Research Report Application of Hybrid Control Method to Braking Control System with Estimation of Tire Force Characteristics Yuji Muragishi, Eiichi Ono

Abstract

To improve the performance of a vehicle's braking control, it is important to be able to estimate the friction force characteristics between the vehicle's tires and the road. Over the last few years, we have proposed a method that allows us to estimate the slope of the friction force against the slip velocity at the operation point, based on the fluctuation of the wheel velocity as a parameter that corresponds to the margin of the friction force between the tires and the road. Moreover, we have proposed a braking control strategy that is based on estimating the parameter for achieving the maximum braking force. Although this produced good braking control, valve switching noise became a problem. In addition, a pressure sensor is needed to control the pressure. This is not needed in conventional braking systems.

This study addresses the application of hybrid control to braking to ensure that braking performance is maintained while reducing the valve switching frequency. Moreover, a control system design that does not need the abovementioned pressure sensor is proposed. This design incorporates a mathematical model that estimates the brake pressure and then uses the estimated values. Tests with an actual vehicle verified that the valve switching frequency is reduced, and that good control is realized despite the absence of a pressure sensor.

Keywords Brake, Friction, Tire, Hybrid control, Estimation, Vehicle test

1. Introduction

The friction force characteristics of vehicle tires change depending on the driving conditions. As a result, therefore, a robust control approach that treats changes in the tire characteristics as plant perturbations provides an effective method of vehicle control.¹⁾ To optimize the performance of vehicles, however, it is necessary to estimate the friction force characteristics of their tires. There are several models for describing the friction force characteristics,^{2, 3)} but estimating the parameters for these models, using on-line identification methods, is inherently difficult.

We have proposed a method of estimating the slope of the friction force against the slip velocity at the operational point, or Extended Braking Stiffness (hereafter referred to as "XBS") as an important parameter that describes the friction force of the tire as shown in **Fig. 1**.⁴⁾ The maximum braking force can be obtained at the point where XBS = 0, with a lower XBS indicating a reduction in the margin of the friction force. Moreover, we have also proposed a braking control strategy that is based on an estimate of XBS.⁴⁾ However, although the proposed braking control proved very effective in achieving the maximum braking force, the valve switching frequency increased which resulted in noise problems. Also, a pressure sensor was needed to control the wheel cylinder pressure. This is not necessary in conventional braking systems.



Fig. 1 Tire/road characteristics and extended braking stiffness (XBS).

We attempted to apply the hybrid control method to braking control to provide optimum braking performance while reducing the valve switching frequency. Hybrid control is attracting attention as a means of controlling systems that involve a parameter that changes continuously as a result of changes in a discrete state.⁵⁻⁷⁾ Moreover, we created a model that estimates the behavior of the braking pressure as a result of valve switching and proposes a pressure control method that relies on this model and dispenses with the pressure sensor.

The following sections describe a method of estimating XBS from the wheel velocity, a corresponding braking control strategy, and the application of hybrid control to braking control. We also present the results of actual experiments.

2. Application to braking control system

2. 1 Estimation of Extended Braking Stiffness (XBS)

A pneumatic tire of a vehicle has a rotational resonance that arises from the wheel's inertia and the sidewall's spring.⁸⁾ However, this resonance vanishes when the driver brakes because of the friction caused by the brake pads. In this case, the rotational dynamics of the wheel (see **Fig. 2**) will be as follows:

$$J \dot{v}_w = r^2 F_x - r T + r^2 d$$
,(1)

where, J is moment of inertia of the wheel, r is the radius of the wheel, F_x is the friction reaction force



Fig. 2 Rotational dynamics of the wheel.

between the tire and the road, and *T* is the braking torque (which is proportional to the pressure in the wheel cylinder), *d* is the disturbance from the road, and v_w is the wheel velocity.

By assuming that the vehicle dynamics are considerably slower than those of the wheel and that F_x is a function of the slip velocity, a wheel deceleration model can be obtained from Eq. (1).

Here, k is the Extended Braking Stiffness (XBS), w is the disturbance arising from the road and fluctuations in the braking torque ($w = r^2 \dot{d} - r \dot{T}$).

If we assume constant deceleration braking, for example, and μ -peak braking on a constant μ road, the braking torque *T* can be treated as a disturbance by differentiating Eq. (1). This implies that XBS can be estimated from the wheel velocity. We do not have to know the pressure in the wheel cylinder. Equation (2) describes the dynamics of wheel deceleration, and XBS is proportional to the breakpoint frequency of the wheel deceleration model. Then, XBS can be estimated by identifying the breakpoint frequency of Eq. (2).

Figure 3 shows the results of experiments we performed to obtain the wheel velocity frequency characteristics during braking. The magnitude of the power spectrum density at low frequencies increases



Fig. 3 Experimental results of the frequency characteristics of wheel velocity with constant wheel cylinder pressure on a packed snow road.

according to the increase in the wheel cylinder pressure, while the breakpoint frequency shifts to the left (low frequency). This indicates that XBS decreases according to the decrease in the margin of the friction force.

By assuming that *w* is white noise, XBS *k* can be estimated by applying the least square method⁹⁾ to Eq. (2), as follows:

$$v[i] = -v[i] + 2v[i-1] - v[i-2]$$
(4)

.....(7)

Here, τ is the sampling time, v is the filtered (2 to 20 Hz band pass) wheel velocity, \hat{k} is the estimated XBS, and λ is the forgetting factor.

The algorithm described by Eq. (3) to Eq. (7) estimates XBS from the fluctuation in the wheel velocity. **Figure 4** shows XBS estimated by Eq. (3) to Eq. (7) using experimental results. XBS falls as a



Fig. 4 Estimated XBS by Eq. (3) - Eq. (7) of experimental results on a packed snow road.

result of hard braking.

Figure 5 shows the relationship between the average XBS during braking and the wheel cylinder pressure. In line with the fall in the margin of the friction force on each road surface, the estimated XBS also decreases. This implies that the maximum braking force on each road surface can be obtained from XBS servo control, i. e., the actuation of the wheel cylinder pressure which minimizes the estimated value of XBS.

2.2 Control system structure

The estimated XBS can be applied to braking control. We propose a method of braking control that obtains a constant μ rate to improve the vehicle braking and steering. A control system that observes the reference value of XBS (XBS servo control) is realized by adopting the three-layered hierarchy



Fig. 5 Relation between averaged XBS during braking and the wheel cylinder pressure.



Fig. 6 Control system structure of XBS servo.

control shown in Fig. 6. To follow the reference value of XBS, the XBS servo calculates a reference value for the wheel deceleration, the deceleration servo calculates a reference value for the pressure in the wheel cylinder, and the brake servo calculates the valve switching command. Since XBS is estimated based on the difference in the frequency characteristics of the wheel velocity as shown in Fig. 3, a faster estimate than the breakpoint frequency of the Wheel Deceleration Model (2 to 20 Hz) cannot be expected. This implies that the estimation delay is too large to use for feedback control in wheel motion stabilization. Therefore, for this study, we adopted a control system that features a deceleration servo. The deceleration servo stabilizes the wheel motion and follows the reference value corresponding to the estimated XBS value. The bibliography⁴ gives full details of how the XBS reference value was set up.

2.3 Brake servo control

Brake servo control involves performing calculation to ensure that the valve switching command follows the pressure in the wheel cylinder as calculated for deceleration servo control. A brake system controls the pressure within a wheel cylinder by switching between three states, namely, pressure increase, pressure hold, and pressure reduction, by using two on-off type solenoid valves (see **Fig. 7**). Conventional brake servo control⁴) uses the switching commands shown in **Table 1** according to the difference in the reference value of the pressure and the wheel cylinder pressure as detected by the sensor.

A hybrid control method can be applied to brake

servo control. First, the sampling time of the valve switching is determined, and the candidates for the valve switching patterns of the next N steps are given as shown in **Table 2**. In this study, the sampling time of the valve switching was set to 0.8 ms and N was set to 4. Then, the wheel cylinder pressures corresponding to the candidate valve switching patterns were predicted for the next N steps by using mathematical pressure models. The valve switching pattern, which is controlled by brake servo commands, is determined according to the following performance index from the candidates in Table 2.

$$J = \sum_{k=0}^{N} \widehat{e}^{T}(k) R(k) \widehat{e}(k) + \sum_{k=0}^{N-1} SP(g(k), g(k-1)),$$
.....(8)

where R is the weighting function, \hat{e} is the difference between the reference pressure and the estimated wheel cylinder pressure, and SP is the penalty



Fig. 7 Structure of brake system.

Table 1	Valve switching commands by the conventional br	ake
	servo control.	

Difference pressure	Valve switching command		
$\begin{array}{c} P_{b} - P_{b0} > 2P_{0} \\ P_{0} < P_{b} - P_{b0} \leq 2P_{0} \\ -P_{0} < P_{b} - P_{b0} \leq P_{0} \\ -2P_{0} < P_{b} - P_{b0} \leq -P_{0} \\ P_{b} - P_{b0} \leq -2P_{0} \end{array}$	Quick pressure reduction Slow pressure reduction Pressure hold Slow pressure increase Quick pressure increase		
0 00 0			

 P_b : measured pressure of the wheel cylinder P_{b0} : reference value of pressure P_0 : threshold value of control

for valve switching, as follows:

$$SP = \begin{cases} 0 \ when \ g(k) = g(k-1) \\ 1 \ when \ g(k) \neq g(k-1), \cdots (9) \end{cases}$$

and g is the discrete state of the braking system, i.e.,

$$g(k) = \begin{cases} I & when \ pressure \ increase \\ H & when \ pressure \ hold \\ R & when \ pressure \ reduction \end{cases}$$
(10)

Assuming that the reference value for the pressure remains constant for the next N steps, the

performance indices corresponding to the candidate switching patterns are calculated, and the valve switching pattern with which evaluation acts as a minimum is adopted. The valve switching patterns shown in Table 2 are chosen based on the following: [1] The pattern does not have to be evaluated clearly, for example, pressure reduction patterns are excluded when a pressure increase is required; [2] Valve switching from pressure reduction to pressure increase or from pressure increase to pressure reduction is not performed; [3] The pressure increase or pressure reduction is forced to continue for more than a certain number of steps in line with the response of the solenoid valve; [4] The number of times valve switching occurs in a series of patterns is set to a maximum of two times.

The pressure in the wheel cylinder for each of the next N steps is estimated from the current pressure estimate and the valve switching patterns for the next N steps. The flow rate through the solenoid valve is modeled as follows:

$$Q_{vi} = (CA)_i \sqrt{\frac{2}{\rho} \Delta P} \quad (i = A, B),$$
.....(11)

where, $(CA)_i$ is the product of the coefficient of discharge and the area of the valve opening, ρ is the density of the fluid, and ΔP is the pressure difference between the upstream and downstream sides of the valve. Subscripts A and B

refer to Valve A and Valve B as shown in Fig. 7. It is assumed that the opening and closing of the solenoid valve are performed in no more than 2 ms after the issue of a command.

The behavior of the pressure in the wheel cylinder is modeled as follows. This model does not consider the influence of the pipe-line on the front face, but does consider that of the pipe-line and the secondorder lag element for the rear face.

Front face:
$$\frac{\delta P_w}{Q_v} = K_m$$
(12)

Rear face:
$$\frac{\delta P_w}{Q_v} = \frac{K_m \omega_n^2}{s^2 + 2\zeta \omega_n s + \omega_n^2}$$
(13)

Here, δP_w is the pressure behavior around unit time, Q_v is the flow rate to the wheel cylinder (= $Q_{vA} - Q_{vB}$), K_m is a constant value, *s* is the Laplace operator, ζ is the damping ratio of the pipe-line, and ω_n is the natural frequency of the pipe-line.

The pressure in the wheel cylinder is estimated using Eq. (11) to Eq. (13) while considering the master cylinder pressure detected by the pressure sensor, with the valve switching commands as input. Brake servo control, which does not require a pressure sensor in the wheel cylinder, is realized by providing control that causes the estimated pressure to follow the reference pressure values.

3. Results of experiments

Figure 8 shows the results of the experiments we performed, involving straight-line braking on an icecovered road with the XBS servo and with the application of hybrid control and conventional brake servo control.⁴⁾ The pressure waveform shown in Fig. 8(a) indicates that the estimation of the wheel cylinder pressure by Eq. (10) to Eq. (12) is realized successfully. Moreover, when hybrid control is applied to the brake servo control, almost the same level of performance can be realized relative to conventional brake servo control, as shown in Figs. 8(a) and 8(b).

Table 3 shows the number of times that valve switching occurs for the different brake servo control methods used in the experiment, as shown in Fig. 8. These results show that the application of the hybrid control method realizes a reduction in the valve switching frequency, while maintaining

Table 2 Valve switching patterns evaluated in hybrid control.

Pattern No.	k	g(-1)	$\left \widehat{e} \right \leq e_0$	$\widehat{e} > e_0$	\widehat{e} < - e_0				
	0 1 2 3 4			When the last pattern is No. 5					
1	IIIII			Pattern No.1 and No.6-8					
2	HIIII	Ι	Pattern No.16	Other	Pattern No.11 and No.16				
3	ННІІІ			Pattern No.1 and No.6-9					
4	НННІІ			When the last pattern is No. 6 or No.12	When the last pattern is No. 6 or No.12				
5	ННННІ		▲	Pattern No.2-5 and No.16	Pattern No.11 and No.16				
6	ІІІН	Н		Other	Other				
7	ІІІНН		· ·	Pattern No.1-5 and No.16	Pattern No.10-11 and No.16				
8	ІІННН								
9	ІНННН	D	♠						
10	RRRRR	K		Pattern No.1-5 and No.16	Pattern No.10 and No.12-16				
11	HRRRR		,						
12	RRRH								
13	RRRHH	I : Pressure increase e_0 : threshold of control							
14	RRHHH	H : Pressure hold R : Pressure reduction							
15	R H H H H								
16	ННННН								

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braking performance.

Figure 9 shows the results of experiments with straight-line braking on an ice-covered road with conventional ABS. If we compare Fig. 8 and Fig. 9, we can see that fluctuations in the wheel velocity and the pressure in the wheel cylinder are suppressed by the XBS servo and that a larger friction force is obtained than with conventional

ABS.

The XBS servo produces a μ peak that conforms to the control even if the maximum friction coefficient μ changes. We evaluated the ability of the XBS servo to adapt to changes in the road friction characteristics. **Figure 10** shows the results of our experiments with the XBS servo involving changes in the road friction characteristics from an



Fig. 8 Experimental results of XBS servo on an ice-covered road.(a) brake servo control using the hybrid control. (b) brake servo control using the conventional brake control.

Table 3 Comparison of the number of times of valve switching by $0.5 \sim 2.5$ seconds after a braking start.

	FR	FL	RR	RL
Hybrid control	267 ~ 275	269 ~281	77 ~ 79	68 ~ 83
Conventional brake control	327 ~ 384	345 ~461	238 ~ 300	222 ~ 262





Fig. 9 Experimental results of conventional ABS on an ice-covered road.

Fig. 11 Experimental results of conventional ABS during change of road friction characteristics from an ice-covered road to a dry asphalt road.



Fig. 10 Experimental results of XBS servo during change of road friction characteristics from an icecovered road to a dry asphalt road. (a) brake servo control using the hybrid control. (b) brake servo control using the conventional brake control.

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ice-covered road to a dry asphalt road.

When the vehicle transitions from an ice-covered road to a dry asphalt road, the estimated value of XBS increases according to the increase in the margin of the friction force. Then, the XBS servo increases the wheel cylinder pressure more rapidly than the conventional ABS, as shown in **Fig. 11**.

4. Conclusion

The XBS parameter plays an important role in identifying tire/road friction characteristics. We applied the hybrid control method to braking control based on XBS with the goal of reducing the valve switching frequency relative to conventional brake control. Moreover, we proposed a method of brake control that does not use a pressure sensor in the shape of a model that estimates the pressure in the wheel cylinder. Tests on an actual vehicle verified that the valve switching frequency is reduced, and that a good control performance could be realized without the use of a pressure sensor.

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Year of Born : 1968 Division : Vehicle Control Lab. Research fields : Automotive fluid power system

Academic society : Jpn. Soc. of Mech. Eng., Jpn. Fluid Power Syst. Soc.



Eiichi Ono

Yuji Muragishi

Year of Born : 1963 Division : Vehicle Control Lab. Research fields : Vehicle dynamics & control Academic degree : Dr. Eng. Academic society : Soc. of Instrum. and Control Eng., Soc. of Autom. Eng.

of Jpn. Awards : 1995 SICE Award for Outstanding Paper, 1999 SICE Chubu Chapter Award for Outstanding Research, 2002 SICE Chubu Chapter Award for Outstanding Technology, 2002 IFAC Congress Paper Prize, 2002 Paper Award of AVEC