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HYBRID ROLL CONTROL USING ELECTRIC ARC SYSTEM CONSIDERING LIMITED BANDWIDTH OF ACTUATING MODULE

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ABSTRACT–This paper presents the design of an active roll control system for a ground vehicle and an experimental study using an devised electric-actuating roll control system. Based on a three degree of freedom linear vehicle model, the controller is designed using lateral acceleration and rollrate feedback. In order to investigate the feasibility of an active control system, experimental work is carried out using a hardware-in-the-loop (Hil) setup which has been constructed by the devised electric-actuating system and the full vehicle model including tire characteristics. The performance is evaluated by an experiment using the Hil setup with limited bandwidth. Finally, in order to enhance the control performance in the transient region, a hybrid control strategy is proposed and evaluated.

KEY WORDS : Active roll control, Hybrid control, Hil setup, Variable damper

NOMENCLATURES

A_o, B_o, L_o, E : System matrices in state-space eqs.
I_x, I_z : Roll, yaw mass moment of inertia
K_a, K_d : Controller gain
K_f, K_r : Cornering stiffness of tire
K_θ : Roll bar stiffness
M : Total vehicle mass
M_d : Desired roll moment
M_s : Mass of car body
V : Forward velocity
a_y : Lateral acceleration
h_s : Distance from roll center to CG point
l_f, l_r : Wheel base
v : Lateral velocity
α_f, α_r : Coefficient of roll angle effect
β : Slip angle
δ_f : Tire steer angle
γ : Yaw angular velocity
ϕ : Roll angle
p : Roll angular velocity

SUBSCRIPTS

f, r : front, rear

1. INTRODUCTION

In a steering maneuver of a vehicle, vertical loads on the outer track of tires increase while those on the inner track decrease. This is lateral load transfer resulting from body roll motion. Roll stability is one of the dynamic factors that needs to be considered in a ground vehicle. This is a problem for automobiles, but is also applicable to railroad vehicles with a high center of gravity. To improve the roll characteristics of a car, the customary approach is to increase the roll stiffness using a stabilizer bar. However, this method is restricted because it affects the ride comfort with respect to high frequency isolation induced by road excitation.

In order to enhance vehicle performance in an active manner, advanced suspension systems, such as active suspension, have been widely analyzed in the literature over the years (Hrovat, 1993; Yoshimura, 1999; Li, 1999).

At present one of the most efficient means for further development is the roll control option combined with a semi-active suspension system (Karnopp, 1991). This system performs the isolation of the driver from uneven roadway noise, road holding on irregular road surfaces using a variable damper system, and safe turning through steering using the active roll control (ARC) system. This provides many of the benefits of a fully active system with much reduced cost and power consumption. Unfortunately, most studies have focused on the active or

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semi-active suspension system and there have been few studies of the ARC system or the combined system.

Lin (1996) performed a theoretical study of active roll reduction in heavy vehicles. Their system used an anti-roll bar equipped with a hydraulic linear actuator which provided the necessary torque to counteract the roll moment of the car body. The lateral acceleration feedback and LQR control scheme were used, based on the linear vehicle model. Ross-martin (1997) performed a simulation study of the ARC system for passenger cars with a hydraulic rotary actuator using lateral acceleration feedback control. Kim (2002) performed a theoretical and experimental study using ARC hardware-in-the-loop setup.

As is the case with any vehicle equipped by ARC system, the ARC system based on a hydraulically actuated system is complicated to implement, with many hydraulic components; pump, control valve, oil tank, tubes, etc, and has some disadvantages related to contamination caused by oil leakage and maintenance of components. Moreover, the controller designed in the framework of a complicated control scheme may have computational burden in practical implementation.

In this paper, the controller design is carried out based on the three degree of freedom vehicle model including lateral/roll dynamics related to steering maneuvers. To investigate dynamic characteristics, the active roll controller is designed by lateral acceleration and body rollrate feedback considering the practical implementation, which is an efficient method such as skyhook control in the framework of active suspension system. In order to evaluate the control performance, a prototype passenger car electric-actuation system is devised, that is comprised of an active anti-roll bar with electric motor and ball screw type actuator. This system can improve the high frequency isolation with high compliance characteristics during inoperative periods as described in section 4.1. Using the hardware-in-the-loop setup including the prototype roll control system and full vehicle model with nonlinear tire characteristics, the control performance is investigated. Finally, considering the limited bandwidth of the actuating system, in order to enhance the control performance both in a steady-state and in a transient region, a hybrid control method is proposed and evaluated by simulation based on the investigation of variable damping system to roll motion control.

2. MATHEMATICAL VEHICLE MODEL

In developing the active controller, it is not desirable to use the complex vehicle model because of sampling time and implementation of the control system. In this paper, the linear vehicle model is used for the design of a controller. This three degree of freedom model (Kim, 2002) includes the yaw and roll dynamics of a car, related to driver steering maneuvers, traveling on a road surface at a constant speed V with the front tire steering angle δ_f. The vehicle is assumed to be symmetrical in the vertical plane, the tire characteristics and road conditions for the left and right tires are the same.

Under the assumption that the lateral tire forces are linear functions with slip angles β_f, β_r of front and rear tire, the governing equations including lateral, yaw and roll motion are written as

$$MV(\dot{\beta}+\gamma)+M_s h_s \dot{p}=2K_f\left(\delta_f+\alpha_f\phi-\beta-\frac{l_f\gamma}{V}\right)+2K_r\left(\alpha_r\phi-\beta+\frac{l_r\gamma}{V}\right) \quad (1)$$

$$I_z\dot{\gamma}=2K_f\left(\delta_f+\alpha_f\phi-\beta-\frac{l_f\gamma}{V}\right)l_f-2K_f\left(\alpha_r\phi-\beta+\frac{l_r\gamma}{V}\right)l_r \quad (2)$$

$$I_x\dot{p}+M_s h_s V(\dot{\beta}+\gamma)=-K_\phi\phi-C_\phi p \quad (3)$$

From previous Equations (1)-(3), the state-space representation can be expressed as Equation (4) with state vector $x_p=[v\ \gamma\ \phi\ p]^T$, active roll moment M_d and steer disturbance δ_f.

$$E\dot{x}_p=A_o x_p+B_o M_d+L_o\delta_f \quad (4)$$

where,

$$A_o=\begin{bmatrix} -\frac{(K_f+K_r)}{V} & \left(-MV-\frac{K_f l_f}{V}+\frac{K_r l_r}{V}\right) & (K_f\alpha_f+K_r\alpha_r) & 0 \\ -\frac{(K_f l_f-K_r l_r)}{V} & -\frac{(K_f l_f^2-K_r l_r^2)}{V} & (K_f\alpha_f l_f+K_r\alpha_r l_r) & 0 \\ 0 & 0 & 0 & 1 \\ 0 & -M_s h_s V & -K_\phi & -C_\phi \end{bmatrix}$$

$$B_o=[0\ 0\ 0\ 1]^T,\ L_o=[K_f\ K_f l_f\ 0\ 0]^T$$

$$E=\begin{bmatrix} M & 0 & 0 & M_s h_s \\ 0 & I_z & 0 & 0 \\ 0 & 0 & 1 & 0 \\ M_s h_s & 0 & 0 & I_x \end{bmatrix}$$

3. CONTROLLER DESIGN

3.1. Lateral Acceleration Feedback Control

In order to reduce the roll response resulting from steer disturbance δ_f, a lateral acceleration signal $a_y(t)$ from a body mounted transducer is used as the main source for the roll control. Thus the desired anti-roll moment is computed like:

$$M_d(t)=K_a a_y(t) \quad (5)$$

where K_a is the controller gain.

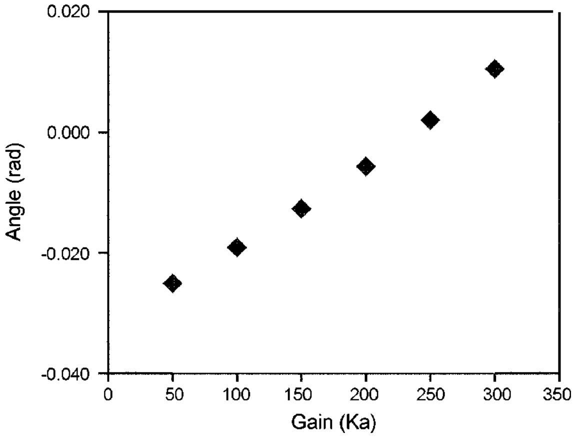

Figure 1. Comparison of steady-state roll angle in J-turn maneuver.

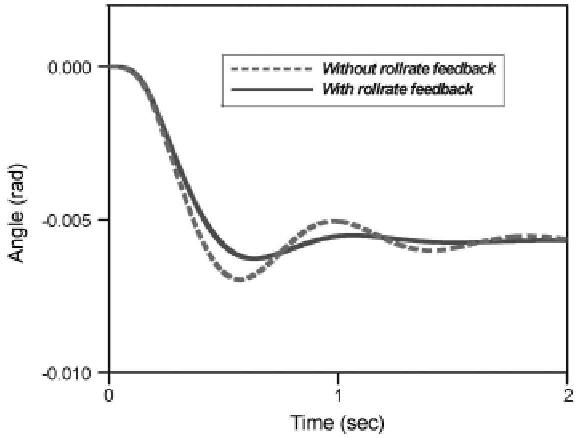

Figure 2. Comparison of body roll responses in J-turn maneuver.

This control law can mitigate the computational burden induced by using the complicated control scheme such as H_∞ control and it is efficient method considering the practical implementation such as skyhook control in the framework of active suspension system.

Figure 1 illustrates the controlled roll angle response in the steady-state region with varying gain value K_a under condition of J-turn steer maneuver with tire steer angle 3.5 deg and duration 0.2 sec. In this figure, within range of 200-250, a car body can be flat in steering operation. If K_a is over 250, body roll has a motion in the opposite direction.

3.2. Lateral Acceleration and Roll Angular Velocity Feedback Control

From the Equation (5), in order to enhance the transient roll behavior, roll angular velocity feedback term $\dot{\phi}(t)$ is added to the controller. The use of derivative feedback - roll angular velocity - can help improve the roll mode damping factor (Shuttlewood, 1992). Thus, the controller computes the desired anti-roll moment in this way:

$$M_d(t)=K_a a_y(t)+K_d\dot{\phi}(t) \tag{6}$$

where K_d is the controller gain of derivative feedback term.

Each constant controller gain K_a, K_d in Equation (6) is determined by simulation based on the preceding vehicle dynamic model and tuned by experiment using Hil setup in section 4.

Figure 2 illustrates the effect of the rollrate feedback term in a transient region with controller gains of K_a=200, K_d = –1000. This figure shows that it can reduce the fluctuation in the transient response rather than using a lateral acceleration signal only under condition of previous J-turn maneuver.

4. PROTOTYPE ARC SYSTEM

4.1. Configuration of ARC System

The active roll control systems can be constructed with a linear/rotary actuator and anti-roll bar (Lin, 1996; Darling, 1992). When the anti-roll mechanism is implemented with a linear actuator connecting the roll bar ends to the suspension tie rods, the desired roll moment is described as the following equation :

$$M_d=G_r K_\theta \theta_d \tag{7}$$

where G_r is the roll bar lever ratio, K_θ is the torsional stiffness of roll bar, and θ_d is the desired torsional angle.

If the actuating system dynamics were considered as a second order system, a closed loop system can be expressed as Figure 3.

The transfer function $T(s)$ between desired torque T_d and actual torque T_a through anti-roll bar can be written as Equation (8):

$$\frac{T_a(s)}{T_d(s)}=\frac{\dfrac{KK_\theta}{R_w}}{s\left(\dfrac{s^2}{w^2 n}+2\dfrac{\zeta_n}{w_n}s+1\right)+\dfrac{KK_\theta}{R_w}} \tag{8}$$

where w_n is the natural frequency, ζ_n is the damping ratio, K is the system gain and R_w is the length of the lever.

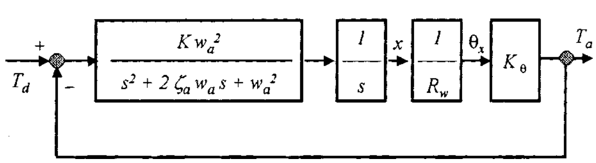

Figure 3. Block diagram of actuating system dynamics.

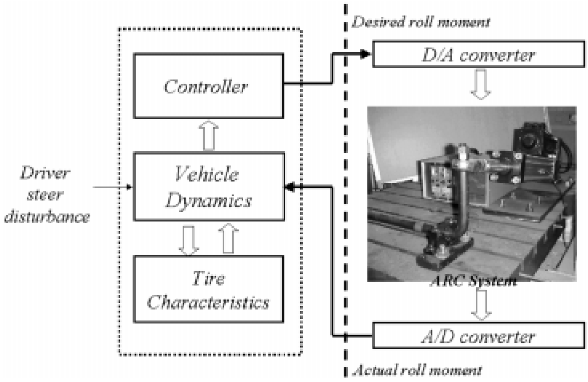

Figure 4. Schematic diagram of Hils.

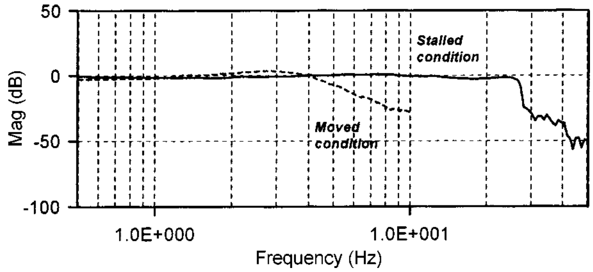

Figure 5. Frequency response of the actuating system.

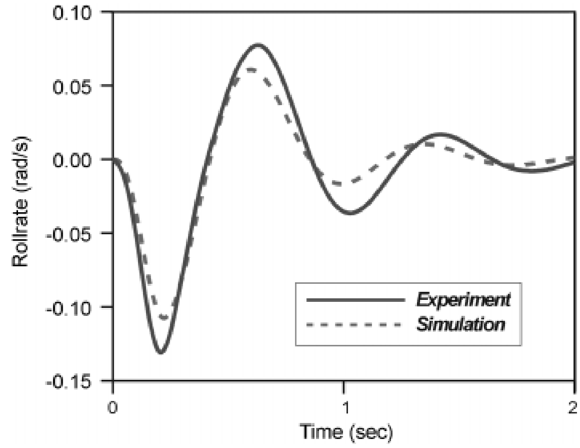

Figure 6. Comparison of roll angle responses.

In order to make the system respond faster, K_θ must be increased because the time constant of the system could be a function of the torsional stiffness K_θ. If there is too much additional roll stiffness, it will spoil the ride comfort because of the high roll excitation which is induced when a bump is met by wheels on one side of the car only (Bastow, 1993).

In this paper, to overcome the aforementioned problem, a prototype active roll control system is devised, which is composed of an electric motor, gear box and a ball screw type actuator which has low friction. This system would effectively remove the additional roll stiffness introduced by the active anti-roll bar and improve the high frequency isolation of the car body during periods when the system is inoperative.

Related to this consideration, Darling (1998) devised a hydraulic actuating system with a P-port closed proportional valve, which allows the actuator to free wheel during straight line driving. Moreover, in the case of the hydraulically actuated system, the system is complex, contamination by leakage during operation has to be considered and maintenance and changing of parts are complicated.

In order to evaluate the performance of the active roll control system to a real car with the devised ARC system, a series of experiments were performed using a Hil (hardware-in-the-loop) setup with ten degree of freedom vehicle dynamics and tire module (Kim, 2002). The Hils (Hardware-in-the-loop simulation) technique is an efficient way to realistically test dynamic vehicle behavior in a laboratory. Not considering the real-time roll moment distribution control between front and rear axle, the actuating system is considered as a single unit. The configuration of the Hils setup is illustrated in Figure 4.

4.2. Experiment

Figure 5 shows the frequency response of the devised ARC system with a ±1 V sinusoidal command signal. In this figure, the bandwidth under stalled conditions is wider than when moving, where stalled means the actuator stroke is fixed.

In J-turn maneuver with vehicle parameters in reference (Kim, 2002) at constant speed 13.9 m/sec, experimental results using the Hil setup are presented in Figure 6 compared to simulation result with bandwidth 3.3 Hz. In this figure, the resulting response was affected by the actuating system dynamics with some bandwidth.

As in the previous work (Lin, 1996), the use of a limited bandwidth actuating system causes some performance degradation in a transient region. To improve the transient response, a high bandwidth actuating system can be used. In practice, this causes an increase in cost. To improve this result, in this paper, a hybrid control strategy is presented in section 5. This is a combined control method using a continuously variable damper subjected to vibration control induced by uneven road excitation , which is based on a car equipped with ARC system as an option.

5. HYBRID CONTROL

In the previous description of section 4, the controlled responses in the transient region are affected by the bandwidth of the actuating system. In order to improve the transient responses, a feasibility of a variable damping system on active roll control is investigated. In general, a variable damper is used for vibration control induced by irregular road input.

The damping force of the variable damper is determined by a controllable damping coefficient as in the following rule.

$$C_d = \begin{cases} \dfrac{f_d}{(\dot{x}_s - \dot{x}_u)} & [f_d(\dot{x}_s - \dot{x}_u) > 0] \\ C_{\min} & [f_d(\dot{x}_s - \dot{x}_u) \le 0] \end{cases} \quad (9)$$

If $C_d > C_{\max}$, then $C_d = C_{\max}$. Where C_d is desired damping coefficient, $C_{\max}$ and $C_{\min}$ are the maximum, minimum damping coefficients of the variable damper. f_d is the desired force for suppression of vibration, $(\dot{x}_s - \dot{x}_u)$ is relative velocity between car body and wheel at each corner.

Considering the roll motion in a steering maneuver, the desired damping force at each damper on roll control can be calculated by the following relationships:

$$Md = \sum_{j=1}^{2} M_j \quad (10)$$

$$M_i = F_{li}\frac{t_i}{2} - F_{ri}\frac{t_i}{2} \quad (F_{li} + F_{ri} = 0) \quad (i = 1, 2) \quad (11)$$

In the above equations, M_d is the total desired roll moment, M_j is the moment component of subscript j with the front and rear axles, F_{li} and F_{ri} are the desired forces of the left and right sides of the axle, t_i is the track width with subscript i of left and right. The desired damping coefficient can then be determined by Equation (9).

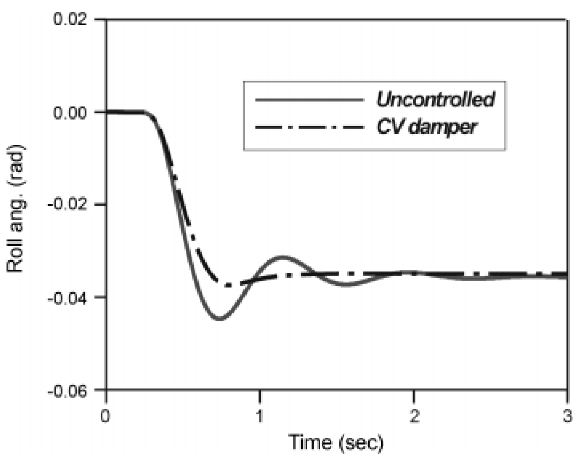

Figure 7. Roll angle responses using variable damping system.

Figure 7 illustrates the roll angle responses in J-turn maneuver at constant speed 13.9 m/sec using the variable dampers with gains $K_a = 230$, $K_d = -1000$ in Equation (6). The result of simulation using variable dampers to roll control in Figure 7, clearly shows that the roll angle response in the transient region can be smooth although there is no improvement in the steady-state region. This resulted from the operating characteristics of a semi-active type actuator which can not generate an active force.

From the previous investigation, each of the variable dampers can be controlled with an anti-roll actuating system simultaneously based on Equations (9)-(11).

Figure 8 illustrates the resulting roll responses using the hybrid control with respect to passive and ARC condition under the previous steering maneuver with an actuating system bandwidth of 3.3 Hz. In the figure, a hybrid control with a variable damping system can give a

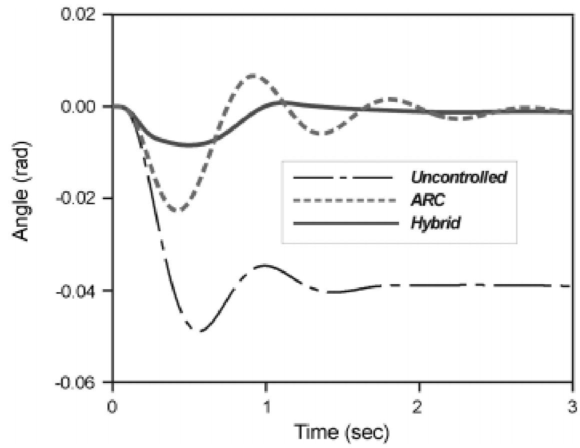

Figure 8. Comparison of roll responses in J-turn maneuver.

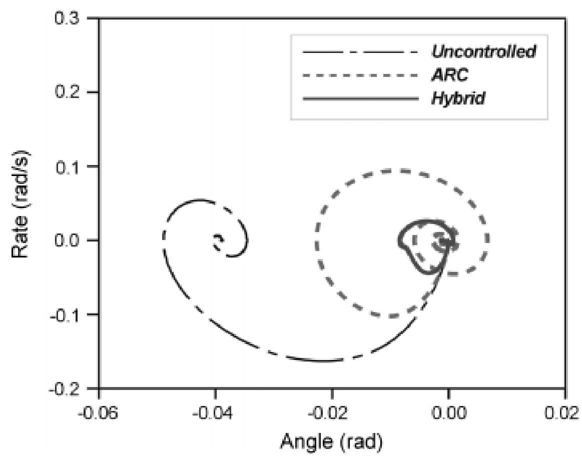

Figure 9. Comparison of roll mode phase portraits.

smoother response in the transient region. In Figure 9 of roll mode phase portrait, this shows that control performance can be achieved both in a steady-state and in a transient region with a smoother response.

6. CONCLUSION

In this work, an active roll control system represented by an electric actuating system and variable damper was proposed and the following conclusions have been met.

- In order to have a high compliance to external excitation during inoperative periods and a compact configuration, an electric actuating roll control system was devised, which could improve the undesirable dynamic response induced by additional roll stiffness.
- An active controller using lateral acceleration and rollrate feedback has been designed based on the three degree of freedom vehicle model and its performance has been compared.
- By experimental work using a hardware-in-the-loop setup and full vehicle dynamics including nonlinear tire characteristics, control performance with tuned gain values has been investigated with respect to actuator dynamics.
- Moreover, after investigation of a variable damping system for roll control, a hybrid control method has been proposed considering the limited bandwidth of electric ARC system. It has been shown that the proposed control strategy is very effective for the improvement both in a steady-state and in a transient region with a smoother response.

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