Analysis of Stability and Dynamic Model Simulation of Mountain Tractor Rollover

Hongtu Yang¹,², Changgao Xia¹, Jiangyi Han¹, Chen Chen¹, Honggang Zhang¹,²

¹School of Automotive and Traffic Engineering, Jiangsu University, Zhenjiang 212013, China
²Department of Vehicle Engineering, Changzhou Vocational Institute of Mechatronic Technology, Changzhou 213164, China

*Corresponding author’s e-mail: xiacg@ujs.edu.cn

Abstract. A rollover model for predicting the operation of mountain tractors in hilly areas is presented. The quasi-static rollover model and the transient rollover model of the mountain tractor are studied, and the dynamic equations are also derived. The simulation model is set in the Carsim/Matlab Simulink environment, the rollover threshold of the tractor when cornering with a certain lateral acceleration under the angular step condition is analyzed. The anti-rollover control is used when one of the four wheels loses contact with the slope surface. The most significant result is that the model suggests that employment of active rollover control of the wheels may hinder the tractor from rolling on the slope. Thus, it proves the rationality of the model presented.

1. Introduction

Vehicle rollover is one of the most serious accidents that threaten the safety of drivers and passengers while the vehicle is in motion. Rollover can be defined as an extremely dangerous lateral movement, which the vehicle rotates 90° or more about its longitudinal axis during traveling, such that the vehicle loses contact with the ground. Rollover can be generated by one or a series of factors, including vehicle structure, driver and road conditions. Vehicle rollover can be roughly divided into two categories, one is maneuver induced rollover caused by curve motion, and the other is tripping rollover[1].

At present, some research on the field of vehicle roll stability and vehicle rollover have been carried out by many scholars at home and abroad, and corresponding mathematical models are established[2-5]. In recent years, with the rapid development of global agricultural mechanization, tractor rollover accidents with high casualty rate and serious economic losses have attracted people’s attention[6]. At the same time, some roll stability and rollover model are also analyzed. However, the tractor rollover model studied by the predecessors is limited to regular driving and flat roads. There are few studies on tractor working conditions with sloped roads, especially for mountain tractors in hilly areas. Due to the impact of uneven road surface, when the tractor passes through sloped road surface, the tractor lateral pressure caused by the ground soft or other obstacles increases. In severe cases, the tractor will“fall out”to some extent. Especially when the tractor turns around on a sloped road, the possibility of rollover is greater. In regular field operations, the working environment of the tractor is relatively flat and the slope of the ground is not large. Since the mountain terrain is complex and the
ground is uneven, the tractor has poor driving stability, the body is tilted and it is easy to roll over. Therefore, the study of the rollover stability of mountain tractor is crucial.

To gain further understanding of the underlying lateral rollover, the mountain tractor's rollover stability based on the slope with a lateral slope angle of about 15° is studied in this paper. The mathematical model of the rollover of the mountain tractor is established, and the dynamic behavior of the simulation model is tested in Carsim/Matlab Simulink environment. The influence of the active anti-rollover control system on the rollover stability of the mountain tractor is analyzed under the angular step condition with a certain lateral acceleration steering.

2. Rollover Model and Dynamics Equations

To study the driving and operation of the mountain tractor on horizontal slope, the mountain tractor is set to drive at a low speed, and the horizontal slope angle is about 15°. Due to the negligence of suspension deformation of the mountain tractor, whose chassis can be regarded as a rigid chassis. In the roll plane, the entire tractor body is required to keep leveled status, and this leveling is achieved through an electronically controlled hydraulic leveling system mounted on the tractor body.

2.1 Quasi-static Rollover Model

The roll-plane tractor vehicle model is shown in Fig 1. Here the mountain side slope angle $\beta$ is small, that is $\sin \beta \approx \beta$, $\cos \beta \approx 1$. The kinetic equations of motion for this system are given below[7].

$$
\left( m_b + m_f + m_r \right) a_y h_G - \left( m_b + m_f + m_r \right) g \beta h_{CG} + F_{zi} B - \frac{1}{2} \left( m_b + m_f + m_r \right) g B = 0
$$

$$
a_y = \frac{1}{2} \beta \frac{F_{zi} B}{h_{CG}} - \frac{\left( m_b + m_f + m_r \right) g}{h_{CG}} = \left[ \frac{1}{2} - \frac{F_{zi}}{\left( m_b + m_f + m_r \right) g} \right] \frac{B}{h_{CG}} + \beta
$$

It can be seen from equation 2-2, as the lateral acceleration $a_y$ increases, the force $F_{zi}$ on the left wheel gradually decreases. When $F_{zi}$ is reduced to zero, the tractor is unable to maintain balance in the roll plane and starts to roll over. In the above equation, $a_y$ is the lateral acceleration on the body fixed coordinate located at the gravity centre. Equation 2-2 is often used to estimate the anti-rollover capability of the mountain tractor, which can be presented in the form:

![Figure 1. Analytical model for roll motion of tractor](image1)

![Figure 2. Transient rollover model of mountain tractor](image2)
2.2 Transient Rollover Model

Figure 2.2 shows the rollover model of the tractor with side acceleration $a_y$. In the process of modeling, the entire tractor is regarded as an equivalent combination of body mass and axle wheel mass (that is, front and rear wheel axle system), and the mass center of the body mass is approximately assumed to be on the roll axis. The mountain tractor's roll, yaw, lateral motion is mainly considered into during the tractor model establish. Certain assumptions are considered appropriate in order to research the new tractor rollover model. The equations of motion for this system are derived by the use of Lagrange’s equation and are given below[8-10].

Lagrange's equation expression is defined as: 
\[ L = E_T - E_V \]  
(3)

The following nonlinear ordinary differential equation are obtained:

(a) Coordinate $Y$:
\[ (m_b + m_f + m_r) y (-\cos \beta) - m_f p \varphi_u \cos \varphi_u - m_b h_r \varphi_h \cos \varphi_h - m_r p \sin \varphi_u + m_b p \varphi_u^2 \sin \varphi_u + m_f h_r \varphi_h^2 \sin \varphi_h - 2 m_b p \varphi_u \cos \varphi_u = (F_{3f} + F_{3r}) \cos \beta \]  
(4)

(b) Coordinate $Z$:
\[ (m_b + m_f + m_r) z u - m_f p z u \sin \varphi_u - m_b h_r \varphi_h \sin \varphi_h + m_r p \cos \varphi_u - m_b h_r \varphi_h^2 \cos \varphi_h - 2 m_b p \varphi_u \sin \varphi_u - m_r p \varphi_u^2 \cos \varphi_u = 2 \cos \beta (F_{3f} + F_{3r}) - m_b g - (m_f + m_r) g \]  
(5)

(c) Coordinate $I_{uf}$:
\[ (I_{uf} + m_u p^2) \varphi_u - m_b p z u \sin \varphi_u + m_r p y \cos \beta \cos \varphi_u + m_b ph_r \varphi_h \cos (\varphi_h - \varphi_u) + m_b h_r \varphi_h^2 \sin (\varphi_u - \varphi_h) + 2 m_b p \varphi_u \sin \varphi_u + (F_{3l} - F_{3r}) B \cos \beta + \frac{F_{uf} (h_u + z_u)}{\cos \beta} \]  
(6)

(d) Coordinate $I_{ur}$:
\[ (I_{ur} + m_u p^2) \varphi_u - m_b p z u \sin \varphi_u + m_r p y \cos \beta \cos \varphi_u + m_b ph_r \varphi_h \cos (\varphi_h - \varphi_u) + m_b h_r \varphi_h^2 \sin (\varphi_u - \varphi_h) + 2 m_b p \varphi_u \sin \varphi_u + (F_{3l} - F_{3r}) B \cos \beta + \frac{F_{ur} (h_u + z_u)}{\cos \beta} \]  
(7)

(e) Coordinate $p$:
\[ m_b p + m_f \varphi_u \cos \varphi_u + m_r \varphi_u \cos \beta \sin \varphi_u - m_f h_r \varphi_h \sin (\varphi_h - \varphi_u) - m_r h_r \varphi_h^2 \cos (\varphi_h - \varphi_u) - m_b p \varphi_u^2 = -m_b g \cos \varphi_u \]  
(8)

(f) Coordinate $\varphi_h$:
\[ (I_h + m_h h_r^2) \varphi_h - m_b h_r \varphi_h \sin \varphi_h + m_f h_r \varphi_h \cos \varphi_h + m_r h_r \varphi_h \cos (\varphi_h - \varphi_u) - m_f h_r \varphi_h^2 \sin (\varphi_h - \varphi_u) + 2 m_f h_r \varphi_h \cos (\varphi_h - \varphi_u) + m_r h_r \varphi_h^2 \sin (\varphi_h - \varphi_u) = m_b h_r \varphi_h \sin \varphi_h \]  
(9)

Table 1 Parameters for the mountain tractor

| Symbol | Definition |
|--------|------------|
| $m_b$  | mass of the tractor body mass (kg) |
| $m_f$  | mass of the tractor front axle and front wheels |
| $m_r$  | mass of the tractor rear axle and rear wheels |
| $h_cG$ | height above ground of the tractor mass center |
| $h_u$  | height above ground of the tractor front or rear axle center |
| Symbol | Description |
|--------|-------------|
| $h_r$  | vertical distance from the tractor mass center to the roll center |
| $F_{yf}$ | lateral force acting on the front wheel |
| $F_{yr}$ | lateral force acting on the rear wheel |
| $B$ | wheel base |
| $a_y$ | lateral acceleration |
| $\beta$ | slope angle [rad] |
| $F_{zL}$ | vertical load on the left tires |
| $F_{zR}$ | vertical load on the right tires |
| $I_b$ | roll moment inertia of the tractor mass |
| $I_{uf}$ | roll moment inertia of the front axle and wheel |
| $I_{ur}$ | roll moment inertia of the rear axle and wheel |

3. Results and Analysis
An example of a system simulation of a mountain tractor rollover is given by the software Carsim. Fig. 3 shows different curves of vertical force of the tractor tire as the tractor rolls over. The forces $F$ represent the vertical tire forces acting on the left or right side of the particular axle. Since the process of tractor rollover is supposed to be initiated at the trailer rearward side, the force could be used as an early indication that the relative roll instability condition had been reached. Fig. 4 shows the lateral acceleration as the body rolls over. As the tractor rolls, tires on one side are compressed while those on the other are unloaded. The as the tractor starts to roll in the other direction, the tire compression force is released, which causes the tractor to bounce up and then drop down, causing a sudden compression of the tire, as shown by sharp peaks in tire forces. Thus, the roll and the vertical jumping motion of the tractor are coupled the tire properties.

Figure 3. Vertical tyre forces acting on the left or right side
Figure 4. Lateral acceleration of the tractor body

Figure 5. The displacement of the tractor body in roll over status

Figure 6. The displacement of the tractor body in leveling status

Fig. 5 and Fig. 6 show the displacement of the gravity center of the tractor body with time under the condition of rollover and leveling status. It can be seen from the figure that under the conditions of leveling and non-leveling, the displacement of the center of gravity of the body in the vertical direction shows a downward trend with time. As it can be induced above, the vehicle starts to slide...
down on the 30° lateral slope, which can be defined as the stable. This critical point is also called the critical threshold for rollover.

4. Conclusion
A nonlinear analytical model for studying the roll motions of mountain tractor has been presented above. The models related to the front and rear wheels are prepared and solved in CarSim and the solutions are imported thereafter to CarSim. The simulation model is set in the Carsim/Matlab Simulink environment, which is valid in obtaining roll limits of the mountain tractor. The rollover threshold of the mountain tractor when cornering with the lateral acceleration under the angular step condition is analyzed. It has been shown here that the actual roll dynamics of such mountain tractor is dependent on the time history of applied lateral force and can induce rollover lateral acceleration critical threshold as well.

Acknowledgments
The authors would like to thank the National Key R&D Plan of China (NO. 2016YFD0700402) for the support given to this research.

References
[1] Guzzomi, A.L. (2012) A revised kineto-static model for Phase I tractor rollover. Biosystems engineering., 113: 65-75.
[2] Azad, N.L., Khajepour, A., McPhee, J. (2009) A survey of stability enhancement strategies for articulated steer vehicles. International Journal of Heavy Vehicle Systems., 16: 26-48.
[3] Chisholm, C.J. (1979) A mathematical model of tractor overturning and impact behavior. Journal of Agricultural Engineering Research., 24: 375-394.
[4] Yu, Z.S. (2000) Automobile theory. Machinery Industry Press Publishing, Beijing.
[5] Guzzomi, A.L., Rondelli, V., Guanieri, A. (2009) Available energy during the rollover of narrow-track wheeled agricultural tractors. Journal of Biosystems Engineering., 104: 318-323.
[6] Verma, M.K., Gillespie, T.D. (1980) Roll dynamics of commercial vehicles. Vehicle system dynamics, 9: 1-17.
[7] Yu F., Lin Y. (2016) Automobile System Dynamics. Machinery Industry Press Publishing, Beijing.
[8] Rondelli, V., Guzzomi, A.L. (2010) Selecting ROPS safety margins for wheeled agricultural tractors based on tractor mass. Journal of Biosystems Engineering., 105: 402-410.
[9] Hasagasioglu, S., Kilicaslan, K., Atabay, O. (2012) Vehicle dynamics analysis of a heavy-duty commercial vehicle by using multibody simulation methods. International Journal of Heavy Vehicle Systems., 60: 825-839.
[10] Kumar, A., Pandey, K.P. (2012) A device to measure dynamic front wheel reaction to safeguard rearward overturning of agricultural tractors. Computers and electronics in agriculture., 87: 152-158.