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Brake Performance Evaluation of ABS with Sliding Mode Controller on a Split Road with Driver Model

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In this paper, Sliding Mode Controller (SMC) is proposed to enhance Anti-Lock Brake System (ABS) performance. To verify SMC performance, a real-time Hardware in the loop simulation has been created with a hydraulic brake line. Therefore, the hydraulic brake model and vehicle model should be properly set up to acquire exact simulation results. In addition, the experiment results are compared with that of the commercial ABS with ECU only, and verified how much the performance is improved. The control strategy is to follow the target slip ratio by means of sliding mode controller and secure the vehicle stability while the vehicle braking on various road conditions, such as dry road, wet road, icy road and even split road condition. The driver model is useless on the uniform slip ratio of a straight road. However, the split road has to adopt the driver model. The split road condition has a different slip ratio at each wheel, causing the vehicle to spin out. Test results show that ABS with sliding mode controller has better performance than existing ABS and also ensures improved vehicle stability. Furthermore, the test result on the split road shows how the vehicle will follow the desired path with the driver model and hold the target slip ratio.

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NOMENCLATURE

A = area of master cylinder (m^2)
f_r = rolling resistance coefficient
F_D = aerodynamic drag force (N)
I_w = moment of inertia of wheel (kgm^2)
T_b = brake torque (Nm)
T_t = tire traction torque (Nm)
T_{roll} = tire rolling resistance torque (Nm)
T_{eng} = output torque of transmission (Nm)
R_w = wheel radius (m),
R_b = distance from center of wheel to brake path
P_M = pressure of a master cylinder by the driver (N/m^2)
P_0 = pressure at previous sampling step (N/m^2)
ω = wheel angular velocity (rad/sec)
ε = lateral offset of the vehicle from the desired path
θ = wheel angle (radian)
δ = steering angle of wheel
γ = yaw angle of vehicle body
λ_d = desired tire slip ratio
λ_s = current tire slip ratio
Φ = thickness of boundary layer

SUBSCRIPTS

b = Brake
t = Tire
x, y, z = Longitudinal, Lateral, Normal

1. Introduction

The anti-lock brake system (ABS) has been applied on passenger cars to improve brake performance and allows the driver to steer a vehicle by preventing the wheels from locking. The fundamental objectiveness of ABS has yet to be changed and many researchers have explored how to improve performance over existing systems.

There have been many research attempts with various control methodologies and algorithms for ABS, but the commercial ABS could not hold the target slip ratio regularly during full deceleration, frequently generating yaw and roll motion. The reason ABS fails to hold the desired slip ratio lies on its dependence on ECU mapping data for the control strategy. In addition, many researchers have studied ways to make up for the demerit of mapping a data-based control strategy. Roderick *et al.* studied the Sliding Mode Control (SMC) to apply to vehicle steering given an uneven friction condition and proposed measurable parameters to the observer,[1] but they adopted a 3 D.O.F. vehicle model with a 4-wheel steering system. Patel *et al*. adopts a sliding mode observer with LuGre friction model and simple vehicle model.[2] The simple vehicle model has a merit of fast calculation time, but many terms have been omitted and it is inaccurate.

The ABS research field has been evolved, however, it is difficult to design an optimal ABS controller that can keep the target slip ratio regularly. This is because there are so many nonlinearities and uncertainties such as hydraulic fluid, friction, rubber, heat transform, and so on. Therefore, Semmler *et al.* adopted feedback linearization to diminish the disturbance as a nonlinearity countermeasure[3] while Will *et al.* applied fuzzy logic to diminish the uncertainty for the ABS control methodologies.[4] S. Park *et al.* proposed a reference slip ratio generation method for adaptive sliding mode control of ABS in railroad systems.[26]

In the research field of hardware systems such as ABS, various methods and approaches for the test, even field tests, have tried both the theoretical method and experimental method.[5-7]

Theoretical method normally means software simulation implementation. The results of system simulation cannot display the perfect movement of the hardware system because the hardware system normally has a nonlinearity and uncertainty. Therefore, a real hardware test is necessary to compensate for the imperfect simulation result, but this entails so much cost and time as well as various test data for performance analysis.

Hardware in the loop simulation (HILS) system can address these weak points of both theoretical method and experimental method.[8] HILS could lower cost and make the test earlier than a field test. The results even have more accuracy than simulation. The HILS system is based on the simulation method, but HILS is different from simulation because that has some actual hardware such as a hydraulic unit. The reliability of HILS results can be obtained by using actual hardware and well-known commercial software such as CarSim.[9] In the research area of vehicle dynamics, Bilin *et al.* designed a real time hardware-in-the-loop simulator and showed test results on steering performance with driver-in-the-loop simulation to enhance the yaw stability.[10]

In a review of control strategy research, researches based on the simulation method have been increased to improve performance by using Sliding Mode Controller (SMC) in many laboratories. This is primarily because SMC has robust control specification for the nonlinearity and uncertainty of the vehicle.[11-13,25]

This paper focuses on the implementation of a SMC for ABS into a real ABS hardware. A HILS system has been also consisted of the actual hydraulic unit, commercial ABS-ECU, and real-time system for validating the designed sliding mode ABS controller. The designed controller's performance is compared with commercial ABS ECU's performance on various road conditions. The test results on dry, wet and icy road could be acceptable on the straight course with sudden stopping, but the split road test should have a driver model because the driver has a tendency to steer to another direction of the vehicle's heading way. This tendency may cause loss of control of the vehicle, making it spin out from the desired path on the road.

2. Vehicle Dynamic Model and Controller Design

The first step in the SMC design procedure is constructing a vehicle dynamics model for predicting dynamics specification. However, it is too difficult to make a full vehicle model with nonlinearity and deformation of parts, so the vehicle model has been made with various simplifying assumptions in order to make the model manageable while still approximating dynamical reality. In this paper, the real time system of the HILS has been constructed with vehicle model in CarSim while control algorithm with SMC is developed and evaluated.

2.1 Reference Vehicle Model and CarSim Vehicle Model

A reference vehicle model designed in this paper has 15 degree of freedom and it has subsystem models such as tire, braking system, steering system, suspension model and interaction model between tires and roads, a power train model. Especially, the slip ratio between tires and the road is a very important parameter on ABS, so SMC is designed to improve the braking performance by controlling the slip ratio stay within the desired zone. Figure 1 shows the reference vehicle model with 15 degree of freedom.

The vehicle model in CarSim supports the 27 degree of freedom vehicle model and it can calculate the vehicle dynamic terms at real

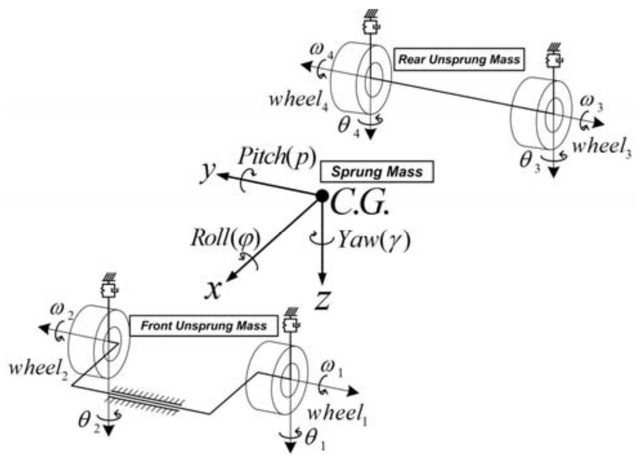


Fig. 1 Vehicle model with 15 degree of freedom

Table 1 Input and output variables of CarSim

	Variables
Input	Brake Pressure
Output	Vehicle Speed
	Vehicle Acceleration
	Rotational Wheel Speed
	Slip Ratio
	Pitch
	Normal Force (Tire)
	Longitudinal Force (Tire)

time using the Opal-RT real time host controller. CarSim has many input and output parameters for the vehicle dynamic control and it is easy to apply the parameter tuning to the controller or algorithm. CarSim can also be used in real-time calculation and can be directly connected with MATLAB/Simulink.[11,14,15]

In this paper, CarSim has been applied to the HILS system and the hydraulic line with a real-time operation system. The vehicle parameters used in CarSim are shown in Table 1.

2.1.1 Tire model

The tire model should consider the interactions between each tire and road and it can be modeled as in the following expression. However, the engine output torque (T_{eng}) and aerodynamic drag force (F_D) can be negligible when the vehicle is decelerating.[16]

$$\begin{aligned} I_{wi}\dot{\omega}_i &= -T_{bi} - T_{ti} - T_{rolli} + T_{eng} \\ &= -AR_bP_{bi} - F_{xi}R_w - F_{rolli}R_w + T_{eng} \end{aligned} \tag{1}$$

where subscript 'i' is number;1, 2, 3, 4, each number mean front left wheel, front right wheel, rear left wheel, rear right wheel respectively. F_{roll} is rolling resistance force and P_b represents brake pressure of each wheel. For calculating the vehicle's longitudinal velocity, Newton's second law can be applied to the vehicle model.

$$m_{total}(\dot{v}_x - v_y\dot{\gamma}) = \sum_{i-1}^{4}((F_{xi} - F_{zi})\cos\delta_{fi} - F_{yi}\sin\delta_{fi}) - F_D \tag{2}$$

$$m_{total}(\dot{v}_y - v_x\dot{\gamma}) = \sum_{i-1}^{4}((F_{yi} - F_{zi})\sin\delta_{fi} - F_{xi}\cos\delta_{fi}) - F_D \tag{3}$$

where m_{total} is the total mass of the vehicle, which consists of the sprung mass (m_s), front unsprung mass (m_{uf}) and rear unsprung mass (m_{ur}). Vehicle's longitudinal velocity v_x can be calculated from equations (2) and (3). Then, F_{xi} and F_{yi} can be obtained by the tire model. The slip coefficient λ_s can also be calculated by the following equation (4).

$$\lambda_s = \frac{R_w\omega_i - v_x}{v_x} \tag{4}$$

2.1.2 Driver model

The vehicle-driver-environment model has been considered and developed over recent years for longitudinal and lateral movements.

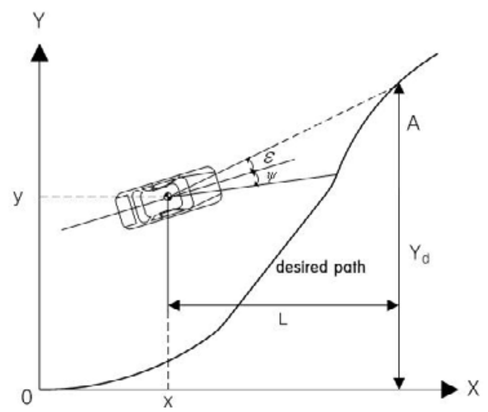


Fig. 2 Strategy of desired path pursuit

However, the driver model cannot mimic a real human driver perfectly and cannot anticipate the road and environment conditions. To describe the functions of the driver mathematically, the driver model should be reduced to several simple parameters.[17]

On the condition that the vehicle drives on a straight road, steering input to the vehicle model is not required, because the simulation history in this paper does not include unexpected obstacle on the road profile. When the vehicle stops suddenly on the split road, the steering of the vehicle is required to keep the desired path and should be exerted gradually for safety. Fig. 2 shows the strategy of the lateral drive model to estimate the steering angle from the driver's desired path based on the dynamic characteristics of the driver and vehicle. Equation (5) describes the geometric relationship from the look-ahead distance (L), yaw angle (ψ) and the desired heading angle (ε).[12] In this paper, the driver is assumed to follow the desired path by means of reducing heading error with time constant (τ_{sd}). Equation (6) denotes the mathematical expression of driver model.

$$\varepsilon = (Y_d - y)/L - \psi \tag{5}$$

$$\tau_{sd}\,\dot{\delta}(t) + \delta(t) = a_1\cdot(V/L)\,\varepsilon(t) - a_2\,\dot{\psi}(t) \tag{6}$$

where a_1 and a_2 are constants which represent the characteristics of the driver and V is the vehicle velocity.

2.1.3 Hydraulic brake model

Because ABS has hydraulic circuits and solenoid valve system to generate the desired brake pressure, it is difficult that the mathematical model of the ABS is established. In this paper, the HILS system which has the actual hydraulic circuits, the solenoid valve system and an actual ECU has been utilized to simulate the actual braking operation of the vehicle.

SMC in ECU gets the steering wheel and brake pedal operation of the driver as inputs, and generates hydraulic pressure commands to the solenoid valve system in the HILS system as outputs.[5,18] Thus, SMC needs the solenoid valves operation algorithm shown in Fig. 3

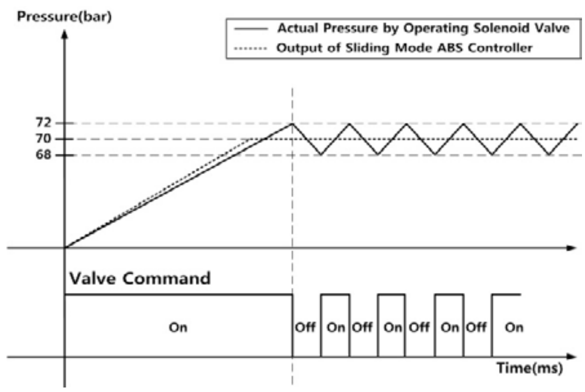


Fig. 3 Scheme of solenoid valve operation logic

Table 2 Solenoid valve control modes

Mode	NO Valve	NC Valve
Increase	0	0
Decrease	1	1
Hold	1	0

to generate the requires hydraulic pressure for stopping the vehicle with ABS function.

Generally, ECU on the passenger car does not monitor the hydraulic braking pressure for the ABS function. Therefore, the relationship between the solenoid valve system and the output pressure of the hydraulic system should be defined to determine the valve's position in the HILS system. The hydraulic pressure model consists of three modes in Table 2, which are determined by valve condition, NO (Normal Open) and NC (Normal Close). In addition, each mode has its own mathematical pressure model as shown in equations (7), (8), and (9), respectively.

$$\textbf{\textit{Increase}}\text{: } P(t) = P_M \cdot e^{-\frac{1}{\tau_1}t} + P_M + \frac{P_0}{\tau_1} \cdot e^{-\frac{1}{\tau_1}t} \tag{7}$$

$$\textbf{\textit{Decrease}}\text{: } P(t) = \frac{P_0 a}{\tau_1} \cdot e^{-\frac{1}{\tau_1}t} \tag{8}$$

$$\textbf{\textit{Hold}}\text{: } P(t) = P_0 \tag{9}$$

where P_M is the master cylinder pressure and P_0 is pressure in the previous sampling step. τ_1 is fluid time constant and a is only used in decrease mode for the initial value.

Each hydraulic pressure mode has its own valve condition and it switches to other modes by means of the difference between SMC output and actual hydraulic output. Fig. 4 shows the diagram of mode switching strategy and the solenoid valve control scheme to generate optimal braking pressure from SMC. On the condition that the difference between both outputs go over the specified boundary (it is set to ±0.2Mpa), the hydraulic pressure model shifts to another mode of the proper valve condition.

2.2 Sliding mode controller design

SMC has been employed in research on an ABS controller. One of the advantages of SMC is that it does not require an accurate model and it has a strong robustness. In addition, it is insensitive to uncertainty and disturbances. However, SMC has to consider the design of sliding surface, to guarantee the existence of the sliding mode and the controller design. In previous research, SMC for the brake torque and slip ratio control has two types of sliding surface design, namely, $s_1 = \tilde{x} = \lambda - \lambda_d$ and $s_2 = \tilde{x} + a\int_0^t \lambda_r dr$. [1,2,11]

These designs have a same strategy to control the slip ratio by the pursuit of tracking error. Hur *et al.*[16] have designed a sliding

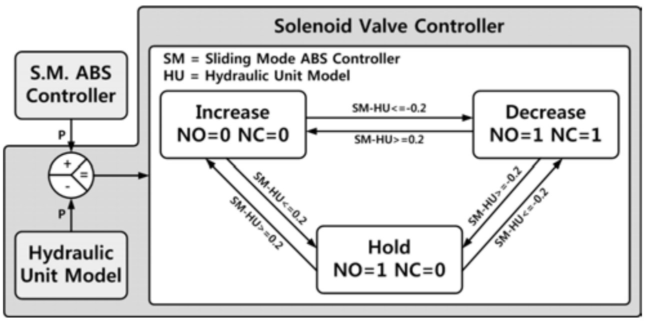


Fig. 4 Solenoid valve control diagram

surface, determined from the condition of $\dot{s}_1 = 0$ and corresponding to bang-bang control. However, system error dynamics between the tracking error and derivative is not incorporated. For the second sliding surface s_2, Song *et al.*[9,11,12] have designed same conditioned controller, however, the exponential error convergence can be found in $\dot{s}_2 = \dot{\tilde{x}} + \tilde{x} = 0$. Recently, the Lyapunov stability method is applied to determine the magnitude of the switching control gain on the slip ratio error domain. Shim *et al.*[19] has compared two types of sliding surface designs commonly used in ABS, and proposed the alternative sliding surface design can improve convergence speed and oscillation damping around the target slip by the Lyapunov stability condition. However, alternative sliding surface takes considerable time to calculate for the real time HILS system.

The equation (1) can be rewritten to the following equation (10).

$$\dot{\omega}_i = -(K_i u_i + \tau_{xi} + \tau_{ri}) \tag{10}$$

where $K_i = \frac{AR_b}{I_{wi}}$, $\tau_{xi} = \frac{F_{xi}R_w}{I_{wi}}$, $\tau_{ri} = \frac{F_{ri}f_{ri}}{I_{wi}}$. The P_{wi} in equation (1) is defined as control input u_i, torque from engine T_{eng} in equation (1) is not considered in equation (10). The sliding surface to design a sliding mode controller is defined as

$$S = \left(\frac{d}{dt} + \lambda\right)^{n-1} \int_0^t \lambda_r dr = \lambda_r + \lambda \int_0^t \lambda_r dr \tag{11}$$

where λ is a strictly positive constant, $\lambda_r = \lambda_{si} - \lambda_{di}$, λ_{di} is defined as 0.2 and n = 2. The optimal slip ratio is between 0.15 and 0.25 as depending on the road conditions, and the desired slip ratio can be set 0.2.[20]

The best approximation $\hat{u}_i$ can be obtained based on the condition $\dot{S} = \dot{\lambda}_r + \lambda \cdot \lambda_r = 0$ of the continuous control method.

$$\hat{u}_i = -\frac{1}{v_x K_i}\left[(\tau_{xi} + \tau_{yi})v_x + \omega_i \dot{v}_x - \frac{v_x^2 \lambda}{R_w}(\lambda_{si} - \lambda_{di})\right] \tag{12}$$

Because precise value of τ_x and τ_r cannot be known and they can be estimated as $\hat{\tau}_x$ and $\hat{\tau}_r$. Equation (12) describes the assumption that the estimation errors of braking torque, $\hat{\tau}_{xi} - \tau_{xi}$ and $\hat{\tau}_{ri} - \tau_{ri}$, are zero. The estimation errors of τ_x and τ_r are assumed to be bounded by the known values, τ_x^* and τ_r^*. But this assumption is not true, a discrete function u_i defined as equation (13) is added to equation (12) to satisfy the sliding condition.

$$\overline{u}_i = \frac{\tau_{xi}^* + \tau_{ri}^* + \eta}{K_i}\operatorname{sgn}(S) \tag{13}$$

where η is strictly positive constant. The control input $\overline{u}_i$ can be obtained from equations (12) and (13) as follows:

$$\begin{aligned} u_i &= \hat{u}_i + \overline{u}_i \\ &= -\frac{1}{v_x K_i}\left[(\hat{\tau}_{xi} + \tau_{yi})v_x + \omega_i \dot{v}_x - \frac{v_x^2 \lambda}{R_w}(\lambda_{si} - \lambda_{di})\right] \\ &\quad + \frac{\tau_{xi}^* + \tau_{ri}^* + \eta}{K_i} + \operatorname{sat}(S/\Phi) \end{aligned} \tag{14}$$

The chattering problem in the equation (13) can be eliminated by using a thin boundary layer of thickness Φ next to the switching surface. The chattering caused by the discontinuity of the sgn(S) function of equation (13) can be adjusted by changing sgn(S) to sat(S/Φ) in the equation (14).[11]

2.3 Simulation environments and conditions

In this paper, the designed SMC should be verified by faithful and economic method. Therefore HILS with hydraulic brake line in the loop system is proposed and JASO test environment and special condition should be specified in detail.[21] Fig. 5 shows the HILS system with hydraulic brake line and brake parts, caliper, disk, rack bar, steering column and dash board, etc. In addition, the HILS system has one host PC, two target PCs and ABS brake systems. ECU in the loop simulation system contains MK25 ESP ECU of Continental Teves as the reference data, which is the existing controller for ABS. For real-time calculation, the HILS system is constructed by MATLAB, CarSim software, and Opal-RT host controller. The controller output operates solenoid valves, and then the brake pressures are consequently generated. The designed SMC with proposed model and strategy can be installed with this HILS system through MATLAB/Simulink, and interface with vehicle dynamics model in CarSim.

CarSim is the main simulation environment in this paper and it is verified by various results in other papers.[8,14,22] To precisely simulate the road condition and vehicle dynamics, input and output variables for the system should be defined as in Table 1. In simulation history, road conditions of various frictions for the performance evaluation of SMC should be setup by the JASO ABS test regulation as shown in Table 3. In addition, the control goal of SMC is to hold the desired slip ratio as 0.2, because this ratio means that the brake force is maximized at a vertical force of the tire irrespective of the road conditions.[22,23]

Fig. 5 HILS system for the simulation test

Table 3 JASO ABS test conditions

Road	Condition	Friction	Velocity
Dry road	dry asphalt	0.8	120 km/h
Wet road	wet asphalt	0.5	80 km/h
Icy road	Ice asphalt	0.2	50 km/h
Split road	dry asphalt	0.8	50 km/h
	ice asphalt	0.2	

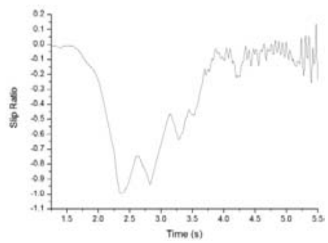
3. Results and discussion

3.1 Dry road test

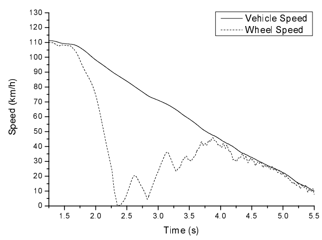
Dry asphalt offers the best condition for braking the vehicle with friction coefficient, approximately 0.8. Fig. 6 shows the results of sudden stop simulation with designed SMC and it is verified by comparing with existing ABS' results. Although the initial velocity of vehicle is 120Km/h, actual decelerating of the vehicle starts from 110Km/h, because the braking begins just after the driver steps on the pedal for a sudden brake. Fig. 6(a), (c) show the time responses of the slip ratio tracking of the vehicle and Fig. 6(b), (d) show the velocity of vehicle decelerating by the ABS and SMC.

Fig. 6(a) denotes that when the vehicle suddenly stops, the slip ratio of a vehicle with commercial and existing ABS only has a tendency to decline radically. Fig. 6(b) even shows the wheel speed runs down to zero at around 2.3 second. This means that the large oscillation of slip ratio causes a big discrepancy between the vehicle and wheel speed. Actually, this is the source of vehicle instability.

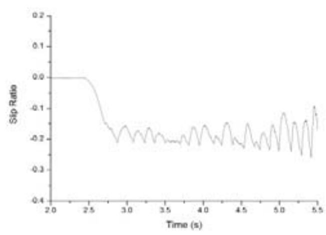
However, the slip ratio of vehicle with designed SMC is confined within the desired slip ratio, around 0.2 in Fig. 6(c).



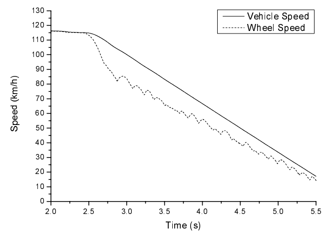
(a) Slip ratio of commercial



(b) Velocity of commercial

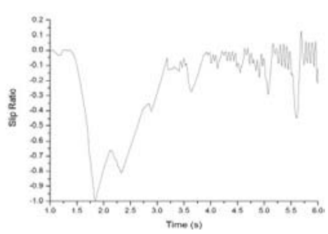


(c) Slip ratio of proposed system

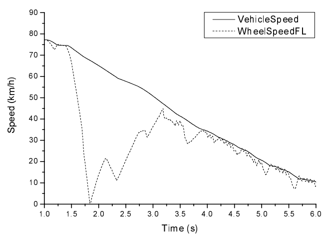


(d) Velocity of proposed system

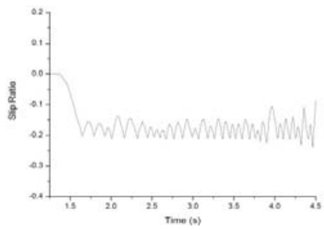
Fig. 6 ABS brake test on dry road



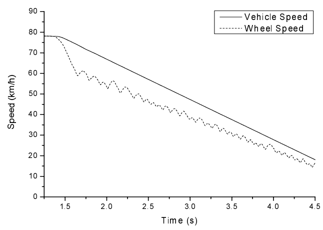
(a) Slip ratio of commercial



(b) Velocity of commercial



(c) Slip ratio of proposed system



(d) Velocity of proposed system

Fig. 7 ABS brake test on wet road

Consequently, Fig. 6(d) shows that the vehicle velocity is decelerated uniformly by SMC and the velocity of wheel slow down without considerable difference.

3.2 Wet road test

The coefficient of friction on a wet road is 0.5 as shown in Table 3. For the same reason in the case of a dry road, the initial velocity of the vehicle is 80Km/h, but actual decelerating of the vehicle starts from 75Km/h. Fig. 7(a) shows that the vehicle slip ratio has a tendency to shake roughly than in Fig. 6(a). Fig. 7(c) proves that SMC has a better performance than ABS at the same simulation and the slip ratio stays within the stable boundary of 0.2. The bottom line in a sudden stop is that the vehicle should stop with minimal braking distance. Fig. 7(b) shows that the vehicle takes 5.5 seconds to 20Km/h after slowing down, but Fig. 7(d) shows that the necessary time is reduced to 4.5 seconds. It means that the shorter time to speed down allows for the shorter braking distance. Therefore, the SMC yields higher safety and comfort than ABS alone.

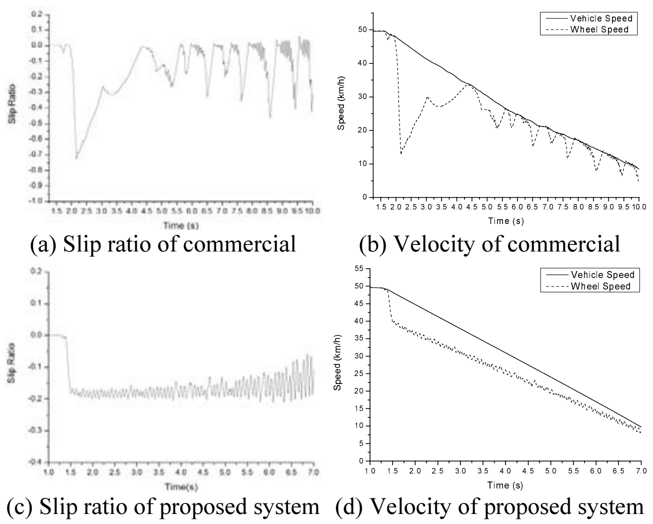
3.3 Icy road test

An icy road poses a very bad condition for braking and its friction coefficient is 0.2 based on the JASO regulation in Table 3. Normally, the velocity of a vehicle below 10Km/h on icy road is the stable speed, because the vehicle can stop almost without slip. Fig. 8(a) shows that ABS without SMC has a large slip just after sudden stopping, causing instability to the vehicle, i.e., pitching and rolling motion.

Consequently, it takes a longer time to reach a stable velocity (10Km/h), about 10 seconds as shown in Fig. 8(b). However, the slip ratio of SMC in Fig. 8(c) verifies that the vehicle performance on the icy road is improved. In addition to this improvement, the settling time to stable speed is cut down to 7 seconds in Fig. 8(d). It denotes that braking distance is shortened as well.

3.4 Split road test

A split road has a different tire friction coefficient on both driver and passenger sides. Simulation of sudden stopping with HILS on a split road requires the driver model in section 2.1.2., because the driver used to steer the vehicle to the desired heading way, not a drifting way.[24] Fig. 9(c) shows the ABS slip ratio without SMC, revealing a large oscillation, especially around 5.6 second. It also causes a pitching motion of the vehicle in Fig. 9(a). At around 5.6 second in Fig. 9(c) and Fig. 9(e), the vehicle's speed is down to a stable zone and the friction of tire and road rise abruptly, which causes yaw and pitch motion. Consequently, the wheel velocity passes vehicle velocity by the time of complete stop in Fig. 9(e). Otherwise, Fig. 9(d) looks like an unstable phenomena, however, the longitudinal velocity v_x is decreasing to zero from equation (4). This is just numerical oscillation. It proves that SMC controls the wheel velocity by the pursuit of desired slip ratio and the wheels follow the vehicle velocity. Consequently, Fig. 9(f) shows both wheels pursue the vehicle's velocity with the most effective steps available. In addition, SMC improves the vehicle stability by confining the pitch motion of the vehicle in Fig. 9(b).



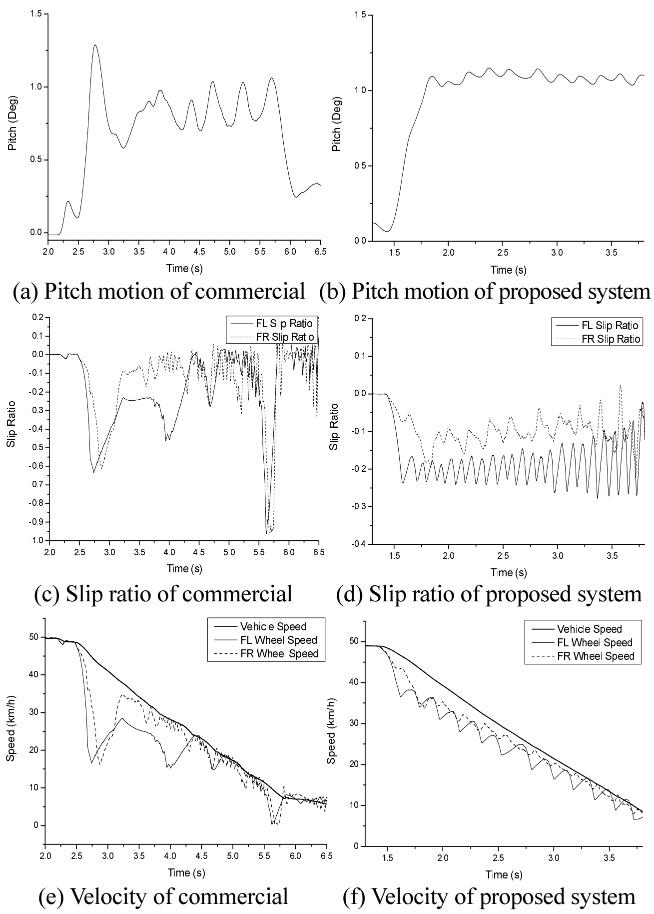
(a) Slip ratio of commercial (b) Velocity of commercial

(c) Slip ratio of proposed system (d) Velocity of proposed system

Fig. 8 ABS brake test on icy road

4. Conclusions

In this paper, a sliding mode controller for an anti-lock braking system is developed to hold the target slip ratio. Simulation results with the HILS system verified the vehicle has a better performance than one controlled by ABS alone.



(a) Pitch motion of commercial (b) Pitch motion of proposed system

(c) Slip ratio of commercial (d) Slip ratio of proposed system

(e) Velocity of commercial (f) Velocity of proposed system

Fig. 9 ABS brake test on split road

The main findings of this investigation are summarized below:

1. Vehicle model set up for the simulation and the driver model and hydraulic brake model for the HILS system are considered in this paper.
2. To verify the performance of SMC designed in this paper, various road conditions with different friction settings were used as per JASO regulation.
3. SMC is compared with the commercial and existing ABS with electrical control unit by HILS system and the results prove SMC has better performance than the ABS-ECU alone.
4. SMC can control the slip ratio to stay within the desired zone and improve vehicle braking and steering stability. The vehicle is more stable on sudden braking and shorter braking distance than commercial ABS-ECU.

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