Analysis of heavy vehicle rollover and stability

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Abstract

Features itself of heavy vehicle led to their poor roll stability and prone to rollover accidents. Vehicle rollover accidents are a significant traffic accidents resulting in loss of life and property, and have become an important issue affecting transport security. Exploration and research of heavy vehicle safety and stability control have become the important topic concerned all over the world. A 3DOF dynamics model is created with heavy truck for example to set the LTR (Lateralload Transfer Ratio) as the rollover evaluation index, study on the relationship between speed, the front corner and centroidal moment with rollover angle, lateral load transfer ratio, then analyse the influence of vehicle condition parameters on the roll stability with both cases of noload and loaded, and make a simulation in SIMULINK. The simulation results show that load mass and centroid position have the greatest impact on rollover stability, vehicle driving stability can be improved by reducing the height mass and increasing the track and wheelbase, the traditional warning method cannot completely reflect the state of rollover of heavy vehicles. All this work finally provides the basis for optimization of heavy vehicles bodies and improves of rollover stability of heavy vehicles.

Keywords: Heavy vehicles, Rollover stability, Rollover model, Lateral-load transfer ratio, Rollover angle.

1 Introduction

According to the statistics of U.S. National Highway Traffic Safety Administration (NHTSA), the damage of car rollover accident is only second to car crash accident in all the car accident [1]. With the feature of large load , high centre of mass and high aspect ratio, heavy vehicles prone to rollover accident when vehicles encounter limited conditions such as emergency steering, excessive speed and too-small turning radius and cause significant damage on economic development and people's life and property. So, exploration and research on safety of heavy vehicle driving and handling stability have become significant issues of concern around the world.

In recent years, scholars made a lot of research on rollover stability of the vehicle, put forward a variety of rollover evaluation and early warning algorithms. Rakheja proposes a rollover warning monitoring algorithm that with lateral acceleration of when wheels just off the ground as the critical threshold, it will warn if the value exceed [2]; Chen.B and Peng.H proposed a kind of real-time rollover warning algorithm based on predicted time (TTR), but for different models, rollover threshold will be different, this leading to the practicality of the algorithm subject to certain restrictions [3]; Dongyong Hyun and other people use lateral load transfer ratio (LTR) to conduct the research on vehicle rollover stability, the algorithm is simple and real-time and now have become the most widely used indicator of a vehicle rollover working conditions [4]; domestic research on vehicle rollover warning started lately, Guizhou University of technology [5], Nanjing University of Aeronautics and Astronautics [6], Jilin University [7], Chongqing University of traffic [8], Huaiyin Institute of Technology [9] and other universities have carried out some research and made some findings.

This paper has a research on rollover stability of heavy-duty vehicle. We will choose the lateral load transfer rate as the rollover evaluation indicator and establish the vehicle dynamics model, and then study the relationship between driving state parameters of heavy vehicles and the lateral load transfer rate, the result will provide basis for rollover warning and anti-roll control of heavy vehicles.

2 Establish of dynamic model

2.1 VEHICLE MODEL

According to the traditional 3-DOF linear model of the vehicle and the idea of Parametric modelling, we build a vehicle rollover model shown in Fig.1.The main movement of the vehicle are the sideways movement along the y-axis, the yaw motion around the z-axis and the roll motion around the x-axis.

To simplify the model, we have the following assumptions for the mathematical model [10]:

Ignoring the impact of the steering system and side wind and the pitching motion around the y-axis of vehicle body;

1) Assuming that dynamic performance of tires are axial symmetric and tires on both sides have the same dynamic performance;

2) Ignoring the impact of unsprung mass;

3) Ignoring the impact on tire cornering characteristics of ground tangential force and the lateral force and the effect of tire aligning moment;

4) Assuming that vehicle speed along x-axis is unchanged and the transverse velocity, yaw angular velocity and angle velocity are much smaller than the speed.

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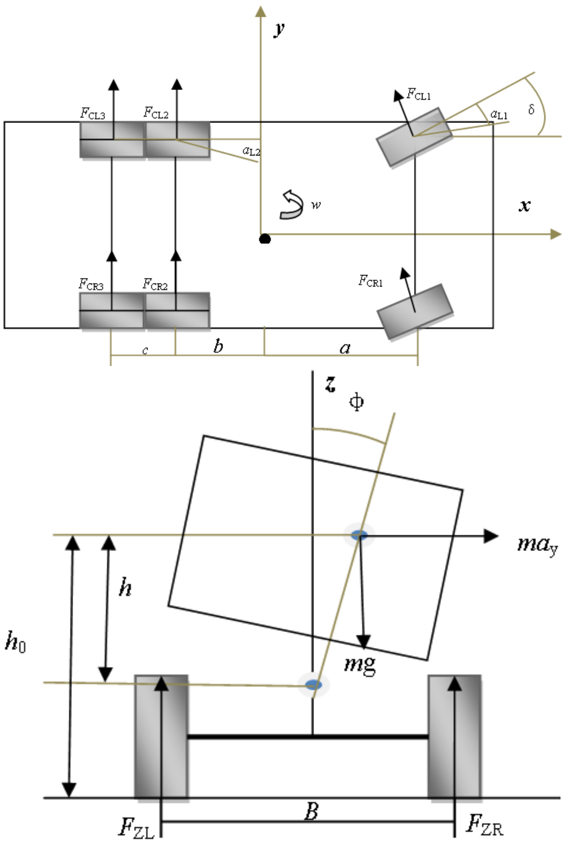

FIGURE 1 3-DOF rollover model

According to assumption above, considering the coupling effect of three directions, combined with the principle of Darren Bell, we have an analysis for the balance of the lateral movement of the force, yaw: Torque moment and movement of the roll motion and we can get the following formula:

The lateral force:

$$2(F_{CL1}\cos\delta+F_{CL3}+F_{CL2})=ma_y-mh\ddot{\phi}. \quad (1)$$

Yaw Torque:

$$2aF_{CL1}\cos\delta-2bF_{CL2}-2(b+c)F_{CL3})=I_z\dot{w}. \quad (2)$$

The sprung mass roll torque:

$$ma_yh+mgh\phi-(k_\phi\phi+c_\phi\dot{\phi})h=I_x\ddot{\phi}. \quad (3)$$

The lateral acceleration:

$$a_y=\dot{v}+wu. \quad (4)$$

The turning radius and yaw rate:

$$R=\frac{a+b+0.5c}{\delta}, \quad (5)$$

$$w=\frac{\sqrt{u^2+v^2}\bullet\delta}{a+b+0.5c}. \quad (6)$$

The symbols in Figure 1 and type (1) - (6) are in the meaning of follows:

m - sprung mass;
v - lateral velocity of vehicle;
u - longitudinal velocity of vehicle;
FCL1, FCL2, FCL3 - tThe cornering force of left wheel;
FCR1, FCR2, FCR3 - the cornering force of right wheel;
FZR, FZL - the vertical force of the wheels;
Δ - The steering angle of the front wheel;
φ - side angle of the vehicle body;
μ - tire-road friction coefficient; tire-road friction coefficienttire-road friction coefficient;
h - distance from centroid to rollover centre;
a, b - the distance between the front axle and the I rear and centroid;
B - tread;
c - the distance between the I rear axle and the II rear;
h0 - the equivalent static height of centroid;
Δh - vertical deformation of Suspension;
w - the yaw rate of vehicle;
aL1, aL2, aL3 - t-he wheel side slip angle;
ay - the lateral acceleration;
k1, k2, k3 - the front and rear tire cornering stiffness;
c1, c2, c3 - The front and rear axle roll steering equivalent coefficient;
kφ - roll stiffness coefficient of the suspension system;
cφ - roll damping coefficient of the suspension system;
Ix - roll inertia of vehicle body;
Iz - sway moment inertia of v-ehicle body;
kst—vertical stiffness of Suspension.

2.2 TIRE LATERAL FORCE

The lateral force subjected of each tire of heavy vehicle:

$$F_{CLi}=-k_ia_{li},\ F_{CRi}=-k_ia_{Ri}\ \text{(i=1,2,3)}. \quad (7)$$

Among,

$$a_{L1}=\delta-\tan^{-1}(\frac{v+wa}{u+0.5wB}),\ a_{R1}=\delta-\tan^{-1}(\frac{v+wa}{u-0.5wB})$$

$$a_{L2}=-\tan^{-1}(\frac{v-wb}{u+0.5wB}),\ a_{R2}=-\tan^{-1}(\frac{v-wb}{u-0.5wB})$$

$$a_{L3}=-\tan^{-1}(\frac{v-w(b+c)}{u+0.5wB}),\ a_{R3}=-\tan^{-1}(\frac{v-w(b+c)}{u-0.5wB})$$

2.3 HEAVY VEHICLE ROLLOVER MODEL

Simultaneous equations (1) - (7), we get the linear ordinary differential equations of heavy vehicle:

$$-mh\frac{d^2\phi}{dt^2}+m\frac{dv}{dt}-2\frac{k_1\cos\delta+2k_2}{u}\bullet v-2(\frac{k_1a\cos\delta-k_2(2b+c)}{u}-mu)w+2\phi(k_1c_1\cos\delta+2k_2c_1)+2k_1\delta\cos\delta=0$$

$$I_z\frac{dw}{dt}-2\frac{k_1a\cos\delta-k_2(2b+c)}{u}\bullet v+2[k_2(2b+c)c_2-k_1c_1a\cos\delta]\phi-2\frac{k_1a^2\cos\delta+k_2(2b+c)^2+k_2b^2}{u}w+2k_1a\delta\cos\delta. \quad (8)$$

$$I_x\frac{d^2\phi}{dt^2}+c_\phi\frac{d\phi}{dt}h+mh\frac{dv}{dt}-(mg-k_\phi)h\phi-mhwu=0$$

Type (8) describes the 3-DOF equations of heavy vehicles, the state parameters of the describing variables can be expressed as:

$$y(t) = \left\{ v, \dot{v}, w, \dot{w}, \phi, \dot{\phi}, \ddot{\phi} \right\}. \tag{9}$$

The input parameters are represented as:

$$x(t) = \left\{ u, \delta, m, h, h_0 \right\}. \tag{10}$$

Describe the state of constant can be expressed as:

$$\Gamma = \left\{ a, b, c, k_1, k_2, c_1, c_2, k_\phi, c_\phi \right\}. \tag{11}$$

3 Lateral load transfer ratio (LTR)

When heavy-duty truck driving in curve, vertical load transfer occurs on tires, the vertical force of two sides' tires is not equal. We can get the torque balance equation by the inner wheel in the plane z-y:

$$\begin{aligned} mg &= F_{ZL} + F_{ZR} \\ F_{ZL}B + mh_0 a_y + mgh\sin\phi - \frac{1}{2}mgB &= I_x\ddot{\phi} \end{aligned}. \tag{12}$$

The lateral load transfer ratio, defined as ratio of the tire vertical load of the left and right side [11]:

$$LTR = \frac{\left|F_{ZL} - F_{ZR}\right|}{F_{ZL} + F_{ZR}} = \left| 2\frac{I_x\ddot{\phi} - mh_0 a_y - mgh\sin\phi}{mgB} \right|. \tag{13}$$

When FZL=0, LTR=1, heavy vehicle reaches a critical rollover state and rollover is imminent;; When FZL=FZR, LTR=0,heavy vehicle completely smooth running. Therefore, the value range of LTR is [0, 1].

4 Influence of the vehicle state parameters on the rollover stability

4.1 SIMULATION MODEL AND VEHICLE PARAMETERS

Build heavy vehicle rollover simulation model in Simulink, as shown in Figure 2.

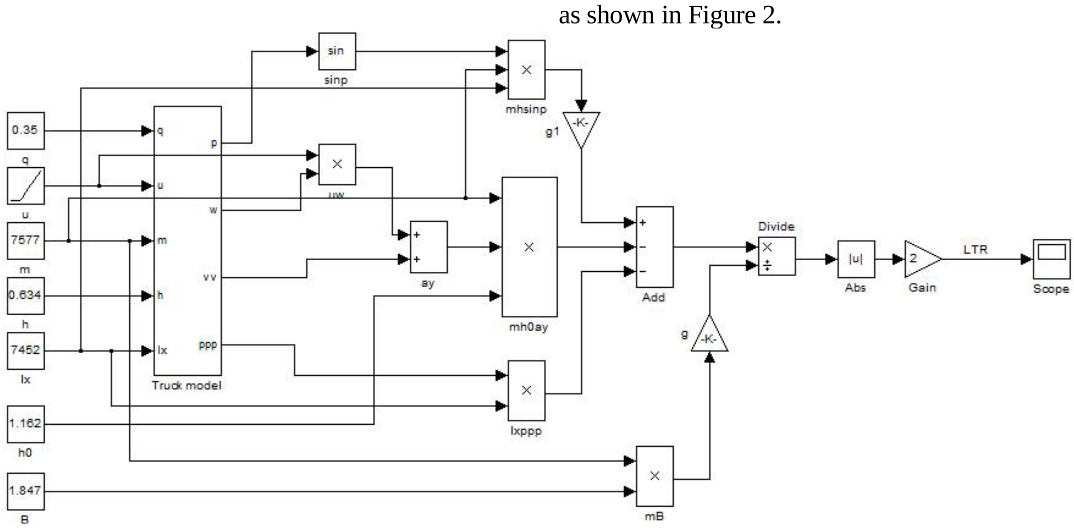

FIGURE 2 Simulation model

The input parameters for the model are u, δ, m, h, Ix, the output parameters are v, w, ϕ, $\ddot{\phi}$ .The main parameters used in the simulation model of the vehicle are: the sprung mass m is 7577kg, the Full load is 16077kg, and the initial velocity v is 10m /s; the equivalent static height of centroid h0 is 634 mm; roll inertia of vehicle body Ix is 7452.9 Kg·m2, sway moment inertia of vehicle body Iz is 88106.1 Kg·m2; the tread B is 1847 mm, the distance between the front axle and centroid a is 1500 mm, the distance between the Ⅰ rear and centroid b is 4365 mm, the distance between the Ⅰ rear axle and the Ⅱ rear c is 1350 mm; the front tire cornering stiffness k1 is 92000 N/rad, the rear tire cornering stiffness k2 is 65500 N/rad, the front axle roll steering equivalent coefficient c1 is 0.09,The rear axle roll steering equivalent coefficient c2 is0.08, roll stiffness coefficient of the suspension system kφ is 300000 N/rad, roll damping coefficient of the suspension system cφ is 22800 N/rad2,acceleration of gravity g is 9.81 m/s2.

Keep the other parameters constant, change the speed, the front wheel angle and the centroid moment and analyze the influence on rollover stability by state parameters of heavy vehicle in difference conditions.

4.2 INFLUENCE OF SPEED OF VEHICLE ON ROLL ANGLE AND LTR

Figure 3 shows the relationship between speed and turning angle; fig.4 shows the effects of the effects of speed on LTR.

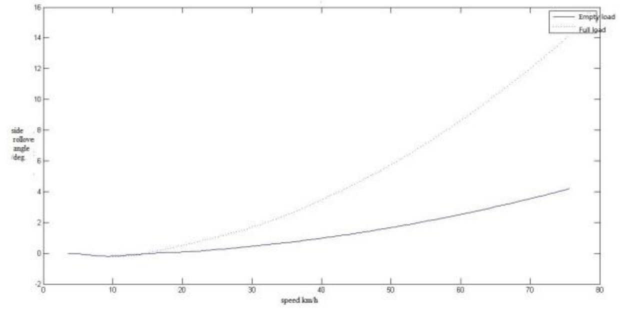

FIGURE 3 The relationship between speed and turning angle

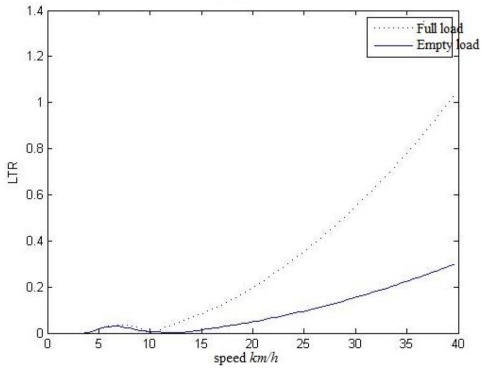

FIGURE 4 Shows the effects of the effects of speed on LTR

The fig.3- Fig.4 show that, with the increase of vehicle speed, angle and the value of LTR is obviously increased, the full load speed greatly increased. When no load, and v<70km/h, the vehicle condition is stable; under the full load condition, when the front wheel angle δ =20, and the speed at 40km/h, the value of LTR is obviously more than dynamic warning threshold value, then the vehicle rollover.

4.3 INFLUENCE OF THE STEERING ANGLE OF THE FRONT WHEEL ON ROLL ANGLE AND LTR

Figure 5 shows the relationship between front wheel angle and roll angle, fig.6 shows the influence of the front wheel angle on LTR.

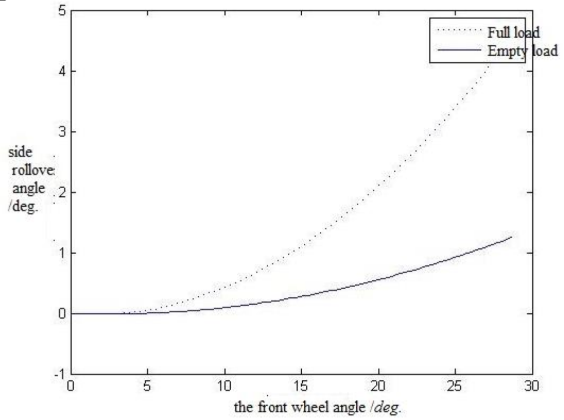

FIGURE 5 The relationship between front wheel angle and roll angle

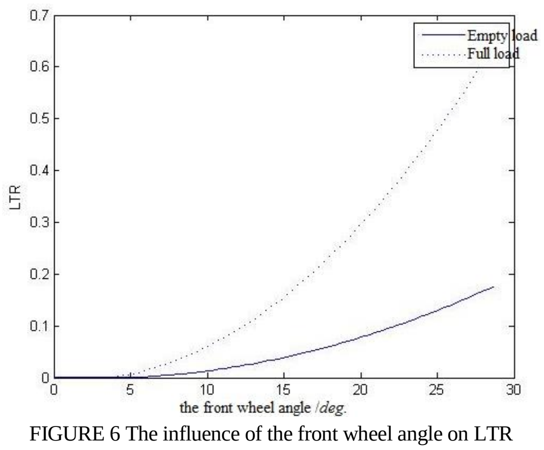

FIGURE 6 The influence of the front wheel angle on LTR

Fig.5- Fig.6 shows that, with the increase of the front wheel angle, roll angle and LTR increased. When the load is small, the front wheel angle has little effect on the roll angle and LTR; but in full load condition, the LTR increased sharply, rollover stability becomes poor, when δ=30。, the state parameters of vehicle approach the rollover safety threshold.

4.4 INFLUENCE OF THE DISTANCE FROM CENTROID TO ROLLOVER CENTRE ON ROLL ANGLE AND LTR

Figure 7 shows the relationship of the distance from centroid to rollover centre and side rollover angle; Fig.8 shows the impact of the distance from centroid to rollover centre on LTR.

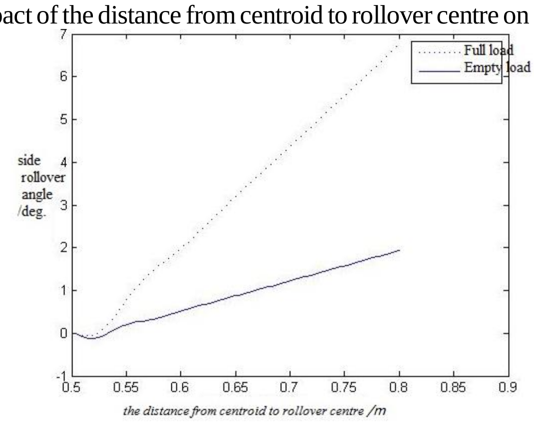

FIGURE 7 Relationship of the distance from centroid to rollover centre and side rollover angle

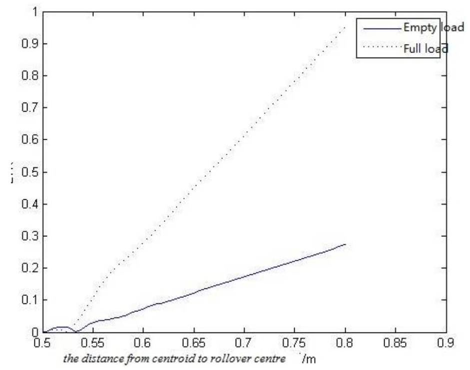

FIGURE 8 The impact of the distance from centroid to rollover centre on LTR

Figure 7- Fig.8 shows that, with the increase of h, roll angle and LTR increased. When the change of h is small, the impact of the distance from centroid to rollover centres on LTR has little effect on the roll angle and LTR; but in full load condition, the LTR increased sharply, rollover stability becomes poor, when h=770mm, the state parameters of vehicle approach the rollover safety threshold.

5 Conclusions

1) We established a differential equation of 3-DOF heavy vehicle rollover model, the model take into account effects of the load, position of centre of mass, moment inertia and cornering stiffness on rollover stability.

2) Simulation results show that: load have the maximum impact on rollover stability, speed and front wheel steering angle and other parameters will also affect the rollover stability, therefore, reduce the centroid height, increasing

tread and wheelbase can improve the running stability of heavy vehicles, and decrease lateral acceleration and lateral angle,then prevent vehicles rollover.

3) The traditional mathematical model and dynamic warning threshold value and cannot reflect the real-time rollover state and stability completely; And there are contradictions between accuracy and complexity.

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