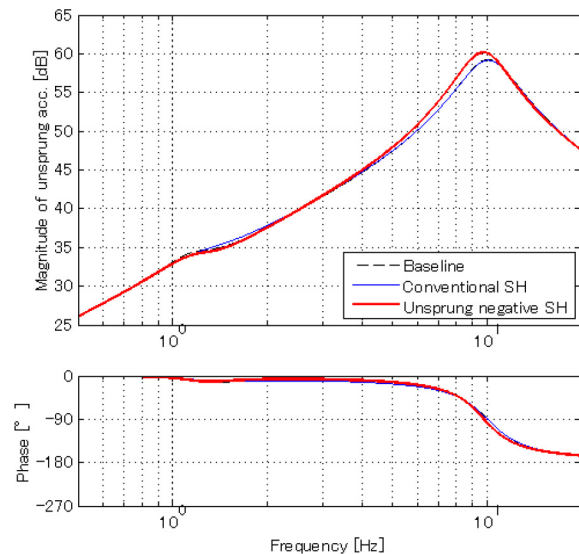


Figure 6. Effect of unsprung negative skyhook damper control on unsprung mass vertical acceleration.