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Reduction of Vertical Vibration for Improvement of Ride Comfort Using In-Wheel Motors

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Abstract

In recent years, consideration for the environment is essential for automobile development. Electric vehicle (EV) is an effective approach for reducing greenhouse gas emissions. EV can be controlled precisely with the quick response of motor. Moreover, using the in-wheel motor (IWM) can be performed for several types of motion controls. However, use of the IWM increased unsprung mass, and it brings negative effects of ride comfort. In this paper, the vibration suppression control which was conventionally done with the active suspension is performed by the anti-dive force of the suspension generated by IWM driving. We confirm how changing suspension parameters affects the ride comfort. We then suggest that the proposed method which is only used sprung information for reduction of vertical vibration.

Keywords: electric drive, control system, vehicle performance.

1 INTRODUCTION

Nowadays, consideration for environment is essential for automobile development, Ecocar such as hybrid vehicle (HV) and EV is an effective technique for reducing greenhouse gas emissions. In addition, EV has the following advantages [1].

- The torque response of motor is 10-100 times faster than Internal combustion engine vehicle.
- The generated torque of motor can be accurately measured by the current flowing in the motor.
- It is possible to disperse motors and drive each wheel independently.

These advantages are usefull in motion control. EVs currently on the market are on-board motor system. However, our laboratory expects electric vehicles equipped with IWM as next generation vehicles. Using IWM makes it possible to perform more precise and complicated control than ever before and it is possible to remove low frequency resonance by shortening the drive shaft. But, the use of the IWM increased unsprung mass, and it brings negative effects of the ride comfort, particularly around the 2 to 12 Hz range, and the 4∼8 Hz is regarded as an uncomfortable frequency range (”ride comfort zone” in this paper) for vehicle occupants. It has been reported that when the vehicle passes through a step on the road surface, the vertical vibration increases and the ride comfort deteriorates [2],[3]. Reduction of vertical vibration with an active suspension has been studied [4]∼[9]. However, this method requires the use of actuators. In this case, it is necessary to attach the actuator, so the cost and weight are increased and the application is limited to some luxury cars. On the other hand, IWM can generate positive and negative anti-dive forces, and by taking advantage of their characteristics, posture and ride comfort can be improved [10]∼[12]. The anti-dive force is a vertical force generated by the mechanism of the suspension.

In this paper, we describe improvement of ride comfort using IWM with only sprung information. Chapter 2 describes experimental vehicles and their models. Chapter 3 describes the suspension characteristics. Section 4 describes the conventional control method , Chapter 5 describes the proposed control method. Chapter 6 describes conclusion.

2 EXPERIMENTAL VEHICLE AND MODELING

2.1 Experimental vehicle

Fig. 1 [13] shows the experimental vehicle used in this study. This experimental vehicle is equipped with IWM on the rear wheel. The structure of IWM is shown in Fig. 2 [13]. Acceleration sensors are attached on the sprung and unsprung parts of each wheel in the experimental vehicle. The control input is determined from the information obtained from the acceleration sensor of each wheel.

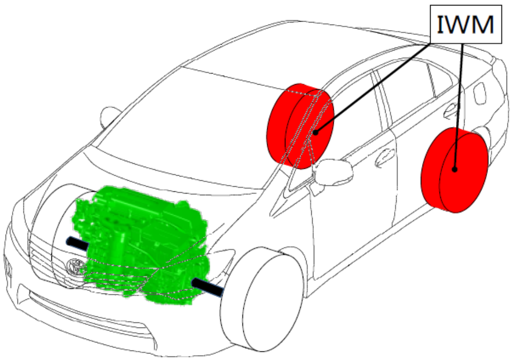

Figure 1: Experimental vehicle with the IWM units [13].

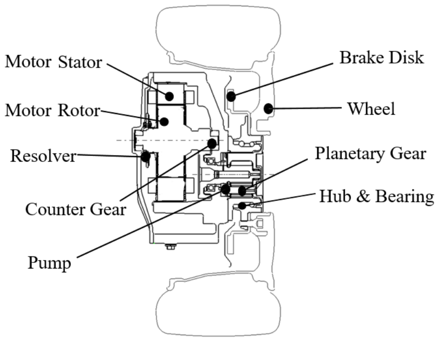

Figure 2: IWM unit [13].

2.2 Modeling

Fig. 3 and Fig. 4 show plant model and the quarter car model. In Fig. 3, m_b and m_u are the sprung mass (body mass) and unsprung mass , k_z and c_z are the suspension stiffness coefficient and damping coefficient, and k_t is the tire stiffness coefficient. z_0, z_u, z_b represent the vertical displacement of the road surface, the unsprung mass, and the sprung mass, respectively. F_{cz} and F_{kz} are the damping part and the stiffness part of the suspension, respectively , and F_t is the elastic force of the tire. F_z is the controlled variable generated by the anti-dive force. The dynamics equations of the quarter car model is represented by

$$m_b \ddot{z_b} = F_{cz} + F_{kz} + F_z \tag{1}$$

$$m_u \ddot{z_u} = -F_{cz} - F_{kz} + F_t - F_z \tag{2}$$

$$F_{cz} = c_z(\dot{z_u} - \dot{z_b}) \tag{3}$$

$$F_{kz} = k_z(z_u - z_b) \tag{4}$$

$$F_t = k_t(z_0 - z_u) \tag{5}$$

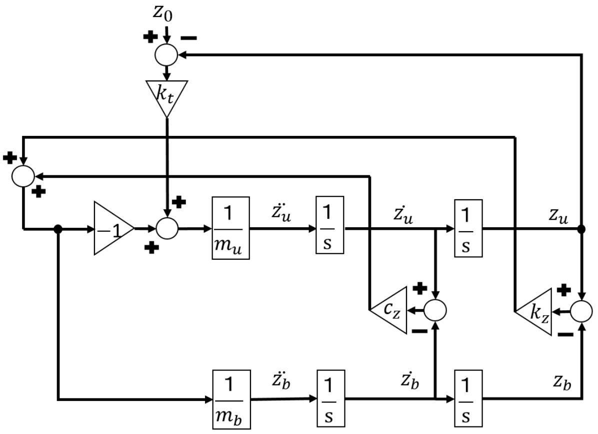

Figure 3: Block diagram of Plant model.

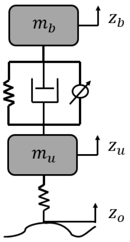

Figure 4: Quarter-car model.

2.3 Anti-dive force

In this section, we describe positive and negative anti-dive force. Fig. 5 shows the suspension geometry due to the driving force of each wheel. From Fig. 5, the positive and negative anti-dive forces are determined by the instantaneous center of rotation angle formed by the line connecting the point of application to the instantaneous center of rotation of the suspension and the horizontal plane. The IWM method possesses a larger instantaneous rotation center angle than the on-board motor car, so the anti-dive force becomes large. A negative anti-dive force acts on the front wheel and a positive anti-dive force acts on the rear wheel. The positive and negative anti-dive forces of each wheel are represented by

$$F_{zfj} = -F_{xfj} tan\phi_f \tag{6}$$

$$F_{zrj} = F_{xrj} tan\phi_r \tag{7}$$

where, j = l, r, and ϕ_f, ϕ_r are the instantaneous rotation center angle of the front and rear wheels.

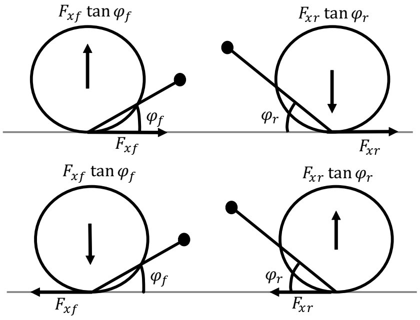

Figure 5: Positive and Negative anti-dive force.

3 ANALYSIS ON SUSPENSION CHARACTERISTICS

3.1 Characteristics of damping part of suspension

Fig. 6 shows the ride comfort characteristics in the vertical direction when the damping coefficient of the suspension varies. By setting the damping coefficient of the suspension to a small value, good characteristics can be obtained in the 2∼8 Hz. However, the sprung mass resonance around 1 Hz and the unsprung resonance around 10 Hz tend to deteriorate. When the damping coefficient is a large value, it shows opposite characteristics. The physical meaning is that the dynamic change of the car body is large for vehicles with low damping. However, it realizes a low ride comfort on the road surface. On the contrary, in a vehicle equipped with a high damping, the movement of the vehicle body becomes small and the stability improves, but the road surface disturbance becomes easy to feel.

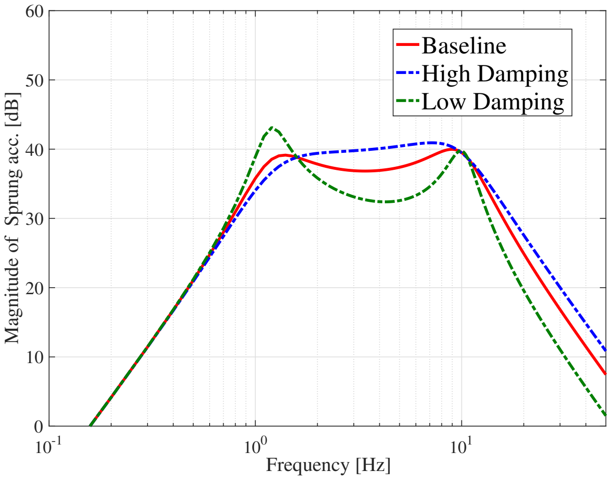

Figure 6: Effecct of changing suspension damping on sprung mass vertical acceleration.

3.2 Characteristics of stiffness part of suspension

Fig. 7 shows the ride comfort characteristics in the vertical direction when the stiffness coefficient varies. When the stiffness coefficient is small, the characteristics near the sprung resonance frequency are improved. The physical meaning is that when the stiffness coefficient is large, the sprung resonance frequency moves upward. Then, the gain characteristic of the ride comfort zone deteriorates and it becomes a hard ride feel.

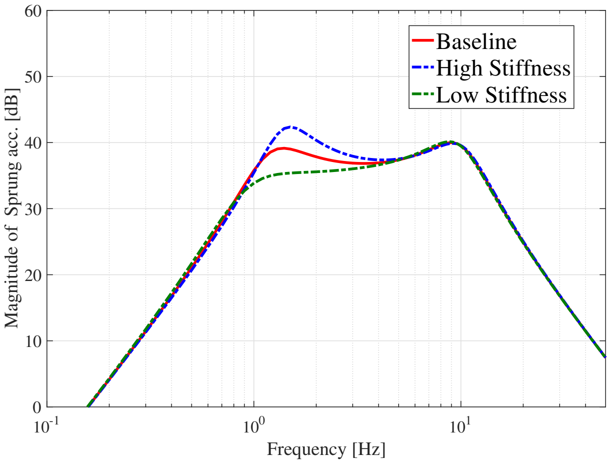

Figure 7: Effecct of changing suspension stiffness on sprung mass vertical acceleration.

4 CONVENTIONAL CONTROL METHOD

4.1 Skyhook damper control

Generally speaking, increasing the suspension damping coefficient improves the convergence, but also leads to contradiction that the input from the unsprung mass increases. The skyhook damper control which adds damping only to the sprung mass has the effect to improve the convergence on the sprung without changing the input from the unsprung mass. This control method has a simple structure and is easy to be applied to actual vehicles. This control adopts $F_z = -c_{sv}\dot{z}_b$ in (1) and uses only the sprung velocity. It is effective in reducing sprung resonance, but it is difficult to improve the characteristics of the ride comfort zone.

4.2 Unsprung negative skyhook damper control

The unsprung negative skyhook damper control requires information on the sprung and controls only with information of unsprung mass. This control adopts $F_z = -c_{uv}\dot{z}_u$ in (1) and uses only the unsprung velocity which can greatly improve the ride comfort zone. The physical meaning is to positively move the unsprung part against the road surface so that the movement of the vehicle body does not become large.

5 PROPOSED CONTROL METHOD

Unsprung negative skyhook damper control is a simple theory that uses only unsprung information. However, attaching the acceleration sensor to the unsprung mass has a problem in terms of cost and wiring difficulty. We analyzed the characteristics of the damper component and the spring component of the suspension and found the characteristics related to the improvement of the ride comfort zone. We predicted that the ride comfort zone can be improved by sprung acceleration feedback including the damper component and the spring component of the suspension. In the following, we describe the sprung acceleration feedback, and analyze the results of actual vehicle test and simulation.

5.1 Sprung acceleration feedback control

Sprung acceleration feedback control uses sprung acceleration and substitutes $F_z = -c_{sa}\ddot{z}_b$ into (1). $\ddot{z}_b$ and F_z are represented by

$$\ddot{z}_b = \frac{c_z(\dot{z}_u - \dot{z}_b)}{m_b + c_{sa}} + \frac{k_z(z_u - z_b)}{m_b + c_{sa}} \tag{8}$$

$$F_z = -\frac{c_{sa}c_z(\dot{z}_u - \dot{z}_b)}{m_b + c_{sa}} - \frac{c_{sa}k_z(z_u - z_b)}{m_b + c_{sa}} \tag{9}$$

The first and second terms of (9) represent low damping absorber and low stiffness, respectively. From Fig. 6, the low damping absorber has improved ride comfort zone, but the gain characteristic at sprung resonance is deteriorate. The low stiffness characteristic of Fig. 7 is compensated for the deterioration caused by the low damping absorber. We could show that the sprung acceleration feedback control has similar characteristics to the unsprung negative skyhook damper control, and possesses the advantage that it does not use a sensor under the unsprung mass.

5.2 Simulation

Fig. 8 shows the block diagram of the proposed method. Fig. 9 shows the sprung acceleration feedback control and the Bode diagram of the proposed method on sprung mass vertical acceleration. The proposed method consists of skyhook damper control and sprung acceleration feedback control. The controlled variable is $F_z = -c_{sa}\ddot{z}_b - c_{sv}\dot{z}_b$.

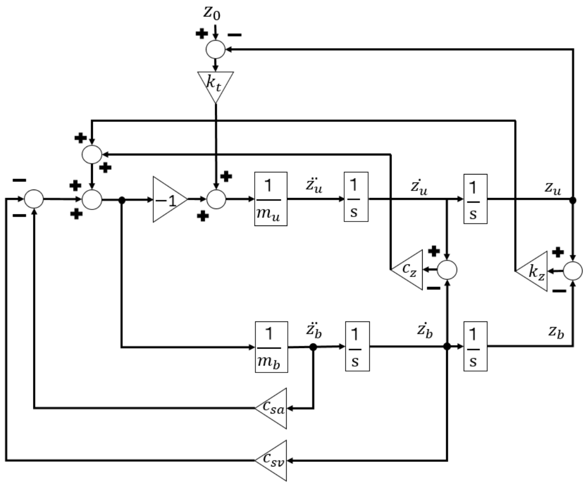

Figure 8: Block diagram of proposed method.

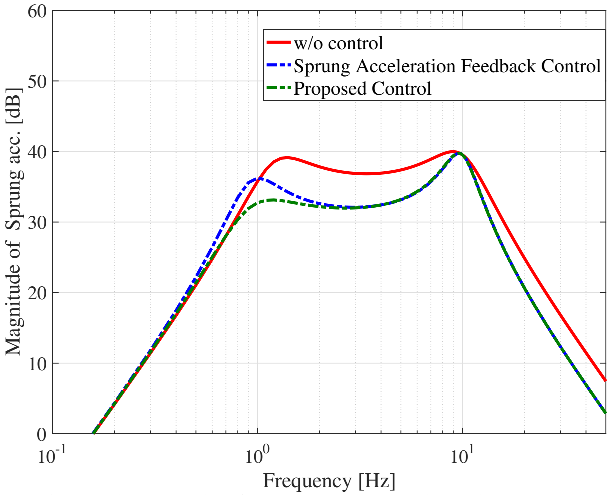

Figure 9: Simulation result of proposed control method on sprung mass vertical acceleration.

It is known that reduction in sprung resonance leads to deterioration of steering stability. The proposed method is effective to improve ride comfort. However, steering stability became worse. Substitute $F_z = -c_{sa}\ddot{z}_b - c_{sv}\dot{z}_b$ into (1) to make it dimensionless, where $c_{sa} = c_{sv}$ and $k_z = k_z^*(1 + c_{sa})$. The transfer function of the sprung mass displacement z_b to the unsprung displacement input z_u is represented by

$$\frac{z_b}{z_u} = \frac{\frac{1}{1+c_{sa}}c_z s + \frac{1}{1+c_{sa}}k_z}{m_b s^2 + \frac{1+c_{sv}}{1+c_{sa}}c_z s + \frac{1}{1+c_{sa}}k_z} \tag{10}$$

$$\frac{z_b}{z_u} = \frac{\frac{1}{1+c_{sa}}c_z s + k_z^*}{m_b s^2 + c_z s + k_z^*} \tag{11}$$

Fig. 10 shows the Bode diagram of proposed control method with the compensation for steering stability on sprung mass vertical acceleration. From (11), it is possible to carry out the unsprung negative skyhook damper control with the only sprung mass information without deteriorating the steering stability. However, it is difficult to change the characteristics of the mechanical suspension. Therefore, a combination of skyhook control and sprung acceleration feedback control is used as a proposed method.

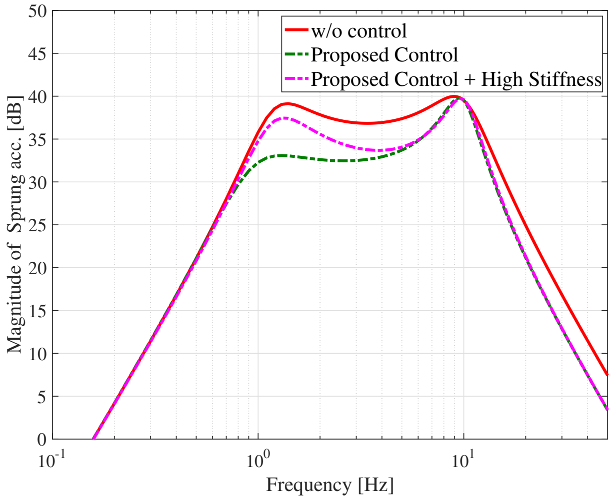

Figure 10: Simulation results of proposed control method with the compensation for steering stability on sprung mass vertical acceleration.

5.3 Experiment

Fig. 11 shows the results of power spectral density (PSD) analysis when the experimental vehicle pass through a random road at a speed of 80 km/h. The ride comfort zone is improved by sprung acceleration feedback control, but proposed control method is getting worse at the unsprung resonance frequency.

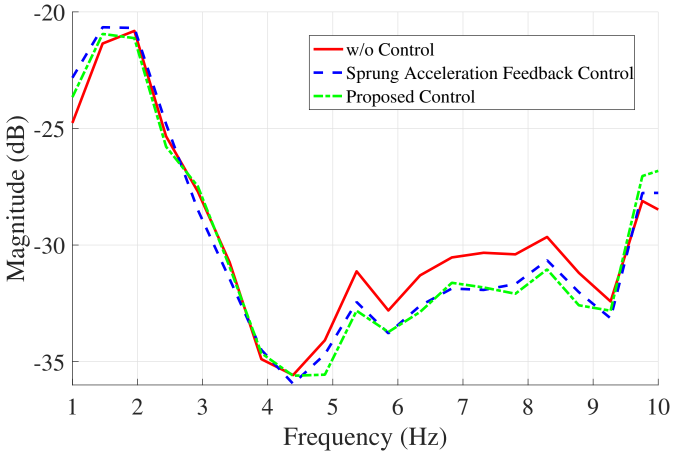

Figure 11: Experimental result of proposed method on sprung mass vertical acceleration.

We describe the reason why the gain characteristics of the experimental result deteriorate at the unsprung mass resonance frequency. In Fig. 9, time lag is not taken into consideration, but is taken in the actual vehicle test. The time lag occurs due to the dead time of CAN communication. Fig. 12 shows the simulation results of proposed control method considering time delay on sprung mass vertical acceleration. A first order lag of 15 ms was used as a time delay. The controlled variable is $F_z = -c_{sa}\ddot{z}_b D(s) - c_{sv}\dot{z}_b D(s)$, and D (s) is the first order lag. From Fig. 12, it can be noticed that the gain characteristic of the sprung mass vertical acceleration deteriorates at the unsprung resonance frequency due to time delay.

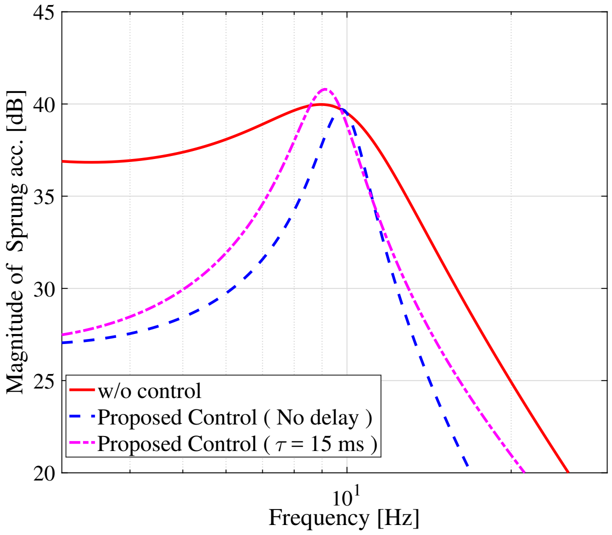

Figure 12: Simulation results of proposed control method considering time delay on sprung mass vertical acceleration.

6 CONCLUSION

In this paper, We improved the ride comfort by using the control that takes advantage of the characteristics of IWM. First, we describe the mechanical features of the suspension, and then we found that it is possible to improve the ride comfort zone with low damping absorbers and low stiffness. The physical meaning of the sprung acceleration feedback control is improvement of stability by increasing the sprung mass. The mathematical meaning is composed of low damping absorber term and low stiffness term. We could show that the sprung acceleration feedback control has similar characteristics to the unsprung negative skyhook damper control, and there is the advantage that it is able to improve the ride comfort zone with only sprung information.

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