EVS27
Barcelona, Spain, November 17-20, 2013

Design of Fail-Safe Controller for Brake-By-Wire Systems Using Optimal Braking Force Distribution

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Abstract

In this paper, a method for mitigating the risk of potential brake actuator failure during braking in vehicles with BBW(Brake-By-Wire) systems is described. When the failure of one brake actuator occures during braking, the vehicle cannot stop as quickly as driver wants and the vehicle may fishtail because of imbalance braking force. To avoid this dangerous situation, a novel fail-safe control algorithm is proposed. The proposed fail-safe control algorithm consists of 2 steps. In the first step, corrective yaw moment is determined to manage the undesired yaw moment caused by asymmetric braking. In the second step, the additional braking forces are re-distributed to normally remaining actuators using optimal design methods such as Lagrange multiplier and KKT (Karush-Kuhn-Tucker) condition. The simulation results demonstrate that the proposed control algorithm can make the vehicle with a failed brake actuator follow the desired deceleration more closely and achieve the remarkable reduction in undesired yaw moment. To validate the proposed fail-safe controller, simulations on the well-known vehicle simulation software, Carsim , were conducted.

Keywords : EMB(Electro-Mechanical Brake),EWB(Electric-Wedge Brake), Fail safe, Sliding Mode Control, KKT condition

1 Introduction

The Brake-By-Wire systems for motor vehicles such as, EV and FCEV and so on, have been receiving a lot of interesting in the automotive industry because of their flexibility and quick response time in controlling vehicle motion [1]. The BBW systems are introduced and developed in some previous researches [2]. In these researches, the models of BBW system and control algorithms are established. The BBW systems are including some ECUs to control the motor which generate the brake torque pressing the brake pad to disc instead of hydraulic actuator.
Meanwhile, it is global trend that the safety and reliability of some automotive parts which contain the electric/electronic parts like the BBW system should be guaranteed by a credible standard such as ISO 26262 [3]. This is one of the reasons why the studies about fault tolerant control algorithms applied in BBW system have been conducted in various research institutes [4].
In this paper, a novel fail-safe control algorithm is developed for a situation that one brake actuator failure occurs during straight-line braking. In this case, the additional braking forces should be determined and distributed by ESC controller

using some algorithms [5][6]. In order to determine and distribute the additional braking forces, the proposed fail-safe control algorithm consists of 2 steps. In the first step, corrective yaw moment is determined using sliding mode control method to minimize the undesired yaw moment caused by asymmetric braking. In the second step, the additional braking forces are distributed to normally remaining actuators using optimum design methods such as Lagrange multiplier and KKT (Karush-Kuhn-Tucker) condition [7][8].

The proposed fail-safe control algorithm is designed with three considerations. First is that the vehicle has to be controlled to reduce the undesired yaw moment. Second is that the vehicle has to stop following the desired deceleration as the driver wants. The last is that the total braking force should not exceed the maximum capacity of tire friction force to avoid the tire locking and remain a possibility that the driver can control the vehicle operating the steering wheel under brake actuator failure.

2 Models of BBW systems

The target BBW system of this research is composed of EWB and EMB. The self-energizing mechanism of the EWB allows it to be used on the front axle of the target vehicle. Then the EMB is applied on the rear axle of the target vehicle. The brake system architecture of the target vehicle is depicted in Fig. 1.

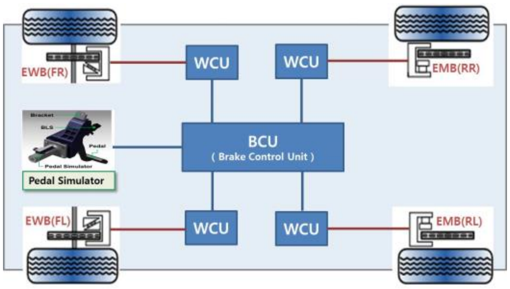


Figure 1 : BBW system Architecture

2.1 Model of EWB system

The model of EWB system is introduced in several previous researches [1],[9]. In Fig. 2, the schematic diagram of the EWB system is shown to represent the self-energizing mechanism. The main equations are as follows from equation (1) to (8).

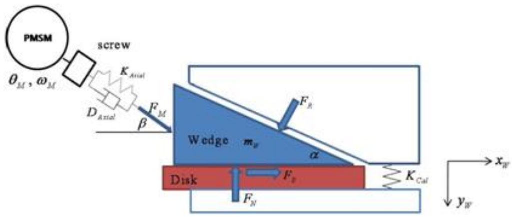


Figure 2: Schematic of EWB system

$$m_W \ddot{x} = F_M \cos\beta + F_B - F_R \sin\alpha \quad (1)$$

$$m_W \ddot{y} = F_M \sin\beta - F_N + F_R \cos\alpha \quad (2)$$

$$y = \tan\alpha \cdot x \quad (3)$$

$$2\pi\eta T_L = LF_L \quad (4)$$

$$F_B = \mu F_N \quad (5)$$

$$L_M \dot{i}_M = -R_M i_M - K_e \omega_M + u_M \quad (6)$$

$$T_M = K_e \omega_M \quad (7)$$

$$F_M = -K_{axial}\left(\frac{X_w}{cos\beta} - \frac{L}{2\pi}\theta_M\right) - D_{axial}\left(\frac{X_w}{cos\beta} - \frac{L}{2\pi}w_M\right) \quad (8)$$

Where, m_W is the mass of a wedge, x and y are positions of a wedge in local coordination, F_M is the force due to an actuating motor, F_B is a braking force, F_R is a reaction force on the inclined plane of the wedge, α is an angle between the best side and the inclined plane of the wedge and β is an angle of the force direction developed by an actuator motor. η is efficiency of actuator and screw, L is a pitch of a screw, F_N is a normal force. L_M and R_M are the reluctance and the resistance of the motor, i_M is a current that drives the motor, K_e is a motor characteristic torque constant, u_M is a control voltage of the motor.

2.2 Model of EMB system

The model of EMB system is also introduced in several previous researches [1],[9]. In Fig. 3, the schematic diagram of the EMB system is shown to represent the brake force generating mechanism. The main equations are as follows from (9) to (12).

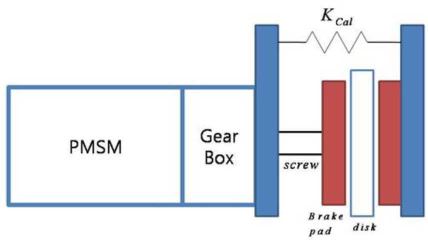


Figure 3 : Schematic of EMB system

$$J_{sys}\dot{\omega}_M = T_M - T_L \tag{9}$$

$$T_L = L/2\pi F_N \tag{10}$$

$$F_B = \mu F_N \tag{11}$$

$$L_M \dot{i}_M = -R_M i_M - K_e \omega_M + u_M \tag{12}$$

The control algorithm of EWB and EMB systems was designed using a sliding mode control law [9]. The developed BBW system models and control algorithms are implemented in the Matlab/Simulink environment.

2.3 Validation of BBW System Models

To validate the models of BBW system, the prototypes of EWB system and EMB system are tested in brake dynamometer. The brake torque can be measured in brake dynamometer. In the Fig. 4 and 5, the results of BBW system model validation are depicted. The resultant braking torques of the simulations are calculated from some specifications of brake system such as the resultant clamping force, disc diameter and friction coefficient between brake pad and disc.

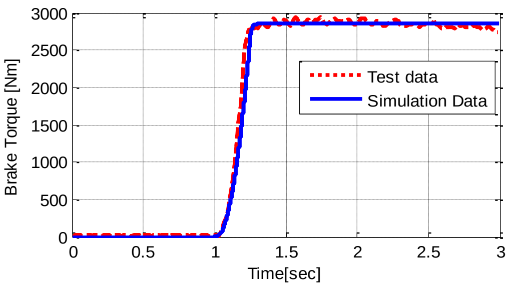


Figure 4 : Validation of EWB model

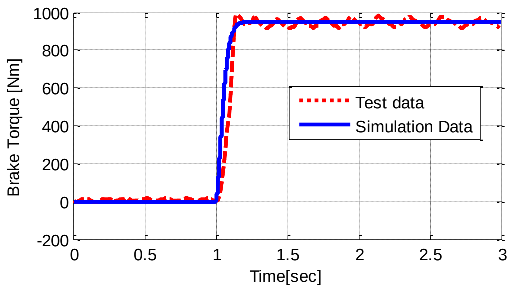


Figure 5 : Validation of EMB model

The commanded clamping forces of the EWB and EMB are 35kN and 10kN, respectively. The step inputs of clamping force are commanded at 1 sec in Fig. 4 and 5. The braking torques of the EWB model and prototype of EWB can reach to steady state in 0.25 sec after clamping force command. And the brake torques of the EMB model and prototype of EMB can reach to steady state in 0.15 sec after clamping force command. As shown in Fig. 4 and 5, the models of BBW system can represent the actual prototypes of BBW systems properly.

3 Fail-safe Controller Design

The main tack of the fail-safe controller considered in this paper is to mitigate the effects of the possible single brake actuator failure on the vehicle motion by distributing the braking force to normally remaining 3 tires. In this section, the brake actuator failure of front right wheel will be considered. The braking force capacity of front axle is greater than rear axle so, the front actuator failure is more fatal than the rear actuator failure.

The fail-safe controller is composed of two steps. First step is to calculate the yaw moment to eliminate the yaw motion induced by imbalance braking force between left wheels and right wheels in the case of front right actuator failure. Then, second step is to distribute the additional braking force to normally remaining 3 wheels to generate the yaw moment calculated in first step and to follow the desired deceleration determined from the brake pedal simulator. In second step, the KKT condition of optimum design method is applied to carry out the optimal solution [8][10].

3.1 Yaw Moment Calculation (1st Step)

In this section, the corrective yaw moment is calculated to eliminate the effect of imbalance brake force generated from the front right actuator failure. Using the planar vehicle model shown in Fig. 6, the corrective yaw moment is determined using the sliding mode control with zero steering wheel input.

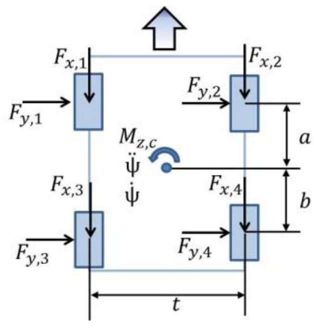


Figure 6 : Planar Vehicle Model

The yaw dynamics of planar model can be expressed in equation (13).

$$I\ddot{\psi} = \frac{1}{2}t\left(F_{x,1} - F_{x,2}\right) + \frac{1}{2}t\left(F_{x,3} - F_{x,4}\right) \\ -a\left(F_{y,1} + F_{y,2}\right) + b\left(F_{y,3} + F_{y,4}\right) + M_{z,c} \quad (13)$$

The lateral forces may be too small and neglected in straight-line braking, so $F_{y,i} = 0$. The sliding surface is defined as a yaw rate error such as equation (14).

$$S = \dot{\psi} - \dot{\psi}_{des} \quad (14)$$

In the case of straight-line braking situation, the desired yaw rate is zero. The sliding control law is defined as equation (15) .

$$\dot{S} = -\eta S \quad (15)$$
$$\dot{S} = \ddot{\psi} - \ddot{\psi}_{des} = -\eta\left(\dot{\psi} - \dot{\psi}_{des}\right) \quad (16)$$

When the front right actuator is failed, $F_{x,2} = 0$. And the corrective yaw moment can be expressed in equation (17) by substituting equation (13) to equation (16).

$$M_{z,c} = -\frac{t}{2}\left(F_{x,1} - F'_2 + F_{x,3} - F_{x,4}\right) - \eta I\dot{\psi} \quad (17)$$

Note that F'_2 is the braking force generated from the regenerative braking system. This force remains on the front right tire under the brake actuator failure.

3.2 Brake Force Distribution (2nd Step)

The objective of control algorithm suggested in this section is to distribute the additional braking force to normally remaining 3 wheels to follow the desired deceleration as closely as possible and to generate the corrective yaw moment derived in last section. To define this problem in numerical equations, the formulations of optimum design problem are cited. Then, to solve this formulated optimum design problem, the Lagrange Multipliers and KKT optimal conditions are used. The formulations of optimum design problem for finding the additional braking forces are as follows :

Find : x_1 , x_3 , x_4

Minimize : $f(x) = -(x_1^2 + x_3^2 + x_4^2)$ (18)

Subject to :

$$h(x) = \frac{t}{2}(x_1 + x_3 - x_4) - M_{z,c} = 0 \quad (19)$$

$$g_1(x) = \beta a_{x,des} - a_x \leq 0 \\ = \beta a_{x,des} - \frac{1}{m}\{(F_1 + x_1) + F_2' + \\ (F_3 + x_3) + (F_4 + x_4)\} \leq 0 \quad (20)$$

$$g_2(x) = (F_4 + x_4) - \mu_{max} N_4 \leq 0 \\ = (F_4 + x_4) - \mu_{max}\left(\frac{a}{2l}\right) mg \leq 0 \quad (21)$$

x_i in equation (18) to (21) are the additional braking force distributed to the normally remaining tires, respectively. And F_i are the normally generated braking forces of each tire without fail-safe controller, respectively. F_i can be estimated easily using the information about desired deceleration and the ratio of brake force distribution between the front and rear axle. Then, $(F_i + x_i)$ are entire braking forces when the fail-safe controller is activated. x_i and F_i are represented schematically in Fig. 7.

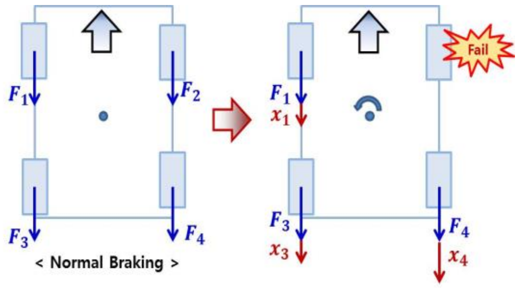


Figure 7 : Braking Forces of Fail-safe Controller

The equation (18) means that capacities of additional braking forces have to be maximized to use the limit of tire force when one brake actuator doesn't work. The equation (19) means that yaw moment induced by the additional braking force is equal to corrective yaw moment calculated in section 3.1. The equation (20) means that the braking deceleration under the brake actuator failure has to be higher than $\beta \times a_{x,des}$. $a_{x,des}$ is the desired deceleration determined from the driver's intention measured in brake pedal simulator.

The range of β is zero to one, ordinarily 0.6~1. When the total braking force including the additional braking forces of normal 3 wheels can generate the desired deceleration, β is set to be one. Suppose that the driver steps on the brake pedal very quickly and the desired deceleration is higher than 0.8g. In this case, the normal 3 actuators with fail-safe controller can't generate the desired deceleration higher than 0.8g, then β is set to be

about 0.6 to avoid the tire locking which can make the vehicle unstable. Intuitionally, if β becomes larger, the potential margin for controlling yaw moment becomes lower. On the contrary, β becomes lower, the potential margin for controlling yaw moment becomes larger.
The maximum value of additional braking force of rear right tire is limited in equation (21). N_4 denotes the vertical force of rear right tire considering longitudinal load transfer. The additional braking force of the other tires, such as front left and rear left tire, may not reach to the maximum friction force, so the equation (21) is sufficient for limiting the capacity of additional braking force.
To solve the above constrained optimum design problem, Lagrange multiplier theorem and Karush-Kuhn-Tucker(KKT) optimality condition theorem were used. Lagrange function for the above equations is defined in equation (22).

$$L(x) = f(x) + \sum v_i\, h_i(x) + \sum u_i\, g_i(x) \qquad (22)$$

The KKT necessary conditions for equation (22) are expressed as follows :

$$\frac{\partial L}{\partial x_1} = 0 \,, \frac{\partial L}{\partial x_3} = 0 \,, \frac{\partial L}{\partial x_4} = 0 \qquad (23)$$

$$\frac{\partial L}{\partial v} = 0 \quad , \frac{\partial L}{\partial u_1} \le 0 \,, \frac{\partial L}{\partial u_2} \le 0 \qquad (24)$$

$$u_1 g_1 = 0, u_2 g_2 = 0, u_1 \ge 0, u_2 \ge 0 \qquad (25)$$

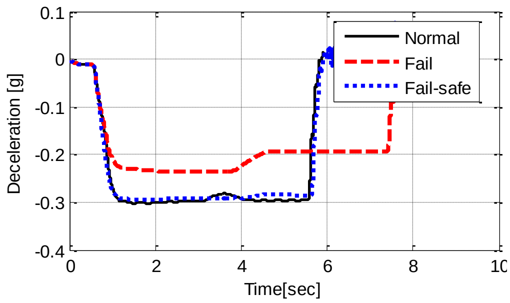
v , u_i are the Lagrange multipliers in equation (24). As shown in equation (25), one way to satisfy the switching conditions of equation (25) is to identify various cases and then solve them for the roots. There are four cases. Solving for all switching conditions, the solution for the switching condition of $g_1 = 0$, $u_2 = 0$ is found as feasible set. The solutions, x_i, are functions of F_i, $M_{z,c}$, $\beta a_{x,des}$ and F_2'. These can be calculated and determined in simple way as commented in previous sections. Through these processes suggested in Chapter 3, the additional braking forces subjected on normally remain 3 wheels can be carried out analytically.

4 Simulation Results

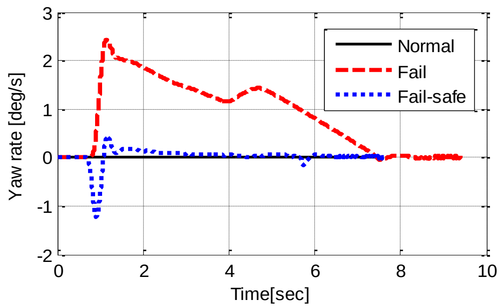
To validate the proposed fail-safe control algorithm, the simulation established in Matlab/Simulink environment was conducted. The assumptions of simulations are as follows :

- The initial vehicle speed is 50kph and the desired deceleration is 0.3g.
- The failure of brake actuator is detected in front right wheel at 0 sec and 1.5 sec. And the driver steps on the brake pedal at 0.5sec.
- The failure of brake actuator can be detected in some appropriate methods.
- The vehicle is driving on dry asphalt, so the road friction coefficient is 1.
- There is no steering wheel input, $\delta = 0$.

The simulations include the three cases, such as normal braking, brake actuator failure without fail-safe algorithm and actuator failure with fail-safe algorithm. In the Fig. 8 and 9, the deceleration and yaw rate of these cases are compared together.



(a) Deceleration



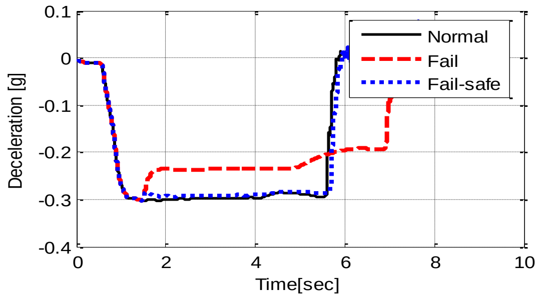
(b) Yaw rate

Figure 8 : Results about Failure at 0 sec

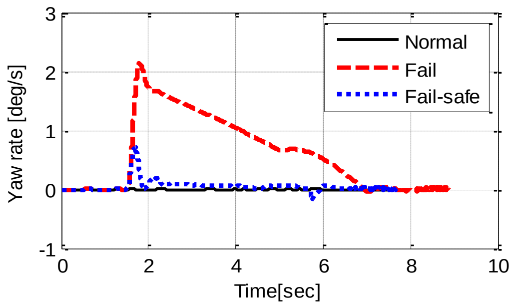
The scenario of brake actuator failure at 0 sec is depicted in Fig. 8. The brake actuator failure happens before driver steps on the brake pedal. The vehicle with the proposed fail-safe algorithm can follow the desired deceleration and the undesired yaw rate converges to zero rapidly. The yaw rate with fail-safe algorithm was reduced dramatically as compared with yaw rate without fail-safe altorithm.
The similar results are also shown in Fig. 9. The brake actuator failure occurs at 1.5 sec, namely in 1.0 sec after the driver steps on the brake pedal. The vehicle with the proposed fail-safe algorithm

can follows the desired deceleration and the undesired yaw rate converges to zero rapidly.

In these cases, β of equation (20) is one, because the desired deceleration is low. Thus, the normal three actuators can make braking force as amount of the desired deceleration and generate the corrective yaw moment.



(a) Deceleration



(b) Yaw rate

Figure 9 : Results for failure at 1.5sec

As shown in these simulation results, the two purposes of the proposed fail-safe control algorithm are achieved properly if there are some appropriate failure detection methods whenever the brake actuator failure happens. The two purposes of the proposed algorithm are that the vehicle should follow the desired deceleration and minimize the undesired yaw rate under one brake actuator failure.

5 Conclusion

The vehicle with BBW system has potential dangers of brake actuator failure caused by malfunction of Electric/Electronic parts such as wheel ECU, actuating motor and some sensors. To overcome this problem, the fail-safe control algorithm is proposed in this paper.

Using the proposed fail-safe control algorithm, the additional brake forces distributed to normally remaining wheels can be calculated by solving the linear simultaneous equations which are derived from KKT optimal conditions. In order to define the constraints of KKT conditions, the corrective yaw moment which the additional brake forces has to generate are derived using sliding mode control method. The effectiveness of the proposed algorithm is verified in computer simulation using Matlab/Simulink and Carsim.

In the future, if some estimation algorithms of the road friction coefficient, lateral tire forces and longitudinal tire forces are applied to the proposed fail-safe controller, this controller may become more robust and applicable in various real driving conditions.

Acknowledgments

This work has been funded in part a grant from Ministry of Trade, Industry and Energy in Korea under contract No.10033817.

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