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SLIP CONTROLLER DESIGN FOR TRACTION CONTROL SYSTEM

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ABSTRACT–Two major roles of the traction control system (TCS) are to guarantee the acceleration performance and directional stability even in extreme road conditions, under which average drivers may not control the car properly. Commercial TCSs use experiential methods such as lookup table and gain-scheduling to achieve proper performance under various road and vehicle conditions. This paper proposes a new slip controller which uses the brake and the throttle actuator simultaneously. To avoid measurement problems and to get a simple structure, the brake controller and the throttle controller are designed using Lyapunov redesign method and multiple sliding mode control, respectively. Through the hybrid use of brake and throttle controllers, the vehicle is insensitive to the variation of the vehicle mass, brake gain and road condition and can achieve the required acceleration performance. The proposed method is validated with simulations based on 15 DOF passenger car model.

KEY WORDS : Traction control system, Multiple sliding mode control, Lyapunov redesign method

1. INTRODUCTION

Excessive traction torque while starting on slippery roads and slopes causes the wheels to spin. As the slip between the tire and the road increases, the abrasion of tire and drivetrain becomes severe and the traction and adhesion forces between the tire and road decrease. Consequently, it results in the reduced traction performance, directional stability and steerability. To resolve this problem, researches on the traction control system (TCS) that maintains slip ratios of driven wheels low by controlling the traction torque using various vehicle parts have been conducted.

Recently developed TCSs employ the simultaneous use of the brake controller that has fast-response time and the throttle controller that can be used in the various velocity ranges. The control methods used in Toyota Crown (Asami et al., 1989) and Lexus (Ise et al., 1990) are basically look-up table approaches applying the control input to the brake and throttle actuator according to the gain map which is the function of the reference and current velocities. These methods need much efforts and time to decide the reference values because of the complexity of the vehicle and the variation of the driving conditions. A controller design scheme that is not based on the experiential trial and error and robust to the variation of the vehicle and road conditions can reduce the necessary time and efforts drastically. The slip controller, which prevents the over-slips of the driven wheels, controls the brake pressure for the slipratio of the driven wheels to track the desired value at which the maximum traction force is generated. Researchers have used the sliding mode control to regulate the slipratio under the variation of brake gain, vehicle mass and road conditions. Tan (1992) suggested a slip controller which regulates the slipratios of the driven wheels. The controller structure was complex as the dynamics of the hydraulic actuators are omitted. Kawabe et al. (1997) suggested the sliding mode controller which uses the acceleration feedback and gain-scheduling and confirmed the controller performance experimentally. But the use of ON/OFF signals alone for the control input makes the slipratios of the driven wheels chatter. Choi et al. (1998) introduced the first order hydraulic dynamics and the PWM (pulse width modulation) input to prevent the chattering of the slipratios. But the usage of the hydraulic dynamics makes the relative degree of the brake system increase and the number of the measurement signals necessary for the sliding mode control becomes large. Some of the measurement signals are hard to measure in real situations. To avoid

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measurement problems and get a simple structure, the brake controller is designed using Lyapunov redesign method. Using the Lyapunov redesign method, the desired brake pressure is designed so that the slipratios of the driven wheels are robust to the variation of the vehicle state and the road condition and the current brake pressure chases the desired value. With this method, the relative degree of the brake system and the necessary measurement signals can be decreased. The throttle controller is based on the method proposed by Cho et al. (1989). The conversion scheme between the slip controller and the throttle controller is introduced to reduce the usage of the brake system. Feasibility of the proposed method is shown through the simulation based on the 15 DOF nonlinear vehicle model.

2. VEHICLE MODELING

Figure 1 consists of chassis, tire and brake. Each part is important to monitor the performance of TCS and is simplified as much as possible. Chassis model is of 15 DOF, where 6 DOF for sprung mass, 4 DOF for suspensions, 4 DOF for wheel rotation and 1 DOF for steering. UA tire model is employed since it shows similar characteristics to the real tire for various slipratios, slip angles and road conditions. This non-linear model is verified by real vehicle test for the impulse and the step steering.

The brake system consists of master cylinders and TCS modulators that make up the pressure source and the brake cylinder which apply the brake pressure to the wheels. The solenoid valves of brake cylinders repeat on/off motion forming the brake pressure. Dynamics of the brake cylinder pressure (P_b) can be

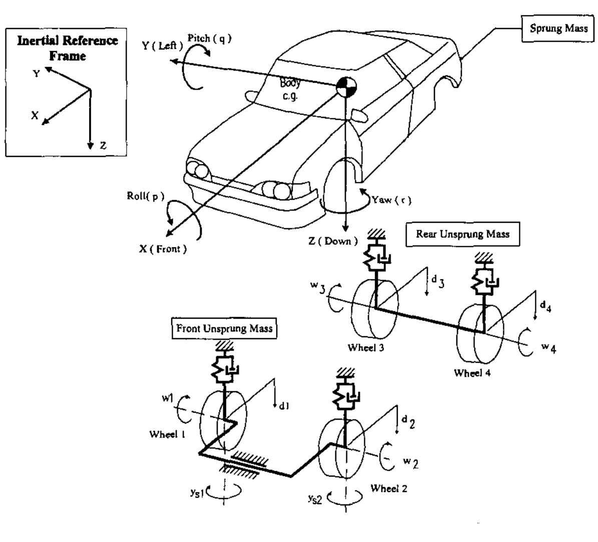


Figure 1. 15 DOF vehicle model.

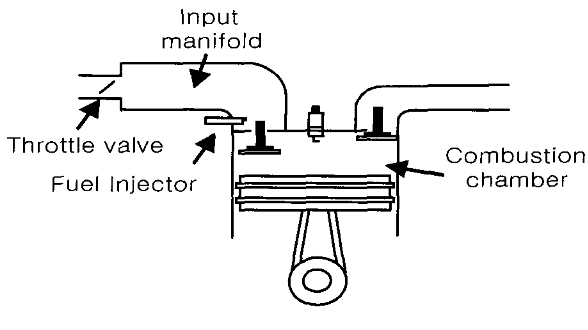


Figure 2. Engine model.

represented as follows (H. E. Merrit, 1967):

$$\frac{V_b}{\beta}\dot{P}_b = c_1 x_1\sqrt{\frac{2}{\rho}(P_m - P_b)} - c_2 x_2\sqrt{\frac{2}{\rho}P_b} \tag{1}$$

where (V_h, β, C_i, P_m, ρ) and x_i are brake cylinder volume, fluid volume coefficient, valve constant, master cylinder pressure, fluid density and solenoid distance, respectively. Master cylinder pressure (P_m) is always assumed to be ready and the dynamics of TCS modulator is omitted.

3. ENGINE & POWERTRAIN MODELING (Cho et al., 1989)

The engine system is composed of 3 states and 1 input. The state variables are engine speed (ω_e), the air mass of the input manifold (m_a) and the injection rate of the fuel ($\dot{m}_{fi}$), respectively. The input variable is the throttle angle (θ_i).

$$\tau_f \ddot{m}_{fi} + \dot{m}_{fi} = \dot{m}_{fc} = \frac{1}{K_f}\dot{m}_{ao} \tag{2}$$

$$I_e\dot{\omega}_e = T_i - T_f - T_p \tag{3}$$

$$T_i = C_T\frac{\dot{m}_{ao}(t-\Delta t)}{\omega_e(t-\Delta t)}AFI(t-\Delta t)\cdot SI \tag{4}$$

where K_f, T_i, T_f, T_p, AFI, SI, I_e, $\dot{m}_{a0}$, Δt and C_t are the optimal air-fuel ratio, engine torque, friction torque in the cylinder, pump torque of the torque converter, air-fuel influence function, spark influence function, the equivalent inertia of the cylinder, the air flow rate to the cylinder, ignition time delay and the explosion torque constant, respectively. The Equations (2), (3) and (4) describe the dynamics of the fuel flow rate, the equivalent engine inertia and the engine torque generated by the explosion. The reference injection rate of the fuel ($\dot{m}_{fc}$) and the ignition time is assumed to be selected to induce the maximum ignition torque. The torque converter that connects the engine and the transmission is experimentally determined. The transmission dynamics is omitted and only the static gear ratio is considered.

4. SLIP CONTROLLER

The conventional brake controller controls the slip of the driven wheels using ON/OFF signals for the solenoid valves ignoring the dynamics of the brake pressure because the measurement of the brake pressure is not easy. But in this case, the variation of the brake pressure is large and the slipratios of the driven wheels chatter. PWM method for the accumulation of the solenoid input can smooth the variation of the slipratios with the information of the brake pressure. With the assumption of the relatively fast solenoid input, the input signal can be assumed to be continuous. Then the brake system can be described as follows.

$$\dot{\omega}_r = \frac{1}{J_w}\left(\frac{1}{2}T_t - F_x \cdot r - K_b P_b\right) \tag{5}$$

$$\dot{V}_x = \frac{1}{m_v}\left(\sum_i F_{xi} - R_x\right) \tag{6}$$

$$\dot{P}_b = \begin{cases} \tau_1 x_1^{\max}\sqrt{P_m - P_b}\cdot D_1 \\ -\tau_2 x_2^{\max}\sqrt{P_b}\cdot D_2 \end{cases} \tag{7}$$

where ω_r, V_x, T_t, r, P_m, D_i, F_x and R_x are rotational velocity of the driven wheel, the longitudinal velocity of the vehicle, traction torque, roll radius of the wheel, master cylinder pressure and duty ratio, the traction force generated by the tire and the rolling resistance, respectively. Equations (5), (6) and (7) describes the dynamics of the driven wheel, the longitudinal dynamics of the vehicle and the dynamics of the brake pressure, respectively. The traction force is the function of the slipratio. The slipratio that describes the rate of the slip generated in the driven wheel is defined as follows:

$$\lambda = \frac{V_x}{r\cdot\omega_r} - 1 \tag{8}$$

The slipratio that generates the maximum traction force varies by the road condition and it is very difficult to find it exactly. Normally, the target of the slip control is not in generating the maximum traction force but in preventing the decrease of the lateral adhesion force. Therefore, a constant value where both of the traction force and the lateral adhesion force are in a reasonable magnitude is selected as the target slipratio. If the control objective is to make the slipratio track the constant desired value, the relative degree of the brake system becomes 2. The controller designed according to the conventional method of the sliding mode control needs the measurements of the time derivatives of the traction torque and the traction force. Normally, the measurement of the traction force and the traction torque is difficult. So the relative degree should be lowered.

4.1. Desired Brake Pressure

To reduce the relative degree of the brake system, the desired brake pressure is designed using the Lyapunov redesign method. The desired brake pressure should be designed for the slipratio to track the desired slipratio and to be robust to the variation of the vehicle states and the road conditions.

$$P_{des} = (\alpha + \beta\omega_r)\left(\lambda - (-0.2)\right) = (\alpha + \beta\omega_r)\left(0.8 - \frac{V_x}{r\cdot\omega_r}\right) \tag{9}$$

where α and β are design parameters. In this research, the desired brake pressure is defined to be proportional to the difference between the slipratio of the driven wheel and the desired value. The rotational speed of the wheel is included in the gain term to hinder the rotational speed from growing excessively. Now, the new sliding surface is designed as follows:

$$z = P_b - P_{des} \tag{10}$$

$$\dot{z} = \tau_1\sqrt{P_m - P_b}\cdot U - (\alpha + \beta\omega_r)\left(\frac{V_x\cdot\dot{\omega}_r - \omega_r\cdot\dot{V}_x}{r\cdot\omega_r^2}\right) - \beta\dot{\omega}_r\left(0.8 - \frac{V_x}{r\cdot\omega_r}\right) \tag{11}$$

Equation (11) is for the case of the brake pressure increase and is derived using Equations (1) and (9). Computing the control input that makes the derivative of the sliding surface equals zero,

$$U_{eq} = \frac{(\alpha + \beta\omega_r)\left(\frac{V_x\cdot\dot{\omega}_r - \omega_r\cdot\dot{V}_x}{r\cdot\omega_r^2}\right) + \beta\dot{\omega}_r\lambda}{\tau_1\sqrt{P_m - P_b}} \tag{12}$$

The equivalent control input is computed from the measurements of rotational speed, longitudinal velocity, their time-derivatives and the brake pressure. Note that the brake gain and the vehicle mass that vary according to the driving conditions are not used in the computation of the control input. This control method is robust to the variation of the vehicle mass and the brake gain. Normally, the rotational speed and the acceleration of the wheels are measurable. If the longitudinal velocity and the acceleration can be estimated from the rotational speed and the acceleration of the wheels, this control input can be implemented in practice.

4.2. Longitudinal Velocity Estimation

The longitudinal velocity of the vehicle can be derived using the rotational speeds of the non-driven wheels. Following equations are for the rear wheel driven car.

$$\tilde{\omega} = \omega_x \tilde{x} + \omega_y \tilde{y} + \omega_z \tilde{z} \tag{13}$$

$$OR = L_1\tilde{x} + L_2\tilde{y} - L_3\tilde{z}, \quad OL = L_1\tilde{x} - L_2\tilde{y} - L_3\tilde{z} \tag{14}$$

$$\begin{aligned}\tilde{V}_L &= \tilde{V}_O + \tilde{\omega} \times L \\ &= (V_x - L_{3L}\omega_y - \omega_z L_2)\tilde{x} + (V_y + L_{3L}\omega_x + \omega_z L_1)\tilde{y} \\ &\quad + (L_2\omega_x - \omega_y L_1)\tilde{z}\end{aligned} \tag{15}$$

$$\begin{aligned}\tilde{V}_R &= (V_x - L_{3R}\omega_y + \omega_z L_2)\tilde{x} \\ &\quad + (V_y + L_{3R}\omega_x + \omega_z L_1)\tilde{y} - (L_2\omega_x + \omega_y L_1)\tilde{z}\end{aligned} \tag{16}$$

where $\tilde{\omega}$, OR, OL and $\tilde{V}_0$ are the yaw rate of the vehicle, the distances from the mass center of the vehicle to the center of right and left non-driven wheels and the velocity of the mass center, respectively. $\tilde{V}_R$ and $\tilde{V}_L$ are the center velocity of the right and left non-driven wheels, respectively in the SAE xyz coordinates as in Figure 3. The velocities of tire centers in the traveling direction of the wheel in the transformed coordinates ($x'y'$) by steering angle (θ_s) are as follows:

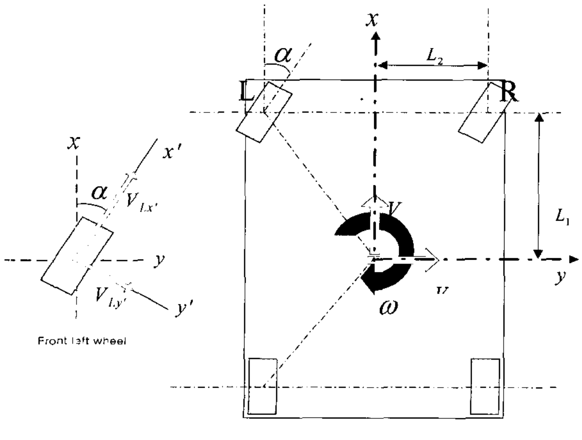


Figure 3. The center velocities of front wheels.

$$\begin{aligned}V_{Lx'} &= (V_x - L_{3L}\omega_y + L_2\omega_z)\cos\theta_s \\ &\quad + (V_y + L_{3L}\omega_x + L_1\omega_z)\sin\theta_s\end{aligned} \tag{17}$$

$$\begin{aligned}V_{Rx'} &= (V_x - L_{3R}\omega_y - L_2\omega_z)\cos\theta_s \\ &\quad + (V_y + L_{3R}\omega_x + L_1\omega_z)\sin\theta_s\end{aligned} \tag{18}$$

It is noted that the difference of two values is almost due to the yaw rate (ω_z). Arranging the average of the $V_{Lx'}$ and $V_{Rx'}$:

$$\begin{aligned}\frac{V_{Lx'} + V_{Rx'}}{2} &= \frac{r_L\omega_L + r_R\omega_R}{2} \\ &= \frac{r_s(\omega_L + \omega_R) - A(\omega_L - \omega_R)\cdot a_y}{2} \\ &= \left(V_x - \frac{L_{3R} + L_{3L}}{2}\cdot\omega_y\right)\cos\theta_s \\ &\quad + \left(V_y + \frac{L_{3R} + L_{3L}}{2}\cdot\omega_x\right)\sin\theta_s\end{aligned} \tag{19}$$

The following estimation of the longitudinal velocity can be derived.

$$\begin{aligned}V_{xe} &= \frac{r_s(\omega_L + \omega_R)}{2\cos\theta_s} \\ &= V_x - \frac{L_{3R} + L_{3L}}{2}\cdot\omega_y \\ &\quad + \left(V_y + \frac{L_{3R} + L_{3L}}{2}\cdot\omega_x\right)\frac{\sin\theta_s}{\cos\theta_s} \\ &\quad - \frac{A}{2\cos\theta_s}(\omega_R - \omega_L)a_y\end{aligned} \tag{20}$$

Considering that
$\omega_x \approx 0$, $\omega_y \approx 0$, $V_y \sin\theta_s << V_x$, $a_y \approx 0$ in the longiudinal driving, the left side of the Equation (20) can be simplified as follows:

$$V_{xe} = \frac{r_s(\omega_L + \omega_R)}{2\cos\theta_s} \cong V_x \tag{21}$$

5. THROTTLE CONTROLLER

The relative degree of the engine system that has the throttle angle as the input and the rotational speed of the wheel as the output is two. Increase of the relative degree makes the controller structure complex and the necessary measurements increase. To reduce the relative degree, multiple sliding mode is employed and the engine speed is selected as the synthetic input. Then the engine system is divided into two subsystems. The first subsystem has the engine speed as the input and the rotational speed of the wheel as the output. The second subsystem has the throttle angle as the input and the engine speed as the output. Then the relative degree of each subsystem becomes one and the number of the differentiation of the rotational speed of the wheel decreases.

The first sliding surface is selected as the difference of the slipratio of the driven wheel and the desired slipratio as follows:

$$S_1 = \lambda - (-0.2) = \frac{V_x}{r \cdot \omega_r} - 0.8 \tag{22}$$

$$\begin{aligned} \dot{S}_1 &= \frac{1}{r \cdot \omega_r^2} \left\{ \omega_r \dot{V}_x - V_x \cdot \frac{1}{J_w} \left(\frac{1}{2} T_{td} - \frac{1}{2} T_t + J_w \cdot \dot{\omega}_r \right) \right\} \\ &= -K_1 S_1 \end{aligned} \tag{23}$$

where λ, V_x, ω_r, r, J_w, J_e, T_{td} and T_t are the slipratio of the driven wheel, longitudinal velocity of the vehicle, the rotational speed of the driven wheel, the roll radius of the driven wheel, the moment of inertia of the driven wheel, the equivalent moment of inertia of the driven wheel, the desired output torque of the torque converter and the output torque of the torque converter, respectively. From the Equation (23), the desired output torque of the torque converter (T_{td}) is computed. Then the desired engine speed can be computed using the experimental formulation between the output torque and the engine speed.

The second sliding surface is selected as the difference of the engine speed and the desired engine speed.

$$S_2 = \omega_e - \omega_{ed} \tag{24}$$

$$\dot{S}_2 = \frac{1}{J_e}\left(T_{id} - T_f - T_p\right) - \dot{\omega}_{ed} = -K_2 S_2 \tag{25}$$

where ω_e, ω_{ed} and T_{id} are engine speed, the desired engine speed and the desired engine torque, respectively. From Equation (25), the desired engine torque is computed. The desired throttle angle can be derived from the desired engine torque.

6. OVERALL CONTROLLER STRUCTURE

The brake controller is activated as the slipratio of the driven wheel rises crossing the desired slipratio. As the adhesion force of the slippery road is very small, the slipratio of the driven wheel rises up to the value of -1 where the full slip happens even though the brake controller is activated. The throttle controller is activated as the slipratio of the driven wheel crosses the value of -0.45 which is selected as a design parameter.

The value K_1 used in Equation (25) is designed as a large value to decrease the throttle angle as fast as possible. When the throttle controller decrease the throttle angle and the brake controller regulates the slipratio of the driven wheel around the desired value, the brake pressure is decayed to zero. After the brake controller is deactivated, the slipratio of the driven wheel is controlled by the throttle controller only. The large K_1 should be lowered to prevent the chattering of the slipratio caused by the time delay of the throttle actuator and the engine system. Overall controller structure is shown in Figure 4.

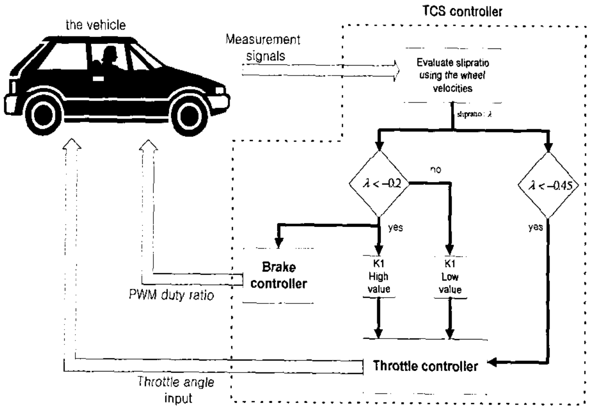
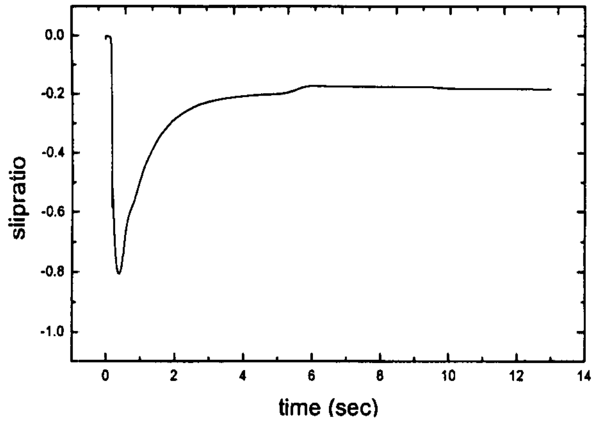


Figure 4. The overall controller structure.

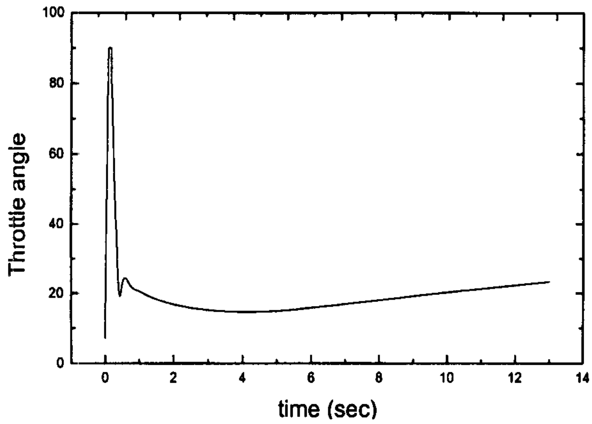
7. SIMULATION RESULTS

7.1. Start Off on the Icy Road

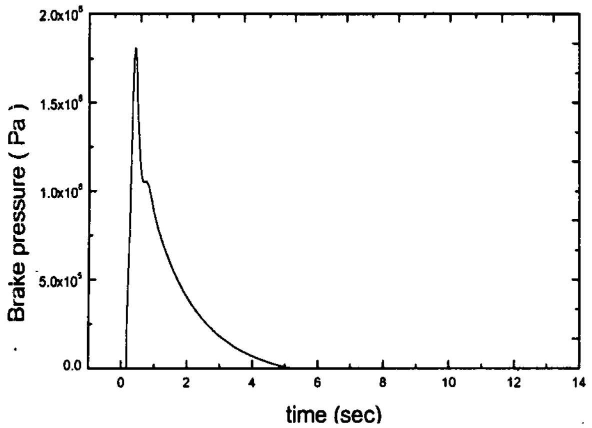
The simulation is about the start off of the vehicle on the slippery road with the driver's excessive acceleration pedal input. The road is assumed to be in the uniform condition. The simulation is conducted using the 15 degrees of freedom vehicle model. The slipratio of the driven wheels rises up to -0.8 by the excessive traction torque because of the slippery road condition as shown in Figure 3(a). As the slipratio rises over the limit, the throttle and brake actuator activates. Throttle angle is decreased to 20 degree and the slipratio tracks the desired value after 1 sec. After 4 seconds from the time of controller actuation, the brake controller is stopped as the brake pressure decrease to zero and only the throttle controller controls the slipratio.



(a) The slipratio of the driven wheel

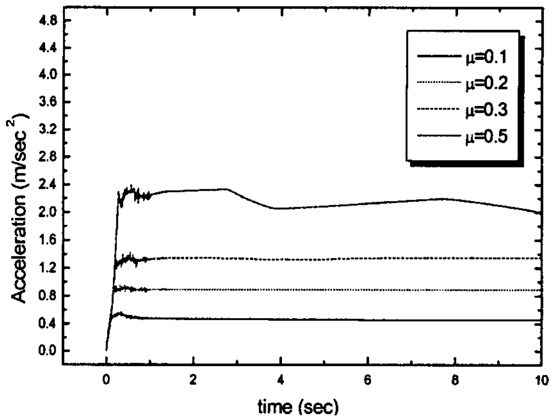


(b) The throttle angle

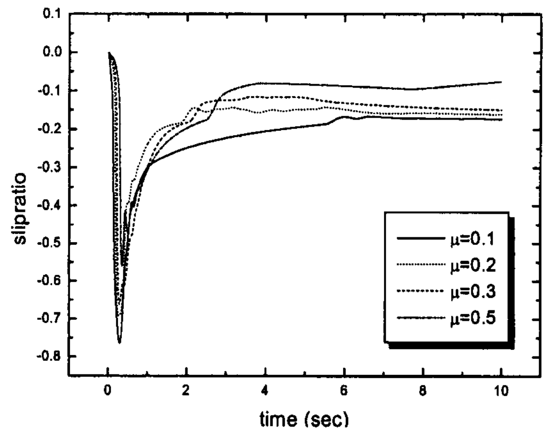


(c) The brake pressure

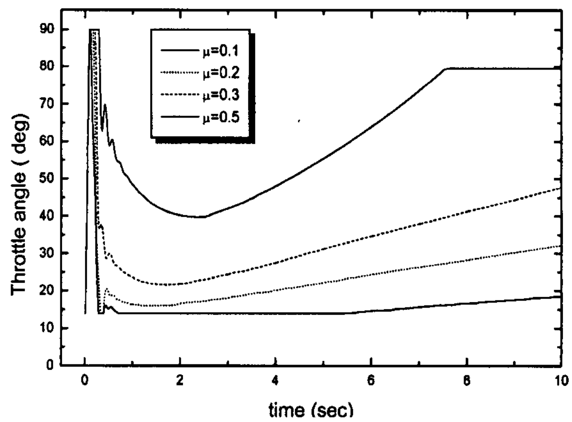
Figure 5. Start off on the icy road.



(a) The accelerations of the driven wheel



(b) The slipratios of the driven wheel

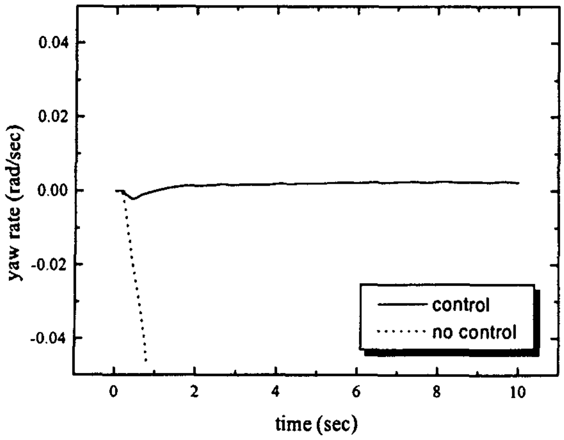


(c) The throttle angle

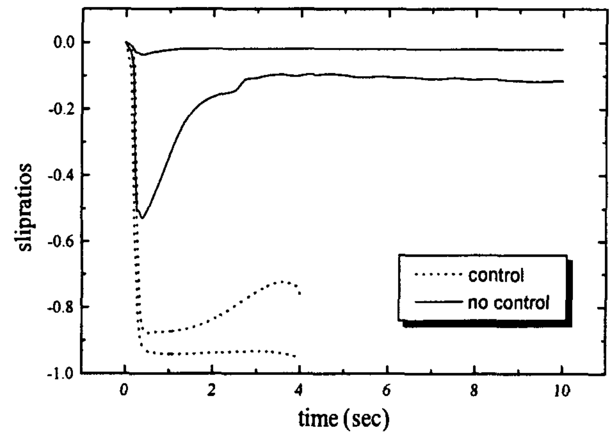
Figure 6. Start off on various roads.

7.2. Start off on the Various Road Conditions

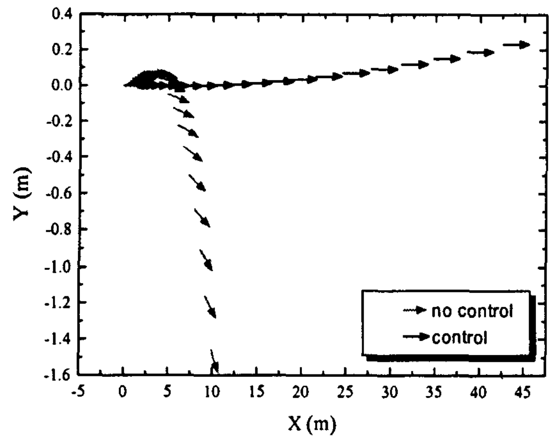
This simulation is for the vehicle to start off with full acceleration pedal operation on the various roads from



(a) The yaw rate of the vehicle



(b) The slipratio of the driven wheel



(c) The vehicle path

Figure 7. Start off on the split-μ road.

the road friction coefficient (μ) 0.1 to 0.5. The slip ratio of the driven wheel converges to the desired value regardless of road conditions and the acceleration performance is satisfied except the case when μ is 0.5 where the traction torque itself is not sufficient to generate the slip of the driven wheel.

7.3. Start Off on the Split-μ Road

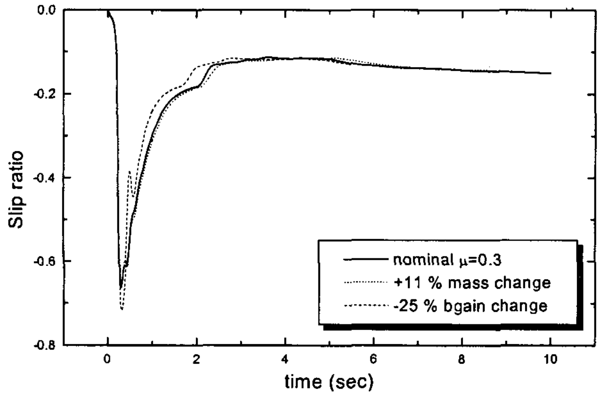
Another major task of TCS is to guarantee the acceleration performance on the split-μ road, and to start without additional steering input. The simulation is conducted with the maximum throttle angle input with μs of the right and left side roads being 0.3 (snowy road) and 0.1 (icy road), respectively. Without the controller, the lateral adhesion forces decrease as the large slip is generated in the driven wheels. If the driver does not make a corrective action, the vehicle spins out to the direction of low μ road and, then, leaves the course. The controller applies the same brake pressure computed from the first slip generated wheel to the another driven wheel. The slipratio of the left wheel converges to the desired value. The slipratio of the right wheel remain near zero as the brake pressure applied to the right wheel is excessive. As the slipratios remain low, the lateral adhesion forces are enough to prevent the yawing. The vehicle with the proposed controller tracks the course as shown in Figure 7. The small yaw rate error and course deviation can be alleviated if individual brake pressures are applied to the driven wheels using the lateral dynamics of the vehicle.

7.4. Vehicle Mass and Brake Gain Change

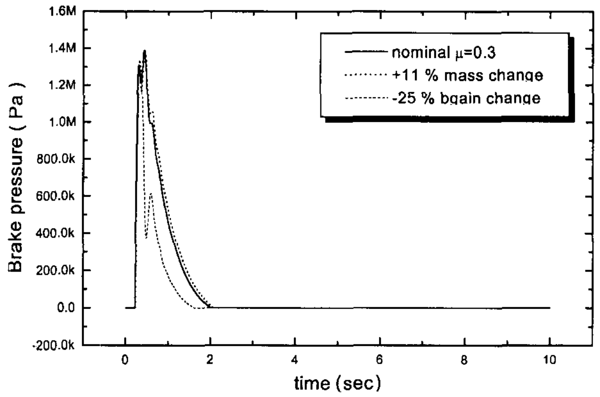
During the driving, the vehicle mass and the brake gain can change significantly according to the weather conditions and vehicle loads. This simulation shows the variation of the slip ratio when the weight of the vehicle increases to 11% and the gain of the brake decreases to the extent of 25 %. Since these parameters are not used directly in the controller, there is little change in the slip ratio as shown in Figure 8.

8. CONCLUSION

The objective of this research is to design the slip controller of TCS systemically based on the vehicle model. The slip controller is designed to achieve the acceleration performance maintaining the slip ratio of driven wheels low. By using the Lyapunov redesign method, the brake pressure is regulated to guarantee the robustness of the system to the vehicle mass variation and the road condition change. The controller structure is simplified and some measurement problems are resolved as the relative degree of the brake system is decreased. The throttle controller is



(a) The slipratios of the driven wheel



(b) The brake pressures of the driven wheel

Figure 8. The variation of the vehicle mass and the brake gain.

designed using the multiple sliding mode control and the hybrid structure of the slip and the throttle controllers is proposed. The proposed method is validated with simulations based on 15 DOF passenger car model. Experimental validation will be needed to apply this controller to the vehicle system.

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