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Development of Vehicle Pitch Vibration Reduction Control Using a Band-Stop Filter

Yuma Matsumoto [1)]

1) Toyota Motor Corporation
1 Toyota-cho, Toyota, Aichi, 471-8572, Japan (E-mail: yuma_matsumoto@mail.toyota.co.jp)

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One of the attractions of electric vehicles is the sensation of acceleration. It has been reported that this sensation is enhanced by the generation of high longitudinal jerks and the suppression of longitudinal head motion through pitch motion.Since electric vehicles have a low center of gravity and a large vertical distance between the center of gravity and the occupants' heads, the fore-and-aft vibration of the occupants' heads caused by vehicle pitch tends to be significant. Therefore, suppressing pitch behavior is critical for achieving a strong sense of acceleration.

A control method had been proposed that suppresses vehicle pitch behavior by applying a band-stop filter to the total drive force of the vehicle body to attenuate the pitch resonance frequency contained within the drive force(Fig.1)(Table1).

However, this approach had the drawback of reducing acceleration jerk as pitch behavior was suppressed. In this study, We focused on the relationship between the parameters of the conventional control system and its frequency characteristics, noting that a broad frequency band around the vehicle's pitch resonance frequency was also being attenuated.To address this, we aimed to achieve both a low pitch rate and high acceleration jerk by improving the design flexibility of the band-stop filter and setting optimal frequency characteristics (Eq.1)(Table2).

We compared the control effects through simulations and vehicle tests(Fig.2), demonstrating that it is possible to achieve both a low pitch rate and a high acceleration jerk compared to the conventional control. Furthermore, by clarifying the trends in control performance in response to changes in control parameters through comprehensive simulations, we made it possible to set control parameters to achieve arbitrary performance characteristics. However, since the metrics used in this study—pitch rate and acceleration jerk—are considered insufficient to fully reflect the effectiveness of this control, the selection of appropriate metrics and parameter design methods utilizing them are the next items for consideration.

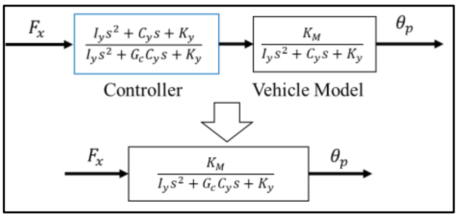

Fig.1 Effect of conventional controller

Table1 Variable descriptions (Conventional)

F_x	Driving Force[N]
θ_p	Pitch Angle[rad]
I_y	Pitch Moment of Inertia [kg m2]
C_y	Pitch Damping [N s/rad]
K_y	Pitch Stiffness [N/rad]
K_M	Gain from F_x to θ_p[rad/N]
G_c	Damping Gain[-]

$$C_p(s) = \frac{s^2 + 2\frac{G_w}{G_d}\zeta(G_f\omega_n)s + (G_f\omega_n)^2}{s^2 + 2G_w\zeta(G_f\omega_n)s + (G_f\omega_n)^2} \qquad (1)$$

Table2 Variable descriptions (Proposed)

ζ	Pitch Damping Coefficient [-]
ω_n	Pitch Resonant Frequency [rad/s]
K_a	Gain from F_x to θ_p[rad/N]
G_d	Damping Gain[-]
G_w	Bandwidth Gain[-]
G_f	Frequency Gain[-]

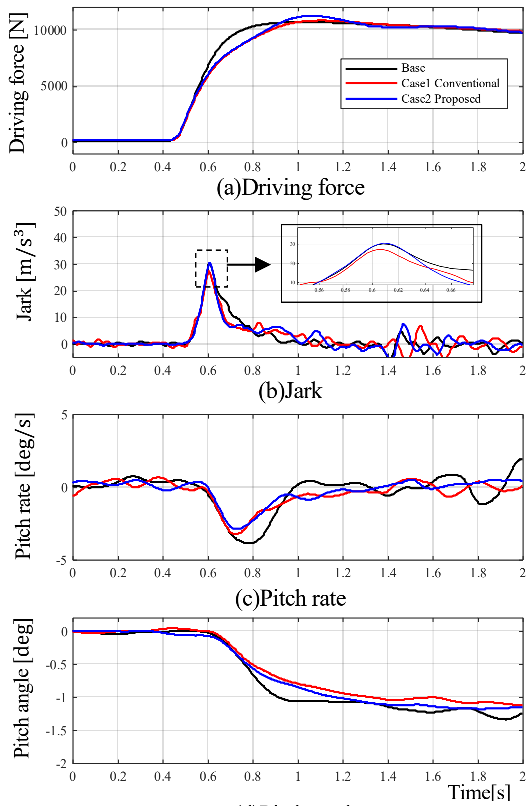

Fig.2 Measurement result of real vehicle tests

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