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Hardware-in-the-Loop Simulation for a Wheel Slide Protection System of a Railway Train

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Abstract: A wheel slide protection (WSP) system of a railway train has the role of reducing excessive wheel slide from brake applications in situations where wheel/rail adhesion is temporarily impaired. The mechanism of the WSP is complex and is related to highly nonlinear dynamics of the train. Hardware-in-the-loop simulation (HILS) for the WSP system can test various dangerous braking conditions which are not possible in actual train tests, and help to find appropriate parameters of the WSP system. This paper presents a HILS unit for the WSP system, which is composed of an actual WSP unit, two actual dump valves, a dSPACE module, and a personal computer with Linux operating system. The dSPACE module simulates railway train dynamics in real-time, which includes dynamic models of wheelsets, bogies, a carbody, and a mechanical brake, and also a dynamic model for creep force generation in wheel/rail contacts. The validity of the HILS system for WSP control is demonstrated by ways of off-line simulation studies and experimental tests using the HILS unit for the Korean tilting train, Hanvit 200.

Keywords: Train control, brakes, HILS, wheel slide protection, real-time simulation, railway train.

1. INTRODUCTION

A wheel slide protection (WSP) system of a railway train has the role of reducing excessive wheel slide from brake applications in situations where wheel/rail adhesion is temporarily impaired due to inclement weather conditions or fouling of the rail. The WSP system is activated by a temporary reduction in braking force, and exploits available wheel/rail adhesion to a maximum and improves it by providing controlled wheel slide so that any increase in braking distance is kept to a minimum. The WSP system (UIC Code 541-05, 2005) shall so vary the braking force as to make maximum use of available adhesion.

When wheel/rail adhesion drops suddenly during braking, the WSP system shall prevent irremediable locking of the wheelsets at vehicle speeds above the initiation threshold for the WSP system, and shall avoid wheel flats and rail damages due to wheel slides. A wheel flat is a flat spot on the rolling surface of the wheel caused by its unintentional slide on the rail by poorly adjusted parameters, frozen or defect brakes, too high braking forces, or contaminated rail such as by leaves, grease, frost and snow (Jergeus, 1998; Jergeus et al., 1999). A damaged wheel with the flat should be taken out of service and be reprofiled as quickly as possible. The wheel flat degrades ride comfort and aggravates running safety, and it costs high maintenance fee of the railway train. There have been several trials to improve WSP performance through advanced control logics such as sliding mode control (Yamazaki et al., 2008; Park et al., 2008).

Hardware-in-the-loop simulation (HILS) for the WSP system can test various dangerous braking conditions which are not possible in actual train tests, and help to find appropriate parameters of the WSP system (Pugi et al., 2006). The mechanism of the WSP is complex and is related to highly nonlinear dynamics of the train.

The way in which the forces are transmitted in the rolling contact between wheels and rails of railway vehicles is complex from a microscopic viewpoint and is able to be represented mathematically in a highly nonlinear fashion (Stuetzle et al., 2008). The understanding of wheel slide and braking mechanism with rolling contact has a crucial role for the safety and efficiency of railroad vehicle operations.

Since Carter (1926) introduced a creep concept for rolling contacts of railway vehicles, the rolling contact theory has been developed well by many researchers. Carter (1926) proposed a solution for creep forces acting on the wheel/rail contacts through two-dimensional analysis. Johnson (1985) generalized Carter's results to approximate 3-dimensional analysis for longitudinal creepage and lateral creepage of circular contact patches between two elastic bodies. De Pater and Kalker (Iwnicki, 2006) established a linear theory on the relationship between creepage and creep force, and Johnson and Vermulen proposed a three dimensional theory (Kalker, 1991, 1979). Shen, Hedrick and Elkins (Kalker, 1990) improved the above theory to be well-matched with experimental results, and Kalker (1979, 1990, 1991) established good creepage and creep force models that are currently used for commercial simulation packages (Iwnicki, 1999; Wickens, 2003).

Seoul Metro Line 4 (Korea) during 2004 and 2009 reports that failures in brake systems form 17.6 percent of total failures, which is the second most part among the failures (Sohn et al., 2009). This contains wheel flats resulted by wheel slides.

Brake system is not only the very important technology that is directly connected to safety issue among train technologies but also very complicated and difficult technology. However, it is very limited to test various real situations including emergency cases with actual trains.

To resolve these problems, Kang et al.(2009) performed brake dynamics simulations of the Korean tilting train, Hanvit-200, and Kim et al.(2009) implemented real-time simulation about mechanical brake dynamics of the Hanvit 200 based on offline simulation research about anti-skid systems.

HILS systems are widely used in many fields to verify and test before actual building of a certain system. Especially, HILS systems are used in industries that take lots of money and manpower to actually build a system (Huang et al., 2010; Spiryagin et al., 2012). Kim et al.(2010) did research on a HILS system using ASCU (Anti-Skid Control Unit) of the Korean tilting train, Hanvit-200.

This paper presents a HILS unit for the WSP system, which is composed of an actual WSP unit, two actual dump valves, a dSPACE module, and a personal computer with Linux operating system. The dSPACE module simulates railway train dynamics in real-time, which includes dynamic models of wheelsets, bogies, a carbody, and a mechanical brake, and also a dynamic model for creep force generation in wheel/rail contacts. The validity of the HILS system for WSP control is demonstrated by ways of off-line simulation studies and experimental tests using the HILS unit for a Korean tilting train, Hanvit 200.

2. CAR DYNAMICS

In this paper, a two-dimensional (2D) model for one trailer car of the railway train is considered to evaluate braking performance in the directions of longitudinal, vertical, and pitch motions of railway trains. Even if this model is simple and convenient to analyze dynamic characteristics, it includes key dynamic characteristics required for designing and analyzing the basic braking logic of the train running in straight rail tracks. To analyze braking performance in curved tracks, of course, we need three-dimensional dynamic model.

The proposed 2D model is composed of a carbody, front and rear bogies, four wheelsets, and primary and secondary suspensions. Secondary suspension includes air spring, yaw damper, and traction link. The wheelset includes the creepage model for wheel/rail contacts. In the model, x denotes the longitudinal forward direction, z denotes the vertical downward direction, and θ denotes the angle of pitch motion. All the parameters used in this paper are given from the tilting train, Hanvit-200 (Korea Railroad Research Institute, 2005). However, the proposed model can be applied to the other railway trains with parameter adjustments. Fig. 1 shows disk brakes of the Hanvit-200, and Fig. 2 shows the schematics of one car model of the tilting train.

Fig. 1. A picture of disk brakes of the Hanvit-200 train

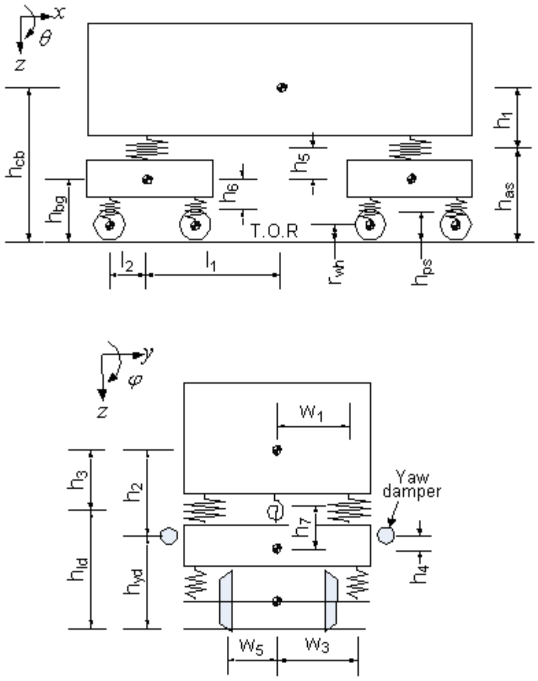

Fig. 2. Two dimensional car model including a carbody, two bogies and four wheelsets

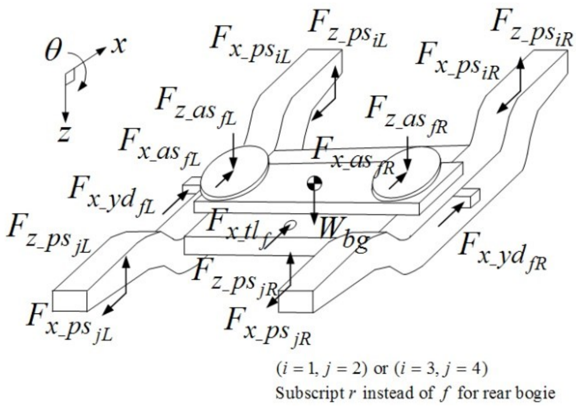

Fig. 3. Free body diagram of the bogies

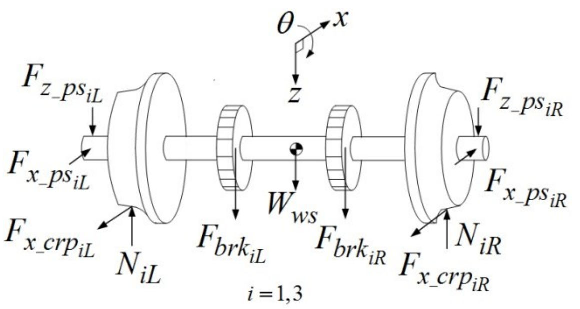

Fig. 4. Free body diagram of the wheelsets

Fig. 3 and Fig. 4 show free body diagrams of the bogie and wheelset. Carbody is connected with bogies by the secondary suspension. Appling Newton's law of motion to the free body diagrams, the following equations of motion are obtained (Kang et al., 2009). In the below, subscript *cb* implies carbody, *as* implies air spring, *yd* implies yaw damper, *tl* implies traction link, *f* and *r* represent front and rear, and *L* and *R* represent left and right, respectively. *m*, *I*, *c*, *k* represent mass, moment of inertia, viscous damping coefficient, spring constant, respectively, and h_i, l_i represent height and length.

In the bogie equation, subscript *bg* implies bogie, *fb* and *rb* imply front bogie and rear bogie, *wh* implies wheelset, and *ps* implies primary suspension. For the rear bogie, subscript *fb* must be replaced by *rb*, *ws*1 and *ws*2 must be replaced by *ws*3 and *ws*4, respectively, and signs of pitch motion terms including θ must be reversed. In the wheelset 1 equation, N_1 implies reaction force acting on one wheel, and *r* implies radius. The subscripts *crp* and *brk* denote creep force and brake force. For the equations of motion for the 3rd wheelset, subscript 1 in the equations must be replaced by 3.

Carbody:

$$\begin{aligned} m_{cb}\ddot{x}_{cb} &= -2\{k_{x\_as}(x_{cb}-x_{fb})+c_{x\_as}(\dot{x}_{cb}-\dot{x}_{fb})\} \\ &\quad -2\{k_{x\_as}(x_{cb}-x_{rb})+c_{x\_as}(\dot{x}_{cb}-\dot{x}_{rb})\}-2\{c_{x\_yd} \\ &\quad (\dot{x}_{cb}-\dot{x}_{fb})+c_{x\_yd}(\dot{x}_{cb}-\dot{x}_{rb})\}-\{k_{x\_tl}(x_{cb}-x_{fb}) \\ &\quad +k_{x\_tl}(x_{cb}-x_{rb})\} \end{aligned}$$

$$\begin{aligned} m_{cb}\ddot{z}_{cb} &= -2\{k_{z\_as}(z_{cb}+l_1\theta_{cb}-z_{fb})+c_{z\_as}(\dot{z}_{cb}+l_1\dot{\theta}_{cb} \\ &\quad -\dot{z}_{fb})\}-2\{k_{z\_as}(z_{cb}-l_1\theta_{cb}-z_{rb})+c_{z\_as}(\dot{z}_{cb}-l_1\dot{\theta}_{cb} \\ &\quad -\dot{z}_{rb})\}+m_{cb}g \end{aligned}$$

$$\begin{aligned} I_{cb}\ddot{\theta}_{cb} &= 2\{k_{x\_as}(x_{cb}-x_{fb})+c_{x\_as}(\dot{x}_{cb}-\dot{x}_{fb})+k_{x\_as} \\ &\quad (x_{cb}-x_{rb})+c_{x\_as}(\dot{x}_{cb}-\dot{x}_{rb})\}h_1-2\{k_{z\_as}(z_{cb} \\ &\quad +l_1\theta_{cb}-z_{fb})+c_{z\_as}(\dot{z}_{cb}+l_1\dot{\theta}_{cb}-\dot{z}_{fb})\}l_1+2\{k_{z\_as} \\ &\quad (z_{cb}-l_1\theta_{cb}-z_{rb})+c_{z\_as}(\dot{z}_{cb}-l_1\dot{\theta}_{cb}-\dot{z}_{rb})\}l_1+2 \\ &\quad \{c_{x\_yd}(\dot{x}_{cb}-\dot{x}_{fb})+c_{x\_yd}(\dot{x}_{cb}-\dot{x}_{rb})\}h_2+\{k_{x\_tl}(x_{cb} \\ &\quad -x_{fb})+k_{x\_tl}(x_{cb}-x_{rb})\}h_3 \end{aligned}$$

Bogie:

$$\begin{aligned} m_{bg}\ddot{x}_{fb} &= 2\{k_{x\_as}(x_{cb}-x_{fb})+c_{x\_as}(\dot{x}_{cb}-\dot{x}_{fb})\} \\ &\quad +2c_{x\_yd}(\dot{x}_{cb}-\dot{x}_{fb})+k_{x\_tl}(x_{cb}-x_{fb})-2\{k_{x\_ps} \\ &\quad (x_{fb}-x_{ws1})+c_{x\_ps}(\dot{x}_{fb}-\dot{x}_{ws1})\}-2\{k_{x\_ps}(x_{fb} \\ &\quad -x_{ws2})+c_{x\_ps}(\dot{x}_{fb}-\dot{x}_{ws2})\} \end{aligned}$$

$$\begin{aligned} m_{bg}\ddot{z}_{fb} &= 2\{k_{z\_as}(z_{cb}+l_1\theta_{cb}-z_{fb})+c_{z\_as}(\dot{z}_{cb}+l_1\dot{\theta}_{cb} \\ &\quad -\dot{z}_{fb})\}+m_{bg}g-2\{(k_{z\_ps}(z_{fb}+l_2\theta_{fb}-z_{ws1}) \\ &\quad +c_{z\_ps}(\dot{z}_{fb}+l_2\dot{\theta}_{fb}-\dot{z}_{ws1})\}-2\{k_{z\_ps}(z_{fb}-l_2\theta_{fb} \\ &\quad -z_{ws2})+c_{z\_ps}(\dot{z}_{fb}-l_2\dot{\theta}_{fb}-\dot{z}_{ws2})\} \end{aligned}$$

$$\begin{aligned} I_{bg}\ddot{\theta}_{fb} &= 2\{k_{x\_as}(x_{cb}-x_{fb})+c_{x\_as}(\dot{x}_{cb}-\dot{x}_{fb})\}h_5 \\ &\quad -2\{k_{x\_ps}(x_{fb}-x_{ws1})+c_{x\_ps}(\dot{x}_{fb}-\dot{x}_{ps1})\}h_6 \\ &\quad -2\{k_{x\_ps}(x_{fb}-x_{ws2})+c_{x\_ps}(\dot{x}_{fb}-\dot{x}_{ws2})\}h_6 \\ &\quad -2\{k_{z\_ps}(z_{fb}+l_2\theta_{fb}-z_{ws1})+c_{z\_ps}(\dot{z}_{fb}+l_2\dot{\theta}_{fb} \\ &\quad -\dot{z}_{ws1})\}l_2+2\{k_{z\_ps}(z_{fb}-l_2\theta_{fb}-z_{ws2})+c_{z\_ps} \\ &\quad (\dot{z}_{fb}-l_2\dot{\theta}_{fb}-\dot{z}_{ws2})\}l_2+2c_{x\_yd}(\dot{x}_{cb}-\dot{x}_{fb})h_4 \\ &\quad +k_{x\_tl}(x_{cb}-x_{fb})h_7 \end{aligned}$$

Wheelset 1:

$$\begin{aligned} m_{ws}\ddot{x}_{ws1} &= 2\{k_{x\_ps}(x_{fb}-x_{ws1})+c_{x\_ps}(\dot{x}_{fb}-\dot{x}_{ws1})\} \\ &\quad -2F_{x\_crp_1} N_1 = \{k_{z\_ps}(z_{fb}+l_2\theta_{fb})+c_{z\_ps}(\dot{z}_{fb} \\ &\quad +l_2\dot{\theta}_{fb})\}+F_{brk_1}+0.5m_{ws}g \end{aligned}$$

$$I_{ws}\ddot{\theta}_{ws1} = 2r_{wh}F_{x\_crp_1}-2r_{disk}F_{brk_1}$$

For the equations of motion for the 2nd and 4th wheelsets, the sign of F_{brk} must be reversed in the above equation, and subscript 1 in the equations must be replaced by 2, 4, respectively. For the motor car with wheel-disk brakes, disks are removed in Fig. 4, and F_{brk} acts on the wheel treads instead of the disks.

The transmitted force on the wheel/rail contact can be simply modeled as Coulomb friction, that is, the wheel and the rail are contacted at a point, and no slide occurs and tangential force at the contact point is equal to the brake force if the brake force is less than μN, and full slide occurs and the tangential force at the contact point is equal to μN if a bigger brake force than μN is applied.

However, this Coulomb friction model cannot represent the wheel/rail contact phenomena in detail, and we cannot use this model for the purpose of braking performance analysis of the train. Thus, we need more realistic model that resembles and characterizes well the real contact phenomena between wheel and rail. In real situation, a part of the contact patch becomes adhesion region and the other part becomes a slip region according to the creepage, i.e., relative slip velocity of the contact patch (Kalker, 1979, 1990; Kim et al., 2009). The resulting tangential force totally becomes a continuous function of slip velocity.

For 2D motions, longitudinal creepages γ_L, γ_R for the left and right wheels can be represented by

$$\gamma_L = \frac{V - r_L\omega}{V}, \quad \gamma_R = \frac{V - r_R\omega}{V}$$

where V is the speed of the contact patch on the rail, that is, the forward speed of the vehicle, $r_L\omega, r_R\omega$ are speeds of the contact patches on the left and right wheels.

For small creepages, the relationship between creep forces and creepages can be assumed to be approximately linear, and then longitudinal creep forces F_{x_crp} can be given by

$$F_{x\_crpL} = -f\,\gamma_L, \quad F_{x\_crpR} = -f\,\gamma_R$$

where the coefficient f depends on the size and shape a,b of the contact patch, and Young's modulus E, and were calculated by Kalker (1991) as follows.

$$f = E(ab)C_1$$

$$ab = [1.5(1-\nu^2)Nr_o / E]^{2/3}$$

In the above equations, a and b imply the major and minor axis of the contact patch, ν implies Poisson's ratio, and N and r_0 imply the wheel load and the nominal wheel radius. Kalker's coefficient C_1 is given by a function of a/b (Wickens, 2003).

3. WHEEL SLIDE PROTECTION SYSTEM

The wheel Slide Protection (WSP) system of the Korean tilting train, Hanvit 200, is considered to implement a HILS system of a mechanical brake system. Fig. 5 shows the functional block diagram of the WSP system of Hanvit 200 that is located at an indoor space between cars.

The WSP system includes one ASCU (anti-skid control unit), two dump valves, and four speed sensors. In actual situations, angular velocities of four wheelsets are detected by speed sensors. Each sensor generates pulse signals, and ASCU makes decisions about skid and re-adhesion according to the change of wheelset speeds obtained pulse signals from speed sensors, and then dump valves control brake forces according to the decision of ASCU by controlling air pressure of brake cylinders (Yujin, 2004).

The ASCU is divided into a control part and a drive part, in which the control part takes charge of calculating speed changes and decision making, and the drive part takes charge of generating actuation signals of dump valves.

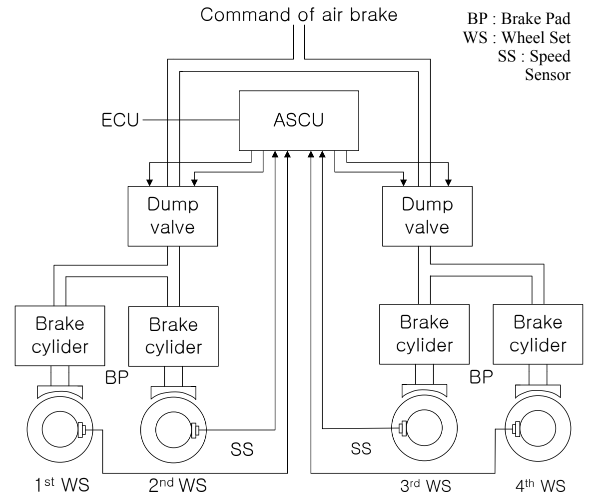

Fig. 5. Functional block diagram of the WSP operations

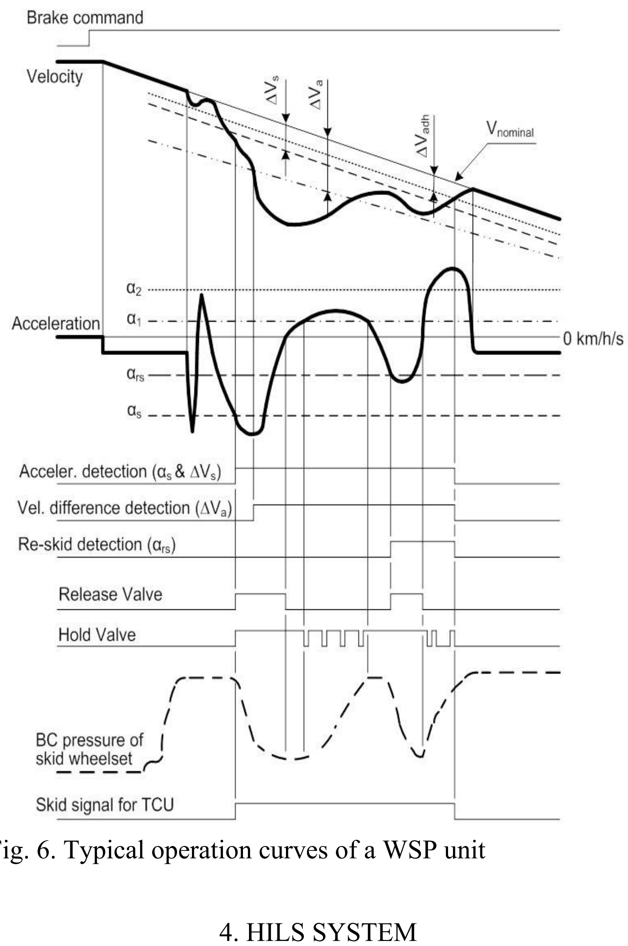

Fig. 6. Typical operation curves of a WSP unit

4. HILS SYSTEM

A HILS system is proposed to test and analyze braking and WSP performances, which is composed of a dSPACE to run vehicle dynamics, real hardware components of ASCU and two dump valves, and Simulink and Real-Time Workshop Toolbox softwares, and PC. Fig. 7 shows the picture of the proposed HILS system, and Fig. 8 shows the conceptual configuration of the HILS system.

As shown in Fig. 8, vehicle dynamics is run in real-time at dSPACE board and the resulting four wheel speeds (corresponding to speed sensors) are transferred to ASCU. Then ASCU generates input signals for two dump valves and two dump valves are activated, which generates required braking forces according to WSP logic. Pneumatic pressure of the brake cylinder is calculated by the approximate first-order model with 0.6 s time constant and 0.15 s time delay. Brake force F_{brk} of the mechanical brake is calculated by

$$F_{brk} = \mu\left(PA_p - F_{spring}\frac{P}{P_{\max}}\right)\times l_r \times \eta$$

where μ is an average friction coefficient between disk and caliper, A_p is an effective piston area, F_{spring} is an average spring force of the brake cylinder, l_r is lever ratio, and η is brake efficiency.

The calculated braking forces are fed back to car dynamics module using the dSPACE Kit. Car dynamics is programmed using Simulink, and converted to source code using Real-Time Workshop Toolbox of MATLAB. Fig. 9 shows a GUI screen of the HILS system, which is programmed using ControlDesk software by dSPACE GmbH. The GUI screen is divided into two parts; one is parameter setting part, and the other is graphical plotting area of simulation results. Starting HILS system after setting initial speed, brake mode, and adhesion coefficients, we can see the real-time plotting of the simulated speed in the GUI screen.

Fig. 7. Picture of the proposed HILS system

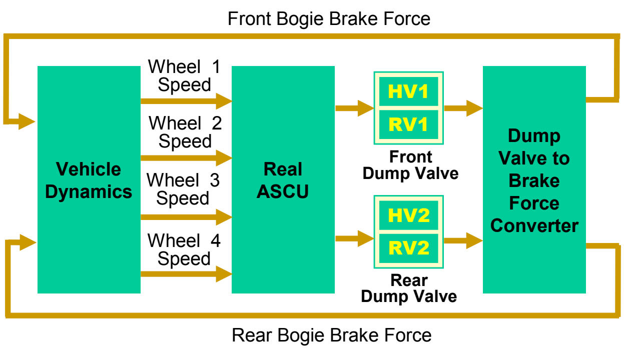

Fig. 8. Conceptual configuration of the HILS system

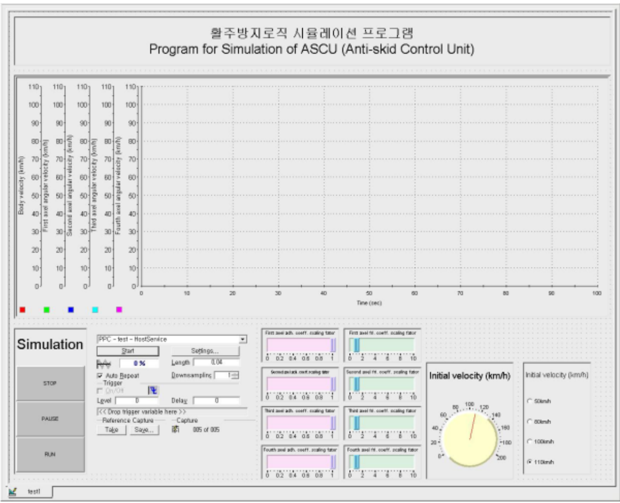

Fig. 9. GUI screen of the HILS system

5. HARDWARE-IN-THE-LOOP SIMULATION RESULTS

To verify the validity of the HILS results, off-line simulation (not real-time) results were compared with the HILS results. When initial speed was 50 km/h and braking force was set to a constant 6 kN, the two simulation results were same up to the second digit below the decimal point.

Now we set the initial speed to 100 km/h, and generate wheel slide between the 1st wheel and rail by lowering friction coefficient from normal value to 5 % of it. Then the anti-skid logic is operated according to the contact condition, and the braking force acting on the brake disk is generated appropriately according to the pre-determined scenario. As a result, skid and re-adhesion are occurred consecutively.

The ASCU used in the HILS system has a function of diagnosis that generates skid situation artificially and tests WSP logic without external speed sensor inputs. When the test switch of ASCU is pressed, it runs diagnosis program automatically. If external speed sensor signals are connected to ASCU, then it operates automatically in the normal mode with green LED blinking as shown in Fig. 10, which is another way to demonstrate appropriate operation of the proposed HILS system.

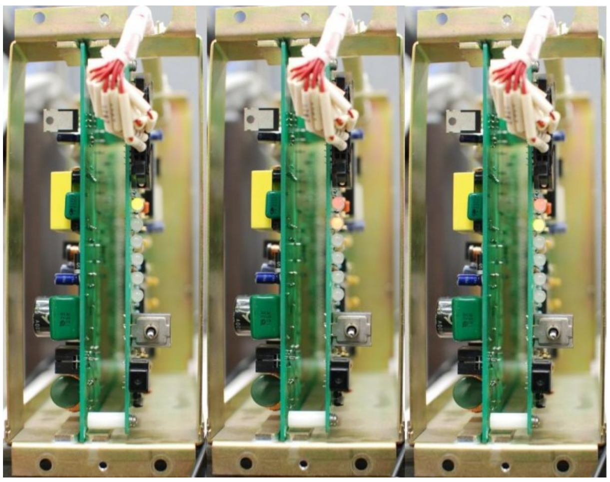

Fig. 10. Hardware-in-the-loop simulation when wheel slip occurred at the 1st wheelset

6. CONCLUSIONS

In this paper, a HILS system to test and analyze braking and WSP performance has been proposed, which is composed of an actual WSP unit, two actual dump valves, a dSPACE module, and a personal computer with Linux operating system. The dSPACE module simulates railway train dynamics in real-time, which includes dynamic models of wheelsets, bogies, a carbody, and a mechanical brake, and also a dynamic model for creep force generation in wheel/rail contacts. The results are as follows.

(1) Mechanical braking system and car dynamics of the Korean Tilting Train Hanvit-200 have been modeled for real-time simulations.
(2) Wheel slide protection logic of the Korean Tilting Train Hanvit-200 has been analyzed.
(3) A HILS system for braking and WSP of the Hanvit 200 has been proposed, which can be easily applied to other types of railway trains by adjusting system parameters.
(4) The proposed HILS system running in real-time has been verified by comparing off-line simulation results with constant brake forces.

Further research has been going on to construct a HILS system additionally including a brake caliper, a pneumatic system and a brake pad with actual elements.

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