Study Of The Effect Of Single-Parameter On Light Vehicle Rollover Tendency Based On LTRd

Shengqin LI, LI ZHAO, YU ZHANG

Traffic College, Northeast Forest University Harbin, China

Abstract: In the paper, taking a light vehicle as the research object, a three degrees of freedom multi-body dynamics model is developed in MATLAB/Simulink, to analyze light vehicle dynamics and rollover stability. Using the J-turn and NHTSA Fishhook 1a virtual tests, the effects of vehicle velocity and steering angle on vehicle gesture, such as lateral acceleration and body roll angle, are analyzed. As the single-factor in the tests, the dynamic lateral-load transfer ratio (LTRd) is proposed, to study the effect of vehicle structure, traffic conditions, and external conditions on the rollover tendency. A correlation between the vehicle parameter of center of gravity location and rollover tendency is studied, and the results show that the vehicle center of gravity is the most important parameter which affects the rollover tendency of light vehicle. At the same time, a vehicle with increased suspension stiffness can improve its rollover resistance capability in the fishhook maneuver. The LTRd can be used as rollover index to judge the vehicle gesture, then to control vehicle rollover stability. The results can be used in light vehicle rollover control research, to improve risk prediction, and reduce the incidence of traffic accidents.

Keywords: light vehicle; LTRd; single-parameter; rollover stability; virtual test; MATLAB/Simulink

1. Introduction

With the rapid development of transportation industry, vehicle major accidents have also increased due to rollover, causing serious casualties and huge economic losses. Rollover crashes is one of the most significant safety problems for all classes of light vehicles especially light trucks (pickups, sport utility vehicles, and vans - LTVs). These rollovers resulted in an average of 9,063 fatalities per year (29 percent of all light vehicle fatalities) and over 200,000 non-fatal injuries per year [1]. According to the 2000 Fatality Analysis Reporting System (FARS), 9,882 people were killed as occupants in light vehicle rollover crashes, including 8146 killed in single-vehicle rollover crashes[2]. FARS shows that 53 percent of light vehicle occupant fatalities in single-vehicle crashes involved a rollover event. In terms of fatalities per registered vehicle, rollovers are second only to frontal crashes in their level of severity. The rollover problem is more serious for light trucks, especially sport utility vehicles. State crash data indicates that, for all types of collisions, LTV’s are only in 68 percent as many crashes per registered vehicle as they are passenger cars[3]. So it has been the focus of the world’s vehicle research, to study and enhance the anti-rollover capability of vehicle, and reduce the incidence of rollover accidents.

Light vehicle has a high center of mass and relatively small suspension stiffness, therefore, it is easy to have a greater body roll angle and lateral load transfer when the vehicle is steering[4][5]. When the vehicle change lane or obstacle avoidance emergency on the high way, it is easy to lead to a great lateral acceleration and roll angle, causing a lot of lateral load transfer. When these exceeds the limit of the tire lateral load transfer, that is the vertical load of the inner wheel is zero, the rollover is very likely to occur. At present, the anti-roll stability study on buses and passenger cars is relatively mature, but it is relatively less for the anti-minivan rollover stability studies. Since some light vehicles are still used as school buses in most city or suburban district, it is necessary to study the anti-rollover stability of the minibus.

At present, the study methods on the light vehicle rollover are the following [6]: to establish vehicle system dynamics model using multi-body dynamics simulation software, and do simulation study on the stability of the vehicle during steering, this method needs some detailed geometry parameters of vehicle, which are difficult to get; sometimes we can do some vehicle test study, and validate and modify the appropriate control methods according the real vehicle operation, but the experimental investigation of vehicle rollover crashes are not only expensive and time consuming but also limited to those maneuvers that can be physically reconstructed; dynamic simulation has proven to be an efficient and accurate method for analyzing vehicles and evaluating their dynamic behavior. Based on the simplifying of the vehicle and theoretical analysis of the minibus system, we can do some vehicle rollover propensity research, using the mathematical motion equation of state of the vehicle while steering, which can be more simple and accurate enough. LI Zhigang has proposed a multi-body dynamics model to analyze mini-bus dynamics and sharp turn stability[7], the influences of mini-bus structure, traffic conditions, and external conditions on the rollover tendency were also analyzed in single-factor experiments. The results show that the road adhesion coefficient is the most important parameter with the center of gravity and other parameters also having some influence on the rollover tendency.

In the paper, a three degrees freedom simplified model that captures the essential vehicle dynamics associated with un-tripped rollover is developed and validated, using MATLAB/Simulink, in order to research the influence on the vehicle rollover tendency of the vehicle structure and operation parameters, the single-factor sensitivity on the vehicle rollover tendency is emphatically analyzed, which can lay a theoretical foundation for anti-rollover control on light vehicle.

2. Vehicle Dynamic Model

Vehicle rollover is a complex mechanical phenomenon that involves the vehicle, road, and the driver. Analysis of real life crashes is necessary in order to understand the mechanism of rollover and develop any potential countermeasures for occupant protection[8][9][10]. The most widely used vehicle rollover model is a three degrees of freedom (3-DOF) model, which includes vehicle lateral velocity, vehicle yaw velocity and body roll angle. In order to build the vehicle dynamic model, we should make some hypothesis[10-13]:

(1) The input of the model is front wheel steering angle, and the influence of steering system is ignored.
(2) The influence of lateral wind on the vehicle is ignored.
(3) Pitch dynamics (which cause longitudinal weight transfer) are neglected since longitudinal accelerations were kept small.
(4) The forward speed \( v \) is constant.
(5) Some nonlinear factors of tires and suspensions are ignored.
(6) The influence on the tire roll characteristic of the ground tangential force and cornering force is ignored.
(7) It is supposed that the two front or rear wheels have the same trajectory.
(8) It is supposed that the two wheels at same axe have the same wheel angle and roll angle, and the change of the tire characteristics due to the speed, pressure and other factors is ignored.
(9) The aligning torque on the tire is ignored.
(10) When the tires contact with the ground, the height between tire rolling center and ground is ignored.

3. Vehicle Model Description
The three degrees of freedom simplified model is shown in Figure 1, based on the hypothesis above. This 3-DOF model is used widely in the study of rollover warning and control system, can describe the vehicle attitude while steering, and reflect the vehicle's rollover tendency [14][15]. The model is simply, but includes most parameters of the vehicle, and can be used to provide the theoretical basis for the research of vehicle rollover stability.

According to the hypothesis above, a three degrees of freedom (3 DOF) vehicle simulation model is developed as the base for the filter design and for the source model, which is used to simulate a real vehicle, the three degrees of freedom are lateral freedom, yaw freedom, and roll freedom, the schematic structure of vehicle is shown in Figure 1.

Additionally, suspension kinematics is considered stationary. The suspension kinematics influences roll centers, track width, and introduces jacking forces during the translation and rotation of suspension members. However in this paper, the effects of suspension kinematics were neglected.

Assuming the longitudinal velocity \( u \) to be constant. Considering the lateral velocity \( v \), yaw angle rate \( r \), and body roll angle \( \phi \), by applying basic kinematics, the non-linear equations of motion could be obtained,

\[
\begin{align*}
(m + m_f + m_j)(\dot{v} + ur) + (am_f - bm_r)v + m_h \phi = F_y + F_r
\end{align*}
\]

(1)

\[
(\dot{am}_f - bm_r)v + L_{ax} \dot{r} + L_{az} \dot{\phi} = a F_y - b F_r
\]

(2)

\[
I_{ax} \phi + D_{ax} \dot{\phi} + (K_{ax} - m_{bh} \phi) \phi + m_{bh}(\dot{v} + ur) + I_{ax} \dot{r} = F_y d_f + F_r d_r
\]

(3)

Where, \( I_{ax} \) is the mass moments of inertia of the vehicle body about \( x \) axe, \( I_{az} = I_{ax} + m_{bh} \phi^2 \); \( I_{az} \) is the mass moments of inertia of the vehicle body about \( z \) axe, \( I_{az} = I_{ax} + I_{ay} + m_{bh} \phi^2 \); \( I_{ax} \) is the product of inertia of the vehicle body about \( x \) and \( z \) axes. \( d_f = h_y - h_{aw} \), \( d_r = h_y - h_{aw} \).

Assuming that the lateral forces acting on each tire, is directly proportional to the slip of that tire and that both tires on an axle share the same forces. This leads to the equations:

\[
F_y = -C_{aw} \alpha_f = C_{aw} (\frac{v + ur}{u} - \delta_f)
\]

(4)

\[
F_r = -C_{aw} \alpha_r = C_{aw} (\frac{v - br}{u})
\]

(5)

Considering equations (3), (4) and (5), the matrix form can be obtained, as shown:

\[
\begin{align*}
0 & m_h & 0 & m & a m_f - b m_r & 0 & v \\
0 & 0 & I_{ax} & a m_f & a m_f - b m_r & I_{ax} & 0 \\
0 & I_{az} & 0 & m_h & 0 & D_{ax} & \phi
\end{align*}
\]

(6)

The equation (6) is the 3 degrees of freedom vehicle reference model for handling performance analysis.

4. Model Validation

In order to validate the vehicle model described above, simulation results were compared to experimental data for the snake maneuver. The snake maneuver uses a steering input consisting of a sine steer at a set entrance velocity. Figure 2 and figure 3 shows the contrast of curves of simulation and experiment under the snake maneuver and double lane change, the vehicle forward speed is 60 km/h, it can be seen that, the curves of simulation and experiment are in good agreement, except that the yaw rate is less than the test value at some point, it may be the reason of some simplifies of vehicle structure. So it is considered that, the simulink model can reflect the basic characteristics of vehicle, and can be used to simulate and analyze the vehicle rollover stability and tendency.
5. Virtual Tests

More commonly tests used to evaluate the vehicle stability are J-turn test and NHTSA Fishhook 1a test. According these two test standards, some virtual tests are proposed in the paper, in the test, the adhesion coefficient of road is set as 0.9, the height of vehicle center of gravity is set as 800 mm.

Both the Fishhook 1a maneuver and J-turn test use the same initial steering input. The steering input consists of an initial steer followed by a counter steer at a set entrance velocity. The velocity profile of this maneuver is characterized by the vehicle reaching a desired steady state speed, known as the entrance speed, and coasting through the rest of the maneuver once the initial steer is begun. The profile consists of an initial zero steer angle followed by going to a steer angle ‘A’ at a rate of 720 degrees per second at the handwheel. ‘A’ is specific to each vehicle configuration, and is defined by multiplying 6.5 by the steer angle of the handwheel at which the vehicle experiences 0.3 g of lateral acceleration in a Slowly Increasing Steer (SIS) maneuver. The SIS maneuver is performed at a constant velocity of 50 mph with a continually increasing steer input of 13.5 degrees per second at the handwheel. ‘A’ is set as 27.5 degree in this paper.

6. Fishhook 1a Test

The initial velocity is 75 km/h, the profile consists of an initial zero steer angle followed by going to a steer angle ‘A’ at a rate of 720 degrees per second at the handwheel, until the steer angle reach to ‘6.5A’. The steer angle ‘6.5A’ is held constant for 0.250 seconds then a countersteer to ‘-6.5A’ at the same rate occurs. The steer angle ‘-A’ is held constant for 3 seconds, after which it returns to zero, completing the maneuver. A graph of this profile is shown in Figure 3. In the paper, the input of the model is front wheel angle, so the handwheel input should be divided by 15.6, which is the angular gear ratio of steering system. Figures 4 shows the lateral acceleration and roll angle graphs resulted from the simulation model.
It can be seen from figure 5 that, the later acceleration graph has similar trends with the handwheel input, the maximum of later acceleration reaches to 1g when the vehicle forward speed is 50 km/h, almost reaches the limit of rollover. The body roll angle has similar trends with the handwheel input too, and the maximum value is 4 degree. That is due to the steering wheel angle rate is large, and the vehicle velocity is high at the same time, which can superimpose affect the sensitivity of vehicle rollover tendency.

7. J-Turn Test

The graph of this profile in the J-turn test is shown in Figure 6. The initial velocity is 80 km/h, known as the entrance speed, and coasting through the rest of the maneuver once the initial steer is begun. The profile consists of an initial zero steer angle followed by going to a steer angle 'A' at a rate of 1000 degrees per second at the handwheel, until the steer angle reach to ‘8A’, then the steer angle is held constant for 4 seconds, after which it returns to zero, completing the maneuver. Since the input of the model is front wheel angle, so the handwheel input should be divided by 15.6. Figures 6 shows the lateral acceleration and roll angle graphs resulted from the simulation model.

It can be seen from figure 7 that, when the vehicle forward speed is 50km/h, while the handwheel angle changes to the opposite direction, the vehicle later acceleration increases rapidly, and the maximum value reaches to 1.3g; the maximum value of body roll angle reaches to 6.5 degree at the same time, the vehicle reaches to the critical instability state, but the vehicle can return to the stability state as the handwheel angle is held constant. It can be concluded that the vehicle velocity can superimpose affect the sensitivity of vehicle rollover tendency.

8. Study Of Single-Actor Influence On Rollover Tendency

Once the simulation model is validated, we can vary the vehicle parameters to access their effect on rollover tendency. The dynamic test used to determine the effects of these properties is the step input. By comparing the rollover velocities with the various vehicle properties, a correlation is identified between rollover tendency and those properties. In the paper, some parameters are selected as the analysis parameters to study the vehicle rollover stability, such as height of center of gravity, vehicle velocity, handwheel angle, and suspension stiffness. In order to analyze the influence of the single-factor on the vehicle rollover stability, one of the parameters is modified to study the later acceleration and body roll angle, while keeping the other three parameters unchanged.

The lateral-load transfer rate (LTR) is selected as the evaluation index to evaluate the vehicle rollover stability, which is a common and effective index to evaluate the vehicle rollover stability. When the vehicle is turning, the vertical force on the two inside tires will decrease, while the vertical force on the two outside tires will increase. The LTR is the difference between the left and right side of the tire vertical reaction vehicle and the ratio of the sum, its value changes between -1 and +1, the sign shows the direction of steering,

\[ LTR = \frac{F_{u1} - F_{u2}}{F_{u1} + F_{u2}} \]

When the vehicle drives in straight line, if the loads are symmetry, the vertical force on both tires are equal, and the value of LTR is zero; when the vehicle has roll motion, the force will transfer between the left and right tires, then the value of LTR isn’t zero; when one tire left ground, that is the vehicle has rollover motion, the force on the tire is zero, and the absolute value of LTR is 1. So the range of LTR is [-1, 1], and the absolute value of LTR is [0, 1], thus it can be seen that when LTR < 1, all the tires of vehicle are kept on the ground, and the vehicle is in the roll stability state; when LTR ≥1, the vehicle is rollover, the more the LTR value is, the worse the vehicle rollover stability is.

Since the LTR can’t be measured directly, the dynamic lateral-load transfer rate is proposed, that is \( LTR_d \). In figure 1, based on D’Alembert principle, the force balance equation can be written as follow,

\[ (F_{u1} - F_{u2}) \frac{L}{2} - m\alpha_H - k_d \phi - c_d \phi = 0 \]

(7)

\[ LTR_d = \frac{F_{u1} - F_{u2}}{F_{u1} + F_{u2}} = \frac{2}{m(H(\dot{\phi} + ur) + k_d \phi + c_d \phi)} \]

(8)

Equation (8) can set up the relationship between the vehicle condition and structure parameters and dynamic lateral-load transfer rate, thus the 3-DOF model can calculate the dynamic lateral-load transfer rate as the rollover degree index. Generally it is considered that, while the absolute value of \( LTR_d \) is larger than 0.8, the vehicle rollover will happen.

© 2016 DEStech Publications, Inc.
doi:10.12783/issn.1544-8053/13/7/154
9. Varying Cg Vertical Location

A series of simulation experiments that varied the CG height was performed on the virtual light vehicle to assess the effect of this parameter on rollover propensity. The light vehicle configuration used was the nominal case, and the only parameter that varied is the CG height. Figure 8 shows the corresponding simulation results. Since the track width is held constant, the value of $LTR$ varies with height between center of gravity and roll center from 0.1m to 0.5m. It can be seen that, the higher the height between center of gravity and roll center is, the larger the value of $LTR$ is, when the height between the center of gravity and roll center is 0.5m, the maximum value of $LTR$ reaches to 1, and at this time the body roll angle reaches to 40 degree, the vehicle is going to be rollover. The observed trend of the data is not surprising since it is well known that vehicles will rollover at a lower velocity if the height of CG is raised. So the height of center of gravity should be decreased on the premise of meeting the basic functions of vehicle.

![Figure 8. Effects of vehicle center of gravity](image8)

10. Varying Suspension Roll Stiffness

A series of simulation experiments that varied the vehicle suspension roll stiffness are performed on the virtual light vehicle to assess the effect of this parameter on rollover tendency. The light vehicle configuration used was the nominal case, and the only parameter that varied is the vehicle suspension roll stiffness. Figure 9 shows the corresponding simulation results. It can be seen that, when the vehicle suspension roll stiffness is 10000 N·m/rad, the absolute value of curve slope is bigger, that is, when the suspension roll stiffness is small, it has more effect on the vehicle rollover tendency. When the suspension roll stiffness is bigger than 10000 N·m/rad, the value of $LTR$ has small difference among different suspension roll stiffness, it has less effect on the vehicle rollover stability.

![Figure 9. Effects of suspension roll stiffness](image9)

11. Varying Vehicle Longitudinal Velocity

A series of simulation experiments that varied the vehicle longitudinal velocity are performed on the virtual light vehicle to assess the effect of this parameter on rollover propensity. The vehicle configuration used was the Nominal case, and the only parameter that varied is the vehicle longitudinal velocity. Figure 10 shows the corresponding simulation results. It can be seen that, the maximum dynamic load transfer rate is linear change with the increase of the longitudinal velocity. When the vehicle longitudinal velocity is over 90km/h, the maximum dynamic load transfer rate is 1.2, and the vehicle has been in a state of instability and the rollover propensity increases. When the vehicle is in normal speed range, the value of dynamic load transfer rate is in the stable range, the vehicle can operate stably.

![Figure 10. Effects of vehicle velocity](image10)

12. Varying Handwheel Angle

A series of simulation experiments that varied the vehicle handwheel angle was performed on the virtual light vehicle to assess the effect of this parameter on rollover propensity. The light vehicle configuration used was the nominal case, and the parameters that varied are front wheel angle, and the input of model is step signal. Figure 11 shows the corresponding simulation results. It can be seen that, when the...
vehicle speed is 50 km/h, the vehicle can be in the stable state when the front wheel angle is 20 degrees, the value of $LTR_d$ increases with the handwheel angle.

![Figure 11. Effects of steering angle](image)

13. Conclusion

This paper has shown that a simple vehicle dynamic model developed to study the transient dynamics of light vehicles can capture the dynamics seen in events leading to rollover. To evaluate a vehicle’s rollover tendency through simulation, a 3-DOF model of light vehicle is proposed in MATLAB/Simulink, to study the influence on the vehicle rollover tendency of some structure parameters and operation parameters.

(1) The 3-DOF model was validated by real vehicle test, and the parameters were provided by the vehicle factory. The result showed that the model can be used to analyze the vehicle rollover stability.

(2) Used NHTSA Fishhook 1a and J-turn maneuver, the vehicle virtual tests are proposed, to analyze the vehicle lateral acceleration and body roll angle. It can be concluded that, the vehicle longitudinal velocity have more effect on the vehicle posture when the vehicle were steering, so the rollover control should be combined with braking system.

(3) By introducing the load transfer ratio $LTR_d$: a system performance output could be obtained, which has relationship with the vehicle gesture, such as lateral acceleration, body roll angle and yaw angle. Its value provides an accurate measure for determining the onset of rollover, and could be used as control objectives in the light vehicle rollover control system.

(4) Four parameters were selected to study the vehicle rollover tendency, such as height of CG, vehicle speed, handwheel angle and suspension stiffness. The height of CG had more effect on the vehicle rollover tendency, when the height between center of gravity and roll center was 0.5m, the maximum value of $LTR_d$ reached to 1, and the body roll angle was 40 degree, the vehicle happened rollover.

Of course that, there are other vehicle parameters that play an important role with regard to rollover tendency besides the parameters mentioned in the paper. Future research will investigate the effect of other vehicle properties, such as suspension setup, tire models, etc., on rollover tendency using simulation and scaled experiments.

Acknowledgments

This work was supported by the Science fund project of heilongjiang province (Grant Nos. E2016003) and the Fundamental Research Funds for the Central Universities (Grant Nos. DL13CB07).

References


