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Automatic Anti-Lock Brake System for Anti-Rollover Control of Autonomous Heavy-Duty Truck

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ABSTRACT

In recent years, there are more and more rollover accidents about autonomous heavy-duty trucks or autonomous heavy-duty forklifts in intelligent airports or seaports. These accidents lead to the hot research field about the prevention of rollovers in advance for autonomous vehicles, especially for autonomous heavy-duty trucks or forklifts. This paper develops an automatic Anti-lock Brake System (ABS) as one way to stabilize autonomous vehicles. Through monitoring both vehicle's and wheel's speeds, this proposed ABS helps the autonomous vehicles keep stability control even when the wheels halt or slip on the road. This paper adopts TruckSim to model the vehicle safety dynamics and MATLAB/Simulink to simulate the vehicle stability control. Experimental results show that the elaborate automatic ABS proposed by this paper can smoothly keep the vehicle stable and tractive even under dangerous road conditions of a sharp corner or hairpin turn.

Keywords: Anti-lock Brake System (ABS), anti-rollover, autonomous truck.

1. INTRODUCTION

In the past few years, autonomous heavy-duty trucks or heavy-duty forklifts have been more and more popular in intelligent airports or seaports, but rollover accidents also get more and more frequent. Research survey demonstrates that autonomous heavy-duty trucks during logistics are likely harmful to road safety, such as rollovers, shimmy, or jackknives, due to the high mass center, comparatively big size, or relatively small base (Tianjun, 2010). The most significant hazard to road safety is the rollover of autonomous heavy-duty trucks, which can lead to catastrophic results and significant losses (Hilton & Shankar, 2003).

A vehicle rollover is a phenomenon of vehicle instability caused by extreme steering or extreme roadway. In the United States in 2016, there are over 6 million vehicle crashes. Among these crashes, 17.9% are vehicle roll-off deadly incidents, 8.5% are from massive lorries and buses (National Highway Traffic Safety Administration, 2018).

Unconsidered rollover events are typically caused by high speed or extreme steering. If a vehicle crosses a curved road, it leans outside the curved road because of its centrifugal force. The rollover may happen if the autonomous vehicle misjudges the curved sharpness and keeps excessively fast so as to exceed its lateral acceleration tolerance (Yu, Güvenc & Özgüner, 2008).

The risk of rollover accidents might be reduced by a systematic rollover risk assessment and active anti-rollover control under difficult driving situations (Dong et al., 2018). When the rollover dangers can be predicted, a warning may be given to the autonomous vehicle or the automatic Anti-lock Brake System (ABS). Proper reaction from the autonomous vehicle or the automatic ABS can prevent the rollover accident in advance effectively and efficiently. The active rotating control is also one of the best solutions if the autonomous vehicle does not react appropriately (He et al., 2019). Many researchers are analyzing these stability issues and improving the autonomous heavy-duty trucks or forklifts to lessen the occurrence probability of rollover.

Because the proposed automatic ABS is based on slip control, the slip ratio of the wheel's angular speed to the vehicle speed is determined. The slip ratio may be adapted to supply the wheels with braking power using other methods, for example, a specific speed sensor may estimate the slip ratio. This paper proposes to adopt automatic ABS to keep stability control of autonomous heavy-duty trucks or forklifts. So, autonomous vehicles can always run stable and tractive even in the double lane change scenario or circle lane scenario.

This paper is organized as follows. Section 2 explains the definitions of autonomous heavy-duty vehicle dynamics modeling and the principle of anti-rollover control based proposed ABS and slip control. Then, Section 3 discusses the simulation designs and simulation results by MATLAB/Simulink and TruckSim. Finally, the last section provides the conclusion and future direction of the research.

2. VEHICLE DYNAMICS MODELING

The modeling of significant rolling action in the middle of the mass in autonomous heavy-duty trucks has special restrictions because of its lengthy wheel lowering and dividing the spring weight between the front and back, as shown in Figure 1(a). The front-to-back connective bobbin, the optimum torsional bar, the stiffness of the torsion, and mass for the front and rear spring systems (He et al., 2019) are illustrated in Figure 1(b).

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In view of the equation of different degrees of freedom, the principle can be defined and derived as follows:
Longitudinal motion:

$$m(\dot{u} - vr) = F_{xT} - F_r - 2F_{Y1}\sin\delta_f \tag{1}$$

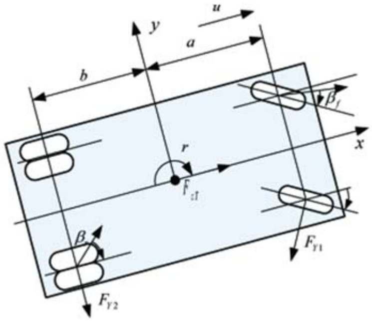
Lateral motion:

$$m(\dot{v} + ur) - m_{sf}h_f\ddot{\varphi}_{sf} - m_{sr}h_r\ddot{\varphi}_{sr} =$$

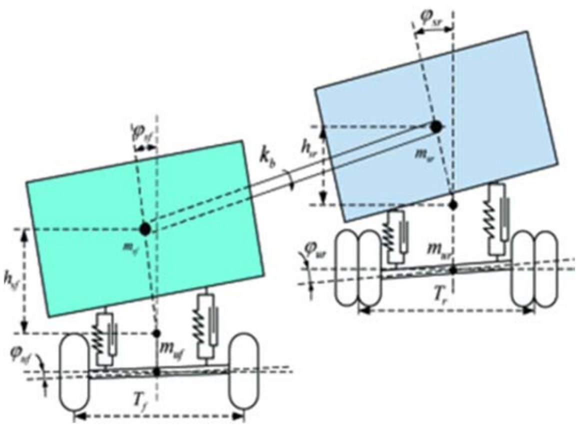
$$2F_{Y1}\cos\delta_f + 2F_{Y2} \tag{2}$$

Yaw motion:

$$I_z\dot{r} = 2aF_{Y1}\cos\delta_f + 2bF_{Y2} \tag{3}$$



(a)



(b)

Fig. 1 (a) Yaw action and (b) rolling action and chassis mechanism, of dynamics model of the autonomous truck.

An ideal torsion bar with set torsional stiffness and no mass is used to link the front and back spring systems, considering the coupling between the front axle roll angle and the rear axle roll angle.
Roll motion of front sprung mass:

$$I_{xf}\ddot{\varphi}_{sf} = -k_f(\varphi_{sf} - \varphi_{uf}) - l_f(\dot{\varphi}_{sf} - \dot{\varphi}_{uf})$$

$$+m_{sf}h_f a_y + m_{sf}h_f\varphi_{sf} + k_b(\varphi_{sr} - \varphi_{sf}) \tag{4}$$

Roll motion of rear sprung mass：

$$I_{xf}\ddot{\varphi}_{sf} = -k_r(\varphi_{sr} - \varphi_{ur}) - l_r(\dot{\varphi}_{sr} - \dot{\varphi}_{ur})$$

$$+m_{sr}h_r a_y + m_{sr}h_r\varphi_{sr} + k_b(\varphi_{sr} - \varphi_{sf}) \tag{5}$$

Because the unsprung mass of heavy trucks is so large, it significantly impacts the roll motion. As a result, a dynamic model with degrees of freedom may be obtained from Figure 1(b).
Roll motion of front unsprung mass：

$$2F_{Y2}h_c + m_{uf}(h_{uf} - h_{cf})a_y =$$

$$-m_{uf}\mathrm{g}(h_{uf} - h_{cf})\varphi_{uf} + k_{uf}\varphi_{uf}$$

$$+k_f(\varphi_{sf} - \varphi_{uf}) - l_f(\dot{\varphi}_{sf} - \dot{\varphi}_{uf}) \tag{6}$$

Roll motion of rear unsprung mass：

$$2F_{Y2}h_c + m_{ur}(h_{ur} - h_{cr})a_y =$$

$$-m_{ur}\mathrm{g}(h_{ur} - h_{cr})\varphi_{ur} + k_{ur}\varphi_{ur}$$

$$+k_r(\varphi_{sr} - \varphi_{ur}) - l_r(\dot{\varphi}_{sr} - \dot{\varphi}_{ur}) \tag{7}$$

Lateral acceleration at the center of mass of the vehicle:

$$a_y = (\dot{v} + ur) \tag{8}$$

Longitudinal displacement of the vehicle：

$$\dot{X} = u\cos\psi - v\sin\psi \tag{9}$$

Lateral acceleration at the center of mass of the vehicle:

$$\dot{Y} = u\sin\psi - v\cos\psi \tag{10}$$

Where *m* refers to the total mass of the autonomous heavy-duty vehicle, the front and rear axles are indicated by $\circledast \in \{f,r\}$. $m_{s\circledast}$ and $m_{u\circledast}$ denote the equivalent sprung and unsprung masses, respectively, the longitudinal distances between mass-center-to-front-axle and mass-center-to-rear-axle are *a* and *b*, correspondingly. $h_{\circledast}$ implies the length between the sprung mass center and the roll center, so $h_{u\circledast}$ and $h_{c\circledast}$ are measured heights of the center of rolls and the center of the unsprung mass, respectively, from the road. F_{xT} represents the longitudinal force of the tires and F_r means the wheel rolling resistance, then for lateral forces of the front and rear axles are the pneumatics, F_{Y1} and F_{Y2}. g is the gravitational acceleration, I_z is the yaw inertia of the heavy-duty vehicles, and $I_{x\circledast}$ is the roll inertia of the sprung mass. *u* is the longitudinal speed, and *v* denotes the lateral velocity. $k_{\circledast}$ and $k_{u\circledast}$ are the equivalent roll stiffness coefficients of the suspension and the unsprung mass, respectively. $l_{\circledast}$ is the equivalent roll damping coefficient of the suspension, *kb* is the torsion stiffness coefficient of the vehicle frame, *r* denotes the yaw rate of the sprung mass, and δ_f is the front-wheel steering angle. φ, $\dot{\varphi}$ and $\ddot{\varphi}$ are the vehicle sprung mass roll angle about the roll axis and it is derivatives. $\varphi_{u\circledast}$ and $\varphi_{s\circledast}$ are the roll angles of the sprung and the unsprung masses, respectively. *X* is the longitudinal displacement, *Y* is the lateral displacement, and ψ denotes the heading angle.

2.1 Static Rollover Threshold

Static Rollover Threshold (SRT) is a maximum lateral acceleration criterion to prevent the vehicle from rolling out. The autonomous truck is at the initial positioned horizontally. The platform is then slowly rotated till the tires become lost on the platform surface. Between the inclining angle between the platform and the ground, the tangent formed is the SRT value. The official definition of SRT is:

$$SRT = \frac{a_t}{g} \quad (11)$$

$$SRT = \frac{T}{2h} - \frac{\Delta_y}{h} \quad (12)$$

Where:

T = Wheel track width

Δ_y = Rollover threshold

2.2 ABS control

ABS is a system for electronic safety scheme which checks and monitors the speed of the wheel during braking. The relative sliding between the wheel and road surface may be assessed in the braking process by the slip ratio (Lu, Zhou & Wang, 2015):

$$S = \frac{V - \omega.R}{V} \text{ x } 100\% \quad (13)$$

Where:

ω = angular velocity of the wheel

R = the effective wheel radius.

With the increase of the brake force, the tire slip ratio raises the braking force coefficient and finally reduces the slip ratio. At the same time, the coefficient of lateral force gradually diminishes. With the glitch ratio reaching S_c, peaks of braking and lateral strength are relatively high. It also ensures the ideal braking performance, and avoids sideslip. By controlling the hydraulic pressure in brake lines, ABS may keep the ratio of slip around S_c (Lu, Zhou & Wang, 2015).

2.3 Lateral Load Transfer Ratio (LTR)

A typical LTR is estimated using the data received at a particular moment. It looks like capturing a snapshot of a dynamic system. Analytical analyses and experimental data help define the rollover threshold based on the predicted LTR values. When the threshold is set to be sensitively low, the LTR will alert or trigger a rollover prevention system even during regular and safe driving period. When the threshold is set to be slightly high, prevention measures might activate too late to prohibit a rollover of the vehicle (Tsourapas et al., 2009).

$$LTR = \frac{F_{zR}}{F_{zL}} - \frac{F_{zL}}{F_{zR}} \quad (14)$$

This index in (14) uses vertical pneumatic forces, F_{zL} and F_{zR}. LTR describes vehicle rollover when lifting from the ground as either on the left or right side of the vehicle. LTR varies from –1 to 1, where –1 and 1 refer to either the left or right vehicle tires losing contact with the ground, and 0 refers to equal vertical forces on both sides of the vehicle (zero rolls) (Tsourapas et al., 2009).

3. SIMULATION RESULTS

Based on the co-simulation of MATLAB/Simulink and TruckSim, modeling the truck dynamics in the TruckSim simulator with several significant parameters, as shown in Figure 2, can make it easier for the developer to perform the evaluation and analysis. Each parameter in TruckSim simulator can be captured and calculated by MATLAB/Simulink.

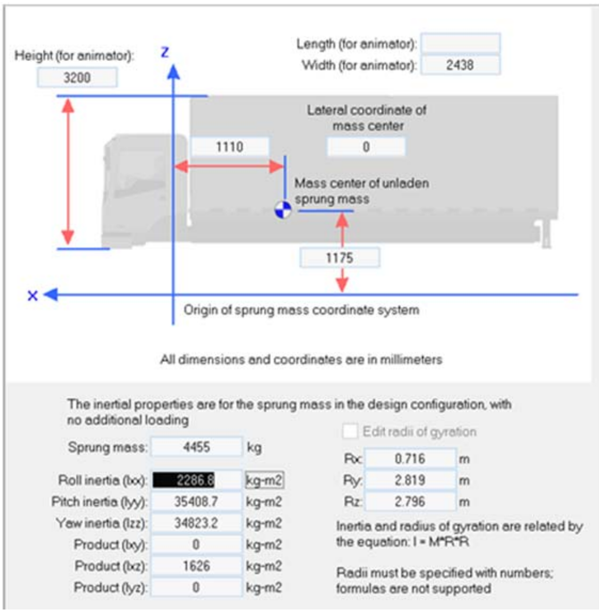


Fig. 2 Vehicle dynamics modelling by TruckSim.

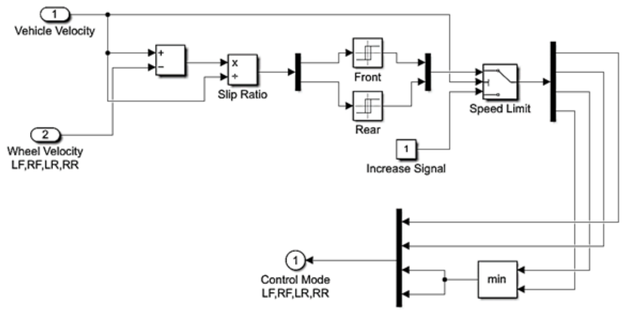


Fig. 3 ABS control model in MATLAB/Simulink.

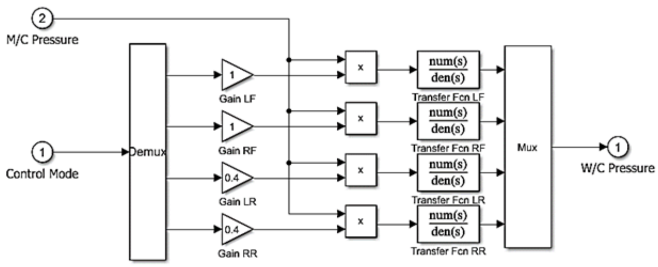


Fig. 4 Brake actuator model in MATLAB/Simulink.

Brake actuators are the instruments that turn a compressed air force into a mechanical force within the vehicle or air reservoir of the truck that triggers the cadence brake. Figure 4 shows MATLAB/Simulink model for brake actuators.

Table 1 Vehicle dynamics model of truck 1

Properties	Value	Unit
Unsprung mass	570	kg
Axle roll & yaw inertia	350	kg-m^2
Left spin inertia	10	kg-m^2
Right spin inertia	10	kg-m^2
Wheel center height	510	mm
Center of gravity	2030	mm
Sprung mass origin	510	mm

Table 2 Vehicle dynamics model of truck 2

Properties	Value	Unit
Unsprung mass	735	kg
Axle roll & yaw inertia	285	kg-m^2
Left spin inertia	20	kg-m^2
Right spin inertia	20	kg-m^2
Wheel center height	530	mm
Center of gravity	1863	mm
Sprung mass origin	530	mm

For a comprehensive comparison, this experiment designs two different vehicle dynamics models for two autonomous heavy-duty truck simulations, which are listed in Tables 1 and 2. From Tables 1 and 2. It is shown truck 2 with lower gravity center has lighter duty than truck 1, but truck 2 with higher other properties has stronger chassis and brake mechanism than truck 1. That is, truck 2 itself has better anti-rollover control and stability recovery than truck 1 itself in the designed experiment.

A series of simulation results in Figures 5-10 are carried out by comparison between two trucks with and without anti-rollover control based on the proposed automatic ABS, respectively. Figure 5 illustrates the steering angle of 2 heavy-duty trucks can keep almost stable during 0-6 seconds, but two trucks without anti-rollover control loses their stability subsequently and loses their steering angle maintenance afterward. Because the steering angle can work normal only when the truck runs along the regular roadway. If the trucks control the longitudinal speed elaborately to adapt some curved roadway situation, the trucks can keep tracking the trajectory to recover the vehicle stability. At this moment, automatic ABS is proposed to prevent these two heavy-duty trucks from rollover by decreasing their longitudinal speed in time, as shown in Figure 6. In Figure 6, it is illustrated the proposed automatic ABS is activated at the second of about 3 for truck 1 and at the second of about 5 for truck 2. This is because the vehicle dynamics model of truck 2 is designed to have better anti-rollover control and stability recovery inherently than the one of truck 1.

Figure 7 illustrates the lateral acceleration comparison between two trucks with and without anti-rollover control based on the proposed automatic ABS, respectively. Two trucks without anti-rollover control suffer from rollover risk, but two trucks with anti-rollover control based on the proposed automatic ABS can overcome the dangerous roadway condition. Braking force elaborately reduces the lateral acceleration based on the principle of LTR as (14). Figure 8 compares the response results of dynamic yaw rate of two trucks with and without anti-rollover control based on the proposed automatic ABS. From Figure 8, it is seen that the braking force intensity applied by the proposed automatic ABS is rightly sufficient to confine the vehicle's yaw rate within safe tolerance.

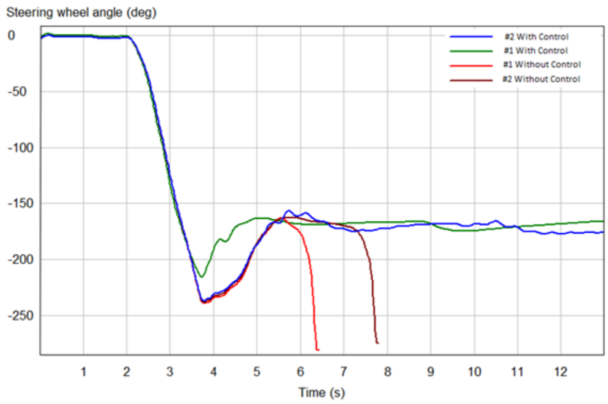


Fig. 5 Steering angle comparison.

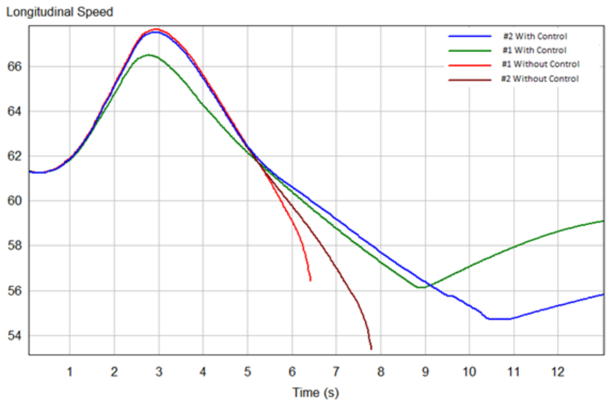


Fig. 6 Longitudinal speed comparison.

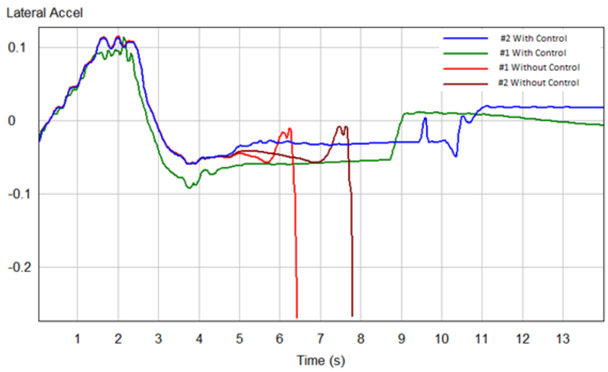


Fig. 7 Lateral acceleration comparison.

The roll angle comparison shown in Figure 9 is used to evaluate the rollover risk of two trucks with and without anti-rollover control based on the proposed automatic ABS, respectively. Figure 9 indicates which trucks are apart from the rollover risk or not. In Figure 9, the roll angle oscillates around zero closely means that the truck stays stable, apart from the rollover risk. Therefore, from Figure 9, it is verified that two heavy-duty trucks with anti-rollover control based on the proposed automatic ABS are obviously more stable and safer than the ones without anti-rollover control.

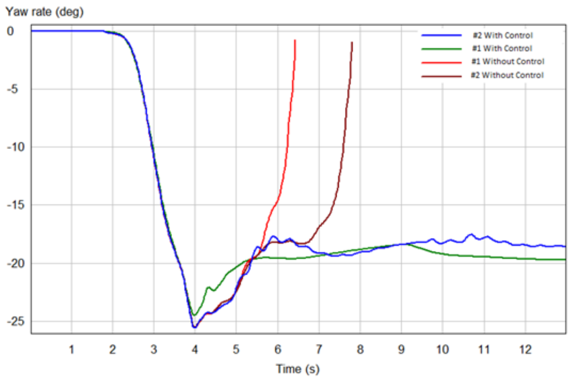


Fig. 8 Yaw Rate comparison.

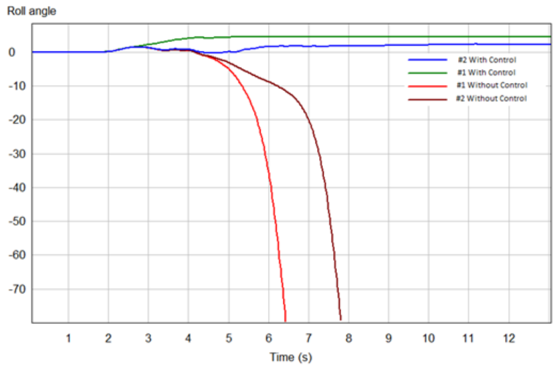


Fig. 9 Roll angle comparison.

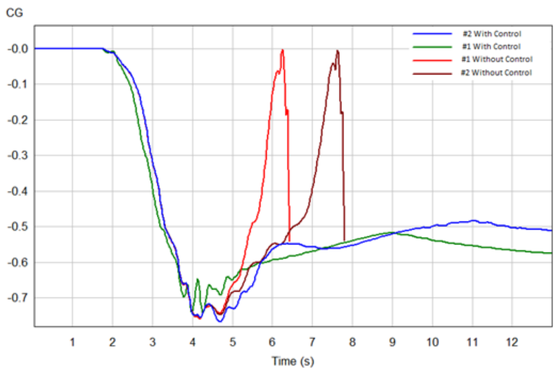


Fig. 10 Center of Gravity

In addition, Figure 10 compares the dynamic vehicle gravity center of two trucks with and without anti-rollover control based on the proposed automatic ABS, respectively. The more significantly the dynamic vehicle gravity center increases, the more closely the truck approaches the dangerous situation of rollover. From Figure 10, it is pointed out that the dynamic vehicle gravity center of trucks 1 and 2 without anti-rollover control begins to raise up significantly and fall into the rollover risk seriously at the second of about 6 and 7.5, respectively. This is because truck 2 itself has better anti-rollover control and stability recovery than truck 1 itself in the designed experiment.

4. CONCLUSIONS

In this paper, the integrated stability control scheme with the proposed automatic ABS increases the stability of autonomous heavy-duty trucks or forklifts. By increasing the brake force elaborately, the tire slip ratio raises the braking force coefficient and simultaneously reduces the slip effect. ABS plays a crucial part in regulating the speed of the wheel on slick and loose surfaces. Most autonomous heavy-duty trucks in such conditions are prone to losing stability control. The proposed automatic ABS has four sensors to detect wheel speed, vehicle speed, electrical and hydraulic valves. The simulation result shows that the proposed stability control by the proposed automatic ABS can work well on two models of two different vehicle dynamics even under dangerous road conditions of sharp corner or hairpin turn.

In the future, the proposed model will be applied to a heavy-duty truck with different loads of cargo corners or switches to different lanes with the implementation of the ABS

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