Anti-lock Brake System With a Semi-Model Based Controller Using Wheel Pressure Sensors

Kwanghyun Cho[1)], Seibum Choi[2)], Kimo Son[3)], and Jongkap Kim[4)]
[1, 2)]KAIST and [3, 4)]Hyundai Motors Corporation

[1,2)]Department of Mechanical Engineering, KAIST, Daejeon, 305-701, Republic of Korea
Phone: (+82-42) 350-4104, Fax: (+82-42) 350-4110
[3,4)]Hyundai & Kia Corporate Research & Development Division, Seoul, 137-938, Republic of Korea
Phone: (+82-2) 3464-7692, Fax: (+82-2) 3464-1114

E-mail: [1)]khcho08@kaist.ac.kr, [2)]sbchoi@kaist.ac.kr,
[3)]kimosohn@hyundai.com, and [4)]ckkim@hyundai.com

The Anti-lock Braking System(ABS) must guarantee to shorten the braking distance and keep the stability of the vehicle as soon as possible during the vehicle braking. To do this work, the target slip(acceleration) must be known to control each four wheel along the road conditions. However, it is difficult to determine the optimal target slip because the road friction coefficient is unknown. In this study, the Anti-lock braking system using additional four wheel pressure sensors has been proposed. The road friction coefficient is calculated by using the wheel dynamics and vehicle longitudinal dynamics. The system is controlled by a semi-model based controller which is the combination of the semi-feedback control and the feedback control. A variety of simulations are carried on to verify the performance of the proposed system using CarSim program. They are compared with the system designed without four wheel pressure sensors. The simulation results show that the proposed algorithm make the braking distance be shorten and the stability of the vehicle be improved.

Vehicle Dynamics, Active Safety Systems

1. INTRODUCTION

The Anti-lock Braking System has been equipped on most vehicles to guarantee the longitudinal stability of the vehicle. The basic function of the system is control the applied braking torque by modulating the brake pressure. It prevents four wheels from locking during the vehicle braking. It must guarantee to make the braking distance as soon as possible and keep the stability of the vehicle simultaneously.

In the braking control systems, there are two variables to be considered for controlling four wheels: wheel longitudinal slip and wheel acceleration.[1] The traditional ABS uses wheel acceleration to control four wheels.[2][3] These systems have many complicated threshold-based control rules, so it takes long time to apply to a variety of vehicles. Therefore, many researches have been performed for the transition from rule-based control to model-based one because of tuning simplicity and robustness.[4][5][6][7][8][9]. Most model-based systems assume that the optimal control target (desired wheel slip or speed) is known. These systems consider the control issue in ABS as only a tracking problem. Actually, it is impossible to define the control target exactly along a variety of road conditions because the road friction coefficient is unknown and it is hard to know one using only equipped sensors such as an yaw rate sensor and a longitudinal accelerometer.

In this paper, a semi-model based anti-lock braking system has been proposed. The system uses four wheel pressure sensors additionally to calculate the road friction coefficient from the wheel dynamics and the longitudinal vehicle dynamics. The control targets of four wheels are obtained from the estimated road friction coefficient. Therefore, it can improve the implementation problem which is main issue of the model-based control system architecture. The rear wheels are controlled by reference wheel accelerations based a semi-feedback controller. And the front wheels are controlled by an adaptive sliding mode controller based on longitudinal wheel slip (wheel speed). The performance of the proposed system is verified by simulation using CarSim program under a variety of road conditions. Also, they are compared with results of a system designed without for wheel pressure sensors.

The paper is organized as follows: In Section II, the estimation scheme of the road friction coefficient is described. The proposed ABS control algorithm is presented in Section III. Two different simulation tests are performed in Section IV to verify the performance of the proposed ABS algorithm. Concluding remarks are given Section V.

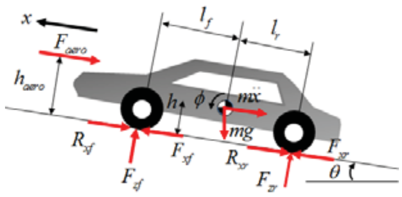

Fig. 1 Vehicle Longitudinal Dynamics

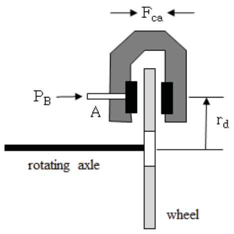

Fig. 2 Relation Between Force and Pressure

2. ROAD FRICTION COEFFICIENT ESTIMATION

In this section, the normal forces and brake torques are calculated to estimate the road friction coefficient. Four wheel pressure sensors, which have been used for a regenerative braking system in the hybrid vehicles or an ACC(adaptive cruise control) system recently, are used to measure the pressure of each wheel. Using wheel dynamics based on these parameters, the road friction coefficient is obtained.[10]

2.1 Normal Force Calculation

The normal forces of the vehicle are calculated by the longitudinal vehicle dynamics. It is described in Fig. 1. When the vehicle is decelerating, the normal forces of front and rear wheels are calculated as follows:

$$F_{zf} = \frac{mgl_r\cos\theta - mgh\sin\theta + m\ddot{x}h - F_{aero}h_{aero}}{L} \tag{1}$$

$$F_{zr} = \frac{mgl_f\cos\theta + mgh\sin\theta + m\ddot{x}h + F_{aero}h_{aero}}{L} \tag{2}$$

where, F_{zf} and F_{zr} are the normal forces of the vehicle, m the total vehicle mass, l_f and l_r the distance from C.G. to front and rear wheel, L the wheel base length, g the gravity acceleration, θ the road elevation, h the height of C.G, and $F_{aero}h_{aero}$ the moment of aero effect.

Here, assume that the road surface is flat, i.e., $\theta \approx 0$ and airdrag is negligible, i.e., $F_{aero}h_{aero} \approx 0$. Then, (1) and (2) are approximated simply as follows:

$$F_{zf} \approx \frac{mgl + m\ddot{x}h}{L} \tag{3}$$

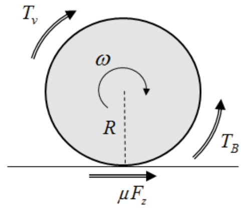

Fig. 3 Wheel Dynamics

$$F_{zr} = \frac{mgl_f + m\ddot{x}h}{L} \tag{4}$$

2.2 Brake Torque Computation

The brake torques are obtained by the relation between force applied to four wheels and cylinder pressure as described in Fig. 2. The force applied to a brake disk pad is obtained as follows,

$$F_B = \mu_b F_{ca} = \mu_b 2F_{BN} = 2\mu_b P_B A \tag{5}$$

where, F_B is the brake force, μ_b the caliper friction coefficient, F_{ca} the clamping force, F_{BN} the normal brake force, P_B the brake pressure, and A the cylinder area.

The brake torque is computed as follows,

$$T_B = F_B \cdot r_d = k_B P_B \tag{6}$$

where, T_B is the brake torque, r_d the disk radius, k_B represents the brake gain which is determined by the specification of brake cylinder. In this study, the brake gains of front and rear wheels are 200Nm/MPa and 70Nm/Mpa.

2.3 Road Friction Coefficient Estimation

The road friction coefficient is calculated by wheel dynamics. It describes dynamical representation for wheel driving on the road as shown in Fig. 3.

The moment balance equation is obtained as follows,

$$I_w\dot{\omega} = RF_x - T_B \tag{7}$$

where, I_w is wheel inertia, ω the wheel speed, μ the road friction coefficient, R the wheel radius, and F_x the longitudinal tire force, and F_z the normal tire force. Also, the relation between the longitudinal tire force, F_x and the normal tire force, F_z is represented as follows,

$$F_x = \mu F_z \tag{8}$$

In (7) and (8), the road friction coefficient is obtained by normalizing the longitudinal tire force using the normal tire force. It is presented as follows,

$$\mu = \frac{F_x}{F_z} = \frac{I_w\dot{\omega} + T_B}{RF_z} \tag{9}$$

The μ obtained in (9) oscillates in accordance with wheel cycling pattern caused by the braking operation such as apply-hold-dump. Therefore, it must be limited by a rate limiter to obtain the maximum road friction coefficient.

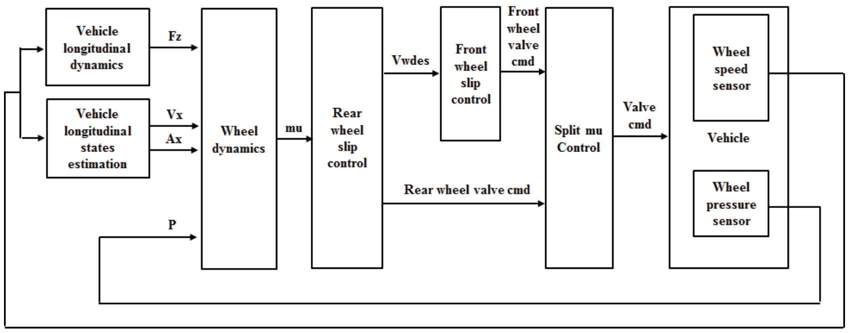

Fig. 4 Block Diagram of ABS Control Algorithm

The rate limiter is implemented as follows:

$$\mu_{peak} = \begin{cases} \mu + \dot{\mu}_{\max} T_s & if \quad \dot{\mu} \geq \dot{\mu}_{\max} \\ \mu - \dot{\mu}_{min} T_s & elseif \ \dot{\mu} < \dot{\mu}_{\min} \\ \mu & otherwise \end{cases} \qquad (10)$$

where $\dot{\mu}_{\max}$ and $\dot{\mu}_{\min}$ are the maximum and minimum rate limitation of μ and T_s is the sampling time.

The maximum friction coefficient, μ_{peak}, represents the maximum braking force obtained during the braking. It is used to obtain the control targets of the rear wheels. It can solve the implementation problem mentioned above. Also, it enables all wheels to be controlled under each estimated road friction coefficient. Therefore, it guarantees the lateral stability of the vehicle under a variety of transient conditions of the road friction, especially such as the split friction road condition.

3. ABS CONTROL ALGORITHM

In this section, the control schemes of the front and rear wheels are described. And the control strategies on the split-friction road are presented. The block diagram of the proposed ABS control algorithm is as shown in Fig. 4.

3.1 Rear Wheel Control

The rear wheels are controlled by wheel acceleration based rules. However, the reference accelerations are obtained from measured wheel pressure signals, and the error between measured and reference signal is also used for control. It works such as a P-type controller. Therefore, the rear wheel brake system is not considered as a simple open-loop controller (only rule-based system) but a semi-feedback one. The valve command to control rear wheels is presented as sum of valve dump command and valve apply command as follows:

$$V_{cmd} = V_{dump} \cdot V_{dhold} + V_{apply} \cdot V_{ahold} \qquad (11)$$

where, V_{cmd} is the total valve command, V_{dump} the valve dump command, V_{apply} the valve apply command, V_{dhold} the valve hold command for dump, and V_{ahold} the valve hold command for apply.

The dump and apply valve commands are obtained by the difference between estimated wheel acceleration and reference wheel acceleration. The commands for the valve dump and apply operation are represented as follows:

$$V_{dump} = \begin{cases} a_w - a_{dump} & if \ a_w - a_{dump} < 0 \\ 0 & otherwise \end{cases} \qquad (12)$$

$$V_{apply} = \begin{cases} a_w - a_{apply} & if \ a_w - a_{apply} > 0 \\ 0 & otherwise \end{cases} \qquad (13)$$

where, a_w is estimated wheel acceleration, a_{dump} the reference acceleration for dump and a_{apply} the reference acceleration for apply.

The reference accelerations, a_{dump} and a_{apply} are obtained from the calculated maximum road friction coefficient, $\mu_{\max}$ with additional margin. For example, the reference accelerations, a_{dump} and a_{apply} are determined as follows:

$$a_{dump} = -1\left[|a_{\max}| 1.20 + 0.7g \right] \qquad (14)$$

$$a_{apply} = -1\left[|a_{\max}| 1.05 + 0.1g \right] \qquad (15)$$

where, $a_{\max}$ means the maximum acceleration during braking on the driving road using the road friction coefficient estimated in (10).

$$a_{\max} = \mu_{peak} \cdot g \qquad (16)$$

The valve hold commands determine whether the valve dump/apply commands are applied or not. The commands of the valve hold are as follows:

$$V_{dhold} = \begin{cases} 0 & if \ |V_{dump}| < V_{holdth} \ for \ t < t_1 \\ 1 & otherwise \end{cases} \qquad (17)$$

$$V_{ahold} = \begin{cases} 0 & if \ V_{apply} > 0 \quad for \ t < t_2 \\ 1 & otherwise \end{cases} \qquad (18)$$

where, V_{holdth} is a threshold for dump hold, t_1 a time threshold to keep small dump command, t_2 a time threshold to be ready to start apply command.

The apply-hold-dump commands make cycling patterns of the rear wheels around the peak friction slip point.

3.2 Front Wheel Control

The front wheels have no direct cycling patterns like one of the rear wheels because it makes the ride very harsh on high-friction surfaces. They are controlled by an adaptive sliding mode controller based on the longitudinal wheel slip(wheel speed). The target speeds for preventing wheels from locking are obtained from controlled rear wheel speed with additional slip margin. The validity of the control target is guaranteed by an assumption that the rear wheels are controlled near a peak friction point through a cycling-pattern. Therefore, the desired wheel speed, ω_{Fwdes} is represented as follows,

$$\omega_{Fwdes} = \omega_{Rw} + \Delta\omega \tag{19}$$

where ω_{Rw} is a rear wheel speed, $\Delta\omega$ an additional slip margin. In (7), the change rate of brake torque is proportional to fluid flow rate, and flow rate is proportional to control valve opening. Therefore, the brake torque rate is proportional to the valve command and then it becomes the control input.[11] Using (7), the system model to design the controller is defined as follows:

$$\ddot{\omega} = -\frac{1}{I_w}\dot{T}_B + \frac{1}{I_w}\frac{d}{dt}\left(\mu R F_z\right) \tag{20}$$

The state-space representation is described as follows,

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ 0 & 0 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ -1/I_w \end{bmatrix} u + \begin{bmatrix} 0 \\ 1 \end{bmatrix} d \tag{21}$$

where,

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} \omega \\ \dot{\omega} \end{bmatrix},\ u = \dot{T}_B,\ d = \frac{1}{I_w}\frac{d}{dt}\left(\mu R F_z\right)$$

Here, the term, d, assumes that it is the disturbance, which is calculated by (3) and (9).

The control objective is to track the desired wheel speed, ω_{Fwdes} with tracking error as small as possible. In this study, the sliding mode controller is proposed and the adaptation mechanism is used to estimate the disturbance, d. In case that the information of d is totally unknown, the adaptation mechanism does not work well because it is a fast time-varying parameter. This makes the tracking performance very poor. However, the nominal value of d, d_n can be calculated using (3) and (9) in this study. Therefore, the state-space representation (21) is obtained as follows:

$$x_2 = -\frac{1}{I_w u} + d_n + \Delta d \tag{22}$$

Here, $\Delta d = d - d_n$ is the error between the actual disturbance and the nominal disturbance. To design the sliding mode controller, the PD-type of sliding surface is defined as follows,

$$S = \dot{e} + \lambda e,\ \lambda > 0 \tag{23}$$

where, $e = x_1 - x_{1d}$ is the tracking error, x_{1d} being the desired wheel speed, ω_{Fwdes} and λ the tuning parameters.

The aim of the sliding mode control is to force the system state to the sliding surface, $S = 0$ and then maintain it on the sliding surface. Therefore, the sliding surface, S must satisfy,

$$S\dot{S} \le 0 \tag{24}$$

And (24) can be satisfied using (25)

$$\dot{S} = -K_S S,\ K_S > 0 \tag{25}$$

From (25), the control law is obtained as follows,

$$u = u_{ff} + u_{fb} = I_w\left[\underbrace{d_n + \Delta\hat{d} - \ddot{x}_{id}}_{feedforward} + \underbrace{\lambda\dot{e} + Ks\left(\dot{e} + \lambda e\right)}_{feedback}\right] \tag{26}$$

Here, the disturbance error must be estimated using the adaptation mechanism. Our adaptation law is designed as follows:

$$\Delta\dot{\hat{d}} = K_a\left(\dot{e} + \lambda e\right),\ K_a > 0 \tag{27}$$

where, K_a is a tuning parameter of the adaptation algorithm.

The control law (26) and the adaptation law (27) guarantee the stability of the system. Let's define a Lyapunov function,

$$V = \frac{1}{2}S^2 + \frac{1}{2K_a}\left(\Delta d - \Delta\hat{d}\right)^2 > 0,\ K_a > 0 \tag{28}$$

The derivative of (28) is as follows:

$$\dot{V} = S\dot{S} + \frac{1}{K_a}\left(\Delta d - \Delta\hat{d}\right)\left(\Delta\dot{d} - \Delta\dot{\hat{d}}\right) \tag{29}$$

The disturbance error, Δd assumes that it is varying slowly because the nominal value, d_n is tracking the actual disturbance, d. Therefore, the derivative of Δd is nearly equal to zero, i.e., $\Delta\dot{d} \approx 0$ and (29) is recalculated using (23), (26) and (27) as follows:

$$\dot{V} = S\dot{S} - \frac{1}{K_a}\left(\Delta d - \Delta\hat{d}\right)\Delta\dot{\hat{d}} = -K_s S^2 \le 0 \tag{30}$$

Therefore, it proves that the system is stable. Using LaSalle's invariant set theorem, the asymptotical stability of the system is also proved. If $e = 0$, only $e = 0$ makes $S = 0$ in (23) and $S = 0$ makes $\dot{S} = 0$. Substituting (26) into (22), above relation makes $\Delta d - \Delta\hat{d} = 0$. Therefore, the equilibrium point and $\Delta d - \Delta\hat{d}$ are asymptotically stable.

3.3 Split Friction Control

In ABS, the large pressure difference between left and right-side wheels makes a vehicle be unstable. It causes spin-out of the vehicle or makes controllability of a vehicle loss. Although the road friction coefficients of both sides are different, the difference of the applied pressures must have some limit condition. The wheel pressure sensor is very useful to apply the limitation because the wheel pressures are measured directly.

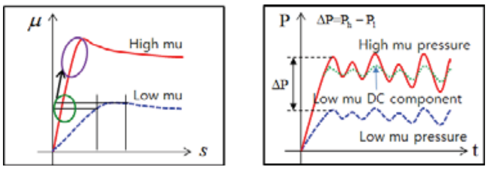

Fig. 5 Mu-Slip Curve and Split-Mu Pressure Control

Table 1 Simulation Case Specification

Case1	100kph	Friction transition high($\mu = 0.8$) to low($\mu = 0.2$)
Case2	100kph	Split friction left($\mu = 0.8$) - right($\mu = 0.2$)

Table 2 Braking Distance

	Without pressure sensors	With pressure sensors
Case1	67.83m	65.41m
Case2	101.25m	96.70m

It makes the vehicle more stable on a split-friction road. The proposed control method is as follows:

$$V_{cmd} = \begin{cases} LPF\left[V_{Lcmd}\right] + V_{apply-Hcmd} & if\ P_h - P_l > P_{th} \\ V_{cmd} & otherwise \end{cases} \tag{31}$$

where, V_{cmd} is a valve command, V_{Lcmd} the valve command of low pressure wheel, $V_{apply-Hcmd}$ the valve apply command of high pressure wheel, P_h the pressure of high pressure wheel, P_l the pressure of low pressure wheel, P_{th} the threshold of pressure difference, and LPF the low pass filtering.

In (31), the valve command of the high pressure wheel is recalculated using dc component of low pressure wheel with apply command of high pressure wheel. It makes the valve command of high pressure wheel be to work like valve command of low pressure wheel with additional pressure margin. In case of rear wheels, the threshold must be lower than that of the front wheel to prevent the vehicle from being spin-out. Using this scheme, the vehicle can be stable under the conditions which have large difference between left and right-side friction such as driving on the split-friction road and full braking on the cornering driving. The description is shown in Fig. 5.

4. SIMULATION

A series of simulation has been performed to verify the performance of the proposed ABS algorithm under a variety of conditions using CarSim program. They are standard ABS test conditions consisting of a variety of road friction coefficients and vehicle velocities. To show the contribution of this study, two controllers are compared through a variety of simulation tests. They are shown in Table 2. The proposed algorithm is compared with one without four wheel pressure sensors in [11].

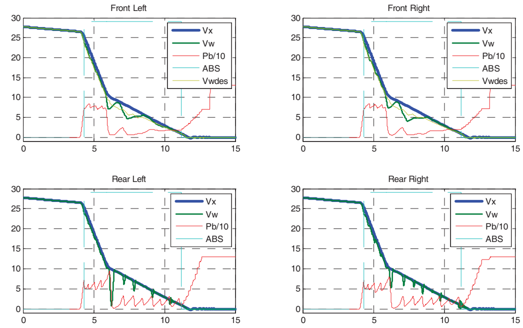

Fig. 6 Case1. High To Low: without Pressure Sensors

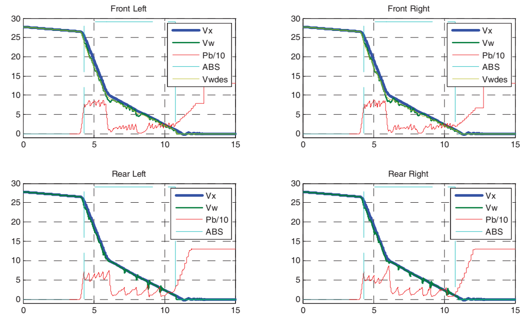

Fig. 7. Case1. High To Low: with Pressure Sensors

The results of the ABS control without wheel pressure sensors are shown in Fig. 6 and 8. And the results of the proposed control are shown in Fig. 7 and 9. In figures, two subfigures at the top represent front left and right-side wheels, and two subfigures at the bottom represent rear left and right-side wheels. The steering wheel angles for case 2 are compared in Fig. 10.

In case of the algorithm without wheel pressure sensors, the road friction coefficient cannot be estimated, so the detection of the friction variation such as friction transition(Case1) and split-friction can be slow or cannot be detected properly. At the moment changing the road friction coefficient from high to low mu in Fig. 6 and 7, the proposed algorithm shows that the wheel slip is smaller than that of the algorithm without pressure sensors because of fast detection of surface friction transition. In case of driving on the split-friction road in Fig. 8 and 9, although the pressure difference between left and right-side wheels and the steering angle are similar to both algorithms as shown in Fig. 10, the braking distance of the proposed algorithm was shortened. The results of the braking distance are compared in Table 3. From all simulation results, the proposed algorithm shows the improved performance when it is compared with the algorithm without pressure sensors in terms of the similar stability of the vehicle and shorter braking distances

5. CONCLUSION

This study has proposed a new Anti-lock Braking System control algorithm using four wheel pressure sen-

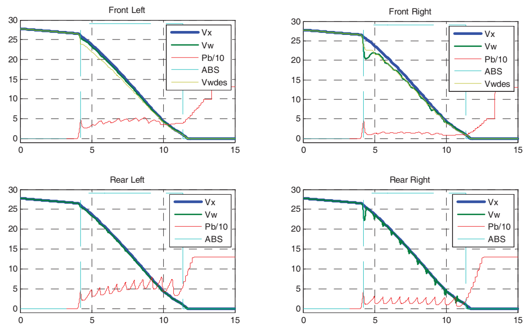

Fig. 8 Case2. Split Mu: without Pressure Sensors

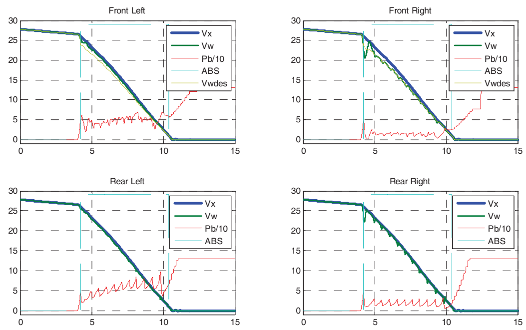

Fig. 9. Case2. Split Mu: with Pressure Sensors

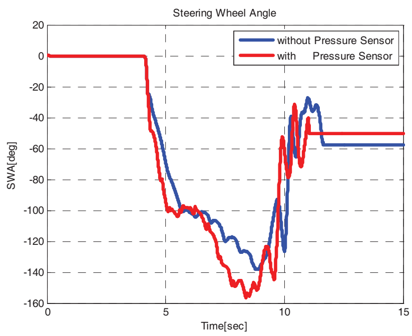

Fig. 10. Steering Wheel Angle Comparison in Case2

sors. The proposed control scheme is considered as a semi-model based control system because the combination of the rule-based and model-based controller is used. The main issue of the model-based control system in ABS has been improved by using the estimated road friction coefficient. Therefore, it is considered as a more practical ABS control algorithm. The estimated road friction coefficient enables all wheels to be controlled under each estimated road friction coefficient. It guarantees the lateral stability of the vehicle on the split friction road condition making the braking distance shorter. The rear wheel brakes are controlled by a semi-feedback controller, and the front ones by an adaptive sliding mode controller. The simulation results show high performance under a variety of road conditions. The braking distance has been shortened while keeping the vehicle stable. However, the proposed system shows some chattering of the front wheel pressure. It must be improved in the future work because it can make the rid harsh on high-friction surfaces.

REFERENCES

[1] Savaresi, S., Tanelli, M. and Cantoni, C., "Mixed Slip-Deceleration Control in Automotive Braking Systems", Journal of Dynamic system, Measurement, and Control, vol. 129, pp. 20-31, 2007.

[2] SAE, "Antilock Brake System Review", Society of Automotive Engineers, pp. 90-102, 1992.

[3] Wellstead, P., and Petit, B., "Analysis and Redesign of an Antilock Brake System Controller", IEEE Proc. Control Theory and Applications, vol. 144, no. 5, pp. 413-326, 1997

[4] Yu, J., "A Robust Adaptive Wheel-Slip Controller for Antilock Brake System", Proc. of the 36th IEEE Conference on Decision and Control, vol. 3, pp. 2545-2546, 1997

[5] Petersen, I. et al., "Wheel Slip Control in ABS Brakes Using Gain Scheduled Constrained LQR", European Control Conference, pp. 146-155, 2001

[6] Buckholtz, K., "Reference Input Wheel Slip Tracking Using Sliding Mode Control", Society of Automotive Engineers, no. 2002-01-0301, 2002

[7] Johansen, T. et. al., "Gain-scheduled Wheel Slip Control in Automotive Brake Systems", IEEE Trans. On Control System Technology, vol. 11, no. 6, pp. 799-811, 2003

[8] Solyom, S., Rantzer, A., and Ludemann, J., "Synthesis of a Model-Based Tire Slip Controller", Vehicle System Dynamics, vol. 41, no. 6, pp. 475-499, 2004

[9] Mirzaeinejad, H., and Mirzaei, M., "A Novel Method for Non-linear Control of Wheel Slip in Anti-lock Braking Systems", Control Engineering Practice, vol. 18, no. 8, pp. 918-926, 2010

[10] Cho, K., et. al., "Design of an ABS Control Algorithm Using Wheel Dynamics", Korean Society of Automotive Engineers 2010 Annual Conference, pp. 1756-1761, 2010. (in Korean)

[11] Choi, S., "Antilock Brake System With a Continuous Wheel Slip Control to Maximize the Braking Performance and the Ride Quality", IEEE Trans. on Control Systems Technology, vol. 16, no. 5, pp. 996-1003, 2002.

ACKNOWLEDGE

This work was supported by the National Research Foundation of Korea (NRF) grant funded by the Korea government(MEST) (No.2012-0000991).