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 in autonomous heavy-duty trucks or forklifts. This paper develops an automatic Anti-lock Brake System (ABS) as one way to stabilize the autonomous vehicles. Then, through monitoring both vehicle's and wheel's speeds, this paper helps the autonomous vehicle keep stability control with automatic ABS 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 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 demonstrates that autonomous heavy-duty trucks are likely harmful to road safety during logistics, such as rollovers, shimmy, or jackknives due to the high mass center, comparatively big size, or relatively small base [1]. The most significant hazard to road safety is the rollover of autonomous heavy-duty trucks, which can lead to catastrophic results and significant losses [2].

A vehicle rollover is a phenomenon of vehicle instability caused by extreme steering or external excitation of the road. 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 [3].

Unconsidered rollover events are typically caused by high speed or extreme roll due to unexpected track modifications. If a vehicle crosses a curved road because of its centrifugal force, it leans outside the curve. The roll-up is possible if the autonomous vehicle misjudges the curve sharpness and keeps excessively fast because of his lateral acceleration tolerance [4].

The chance of a rollover accident might be reduced by a combined rollover risk assessment and active roll control under difficult driving situations. When rollover dangers can be predicted, information may be given to the autonomous vehicle or the stability controller. Proper reaction from the autonomous vehicle or the stability controller can prevent the rollover accident effectively and efficiently. The active rotating control is the best solution if the autonomous vehicle does not react appropriately [5]. Many researchers are analyzing these stability issues and improving the autonomous heavy-duty trucks or forklifts to lessen the occurrence probability of the rollover.

Because automatic Anti-lock Brake System (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, the slip ratio may be estimated by a specific speed sensor. This paper proposes to adopt automatic ABS to keep stability control of autonomous heavy-duty trucks or forklifts. So the 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 definition of autonomous heavy-duty vehicle modeling. Then, in Section 3, it explains the simulation and result in MATLAB/Simulink and TruckSim. Section 4 discusses the simulation result, and the last section remarks 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, dividing the spring weight between the front and back. Considering the front-to-back connective bobbin, the optimum torsional bar works with the stiffness of the torsion and has no mass for the front and rear spring systems [5], as indicated in Figure 1.

In view of the equation of different degree 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:

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

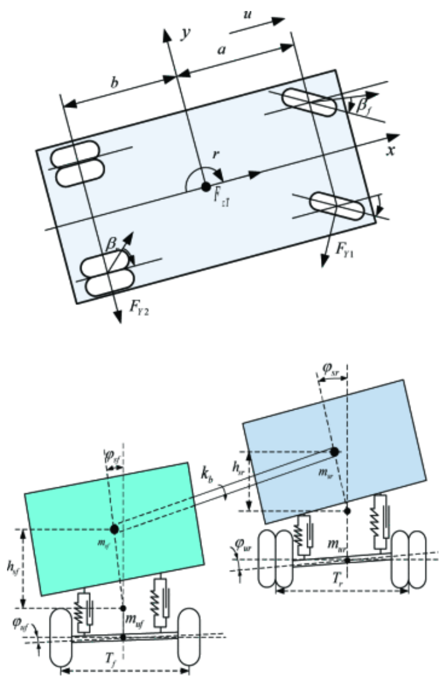

Fig. 1. Dynamics model of the autonomous truck.

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：

$$I_{xf}\ddot{\varphi}_{sf} = -k_r(\varphi_{sr} - \varphi_{ur}) - l_r(\dot{\varphi}_{sr} - \dot{\varphi}_{ur}) + m_{sr}h_r a_y + m_{sr}h_r\varphi_{sr} + k_b(\varphi_{sr} - \varphi_{sf}) \tag{5}$$

Roll motion of front unsprung mass：

$$2F_{Y2}h_c + m_{uf}(h_{uf} - h_{cf})a_y = -m_{uf}g(h_{uf} - h_{cf})\varphi_{uf} + k_{uf}\varphi_{uf} + k_f(\varphi_{sf} - \varphi_{uf}) - l_f(\dot{\varphi}_{sf} - \dot{\varphi}_{uf}) \tag{6}$$

Roll motion of rear unsprung mass：

$$2F_{Y2}h_c + m_{ur}(h_{ur} - h_{cr})a_y = -m_{ur}g(h_{ur} - h_{cr})\varphi_{ur} + k_{ur}\varphi_{ur} + k_r(\varphi_{sr} - \varphi_{ur}) - l_r(\dot{\varphi}_{sr} - \dot{\varphi}_{ur}) \tag{7}$$

Lateral acceleration at the center of mass of the vehicle:

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

Longitudinal displacement of the vehicle：

$$\dot{X} = u\cos\psi - v\sin\psi \quad (9)$$

Lateral acceleration at the center of mass of the vehicle:

$$\dot{Y} = u\sin\psi - v\cos\psi \quad (10)$$

Where *m* refers to the total mass of the autonomous heavy-duty vehicle, the front and rear axles are indicated by *{f,r}*. m_s and m_u 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* implies the length between the sprung mass center and the rolling center, so h_u and h_c are measured upwardly 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 is the roll inertia of the sprung mass. *u* is the longitudinal speed, and *v* denotes the lateral velocity. k and k_u are the suspension and the unsprung mass of the equivalent roll stiffness coefficients, respectively. *l* 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_u and φ_s 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:

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

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

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 [6]:

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

Where:
ω = angular velocity of the wheel
r = the effective wheel radius.

With the increasing 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 [6].

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 [7].

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

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) [7].

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.

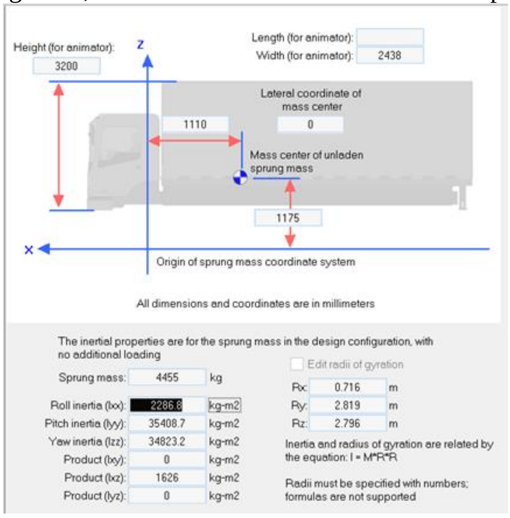

Fig. 2. Vehicle dynamics modelling by TruckSim.

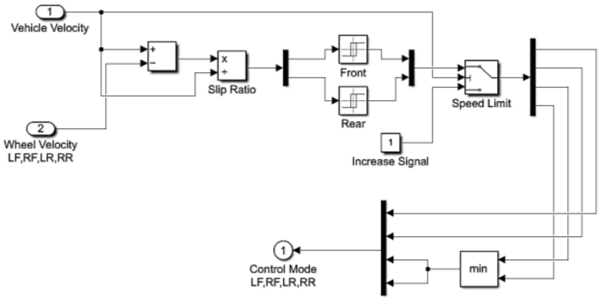

Fig. 3. ABS control model in MATLAB/Simulink.

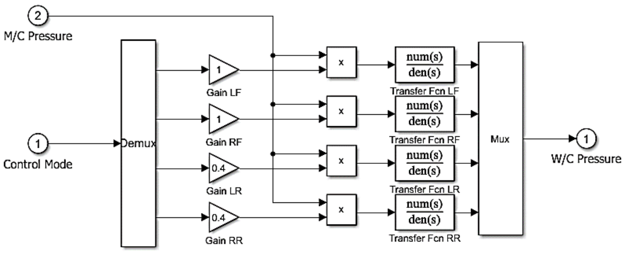

Fig. 4. Brake actuator model in MATLAB/Simulink.

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 vehicle 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 vehicle 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 comprehensive comparison, this experiment designs two models of two different vehicle dynamics for autonomous heavy-duty truck simulation, which are listed in Table 1 and Table 2, respectively.

A series of simulation results are performed and shown in Figures 5-10. Figure 5 shows the lateral tracking of the vehicle running on some straight or curved road situations. It means that the vehicle may track the trajectory to recover the vehicle stability and then follow some curved road situation with elaborate longitudinal speed control, as dynamically depicted in Figure 6. In Figure 7, the vehicle's pitch angle begins oscillating at the second of 13 while the variation of lateral tracking becomes larger, as shown in Figure 5. This is because the vehicle expects to decrease its longitudinal speed by proposed automatic ABS, as shown in Figure 6, and rotate its steering angle to follow the curved road situation, as shown in Figure 5. In Figure 7, the vehicle's pitch angle has been oscillating during the second of 13 to 22, and its ripple drops from the degree of -0.540 to the degree of-0.590 because of the vehicle speed's boost-up since the second of 22. Then the yaw angle of sprung mass in figure 8 has kept constant until the second of 13. Afterward, the yaw angle of sprung mass has been increasing up until the second 23.6 to make the vehicle follow the curved road situation, as shown in Figure 5. Finally, Figures 9 and 10 verify that the vertical tire forces of vehicle 1 and vehicle 2 has kept

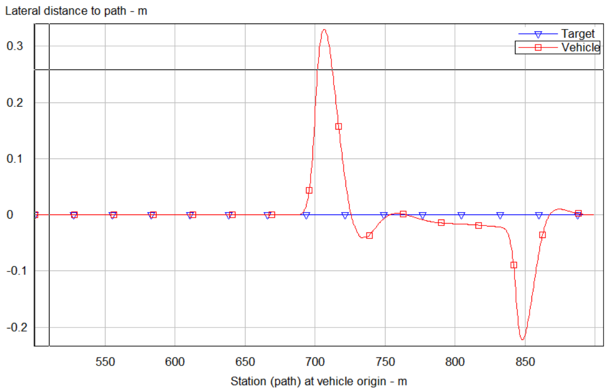

Fig. 5. Vehicle's lateral tracking under some road situation.

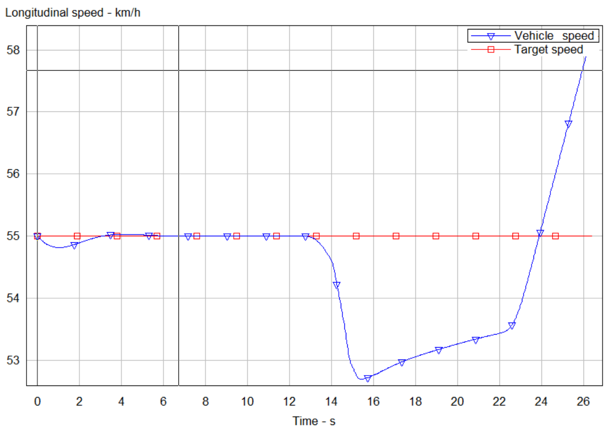

Fig. 6. Longitudinal speed and mass center.

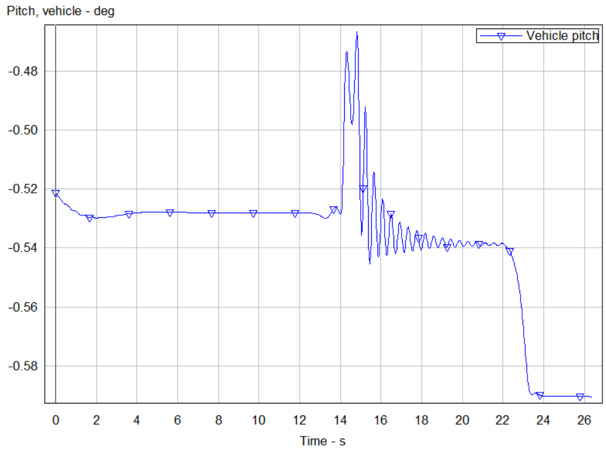

Fig. 7. Pitch angle of sprung masses.

the symmetry variation between the vertical tire force of L and R to ensure the vehicle's four tires to contact closely with the road surface all the time, through the proposed automatic ABS with confined slip ratio.

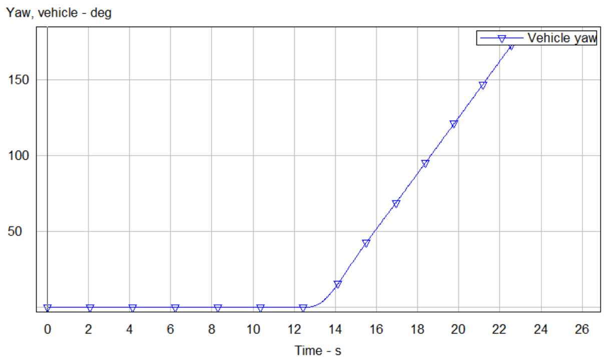

Fig. 8. Yaw angle of sprung masses.

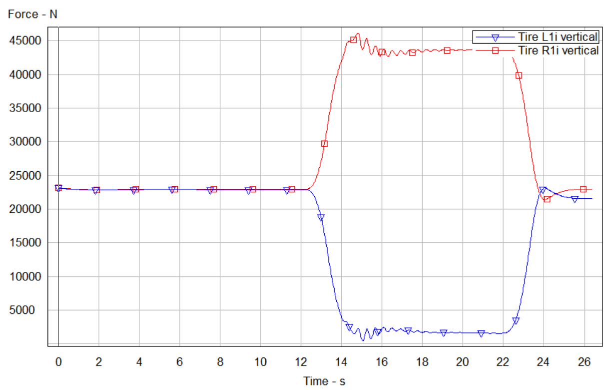

Fig. 9. Vertical tire forces of vehicle 1.

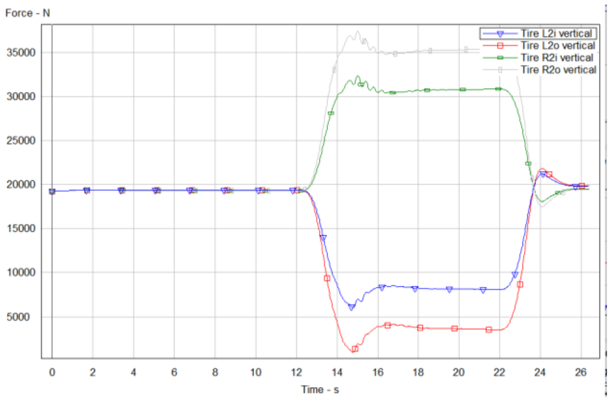

Fig. 10. Vertical tire forces of vehicle 2.

4. Conclusions

In this paper, the integrated stability control scheme of autonomous heavy-duty trucks or forklifts increases the stability by proposed automatic ABS. By increasing the brake force elaborately, the tire slip ratio raises the braking force coefficient and ultimately reduces the slip ratio. 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 tend to lose control of stability. The ABS has four sensors of wheel speed, vehicle speed, electrical and hydraulic valves. The simulation result shows that the proposed stability control by automatic ABS can work well on two models of two different vehicle dynamics even under dangerous road conditions of sharp corner or hairpin turn.

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Authors Introduction

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