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Analysis Rollover Condition of Tractor Semitrailer while Turning Maneuver with High Forward Speed

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Abstract

Nowadays, there are many tractor semitrailer vehicle accidents caused by lateral instabilities, which may be classified into two types: yaw instability and rollover. The rollover of tractor semitrailer frequently occurs while directional maneuvers at high forward speed. In this paper, a full dynamic model of tractor semitrailer is developed based on Multi-body System Method and Newton-Euler equations. The rollover condition is based on the load transfer ratio which corresponds to the load transfer between the left and the right sides of the vehicle. This model is applied to determine the rollover condition of the tractor semitrailer while turning maneuvers on the high forward speed.

Keywords: tractor semitrailer vehicle, rollover condition, high forward speed, load transfer ratio, roll safety factor.

1. Introduction

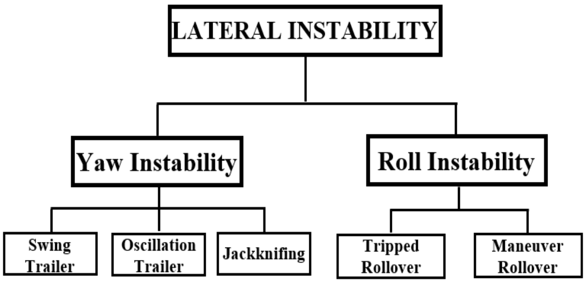

Fig. 1. Lateral Instability categorization

For the recent years, transportation by articulated vehicles has developed robustly to improve transportation productivity and reduce traffic jams, emission and environmental pollution. However, articulated vehicles often pose serious highway safety risks due to their excessive weights, larger dimensions, coupling between tractor and semitrailer vehicle...

Vehicle dynamic instability can be defined as an unexpected response maneuver induce disturbance, occurring in the ground plane: the longitudinal, lateral, vertical, pitch, yaw and roll direction, or combinations of those. For a tractor semitrailer, lateral instability can be classified into two types: yaw instability and roll instability (Fig.1). The yaw instability of the tractor semitrailer is defined as swing trailer, oscillation trailer and jackknifing. Jackknifing is characterized by rapid and uncontrol relative angular yaw motion between the tractor and the semitrailer [1]. The roll instability occurs when the centrifugal forces imposed on the vehicle during a maneuver exceed the rollover threshold of the vehicle. The rollover of vehicle constitutes two main categories: maneuver rollover and tripped rollover. The tripped rollover causes by colliding with another vehicle or any obstacle. The maneuver rollover occurs while lane change or turning maneuver on the high adhesion coefficient of roads with high forward speed. In this case, the roll angle is increased. The rollover condition of tractor semitrailer vehicle is determined when tires on axles lose road contact (wheel lift-off). Sampson [2] defined that the rollover threshold is the limit of steady state lateral acceleration that a vehicle can sustain without losing roll stability. The yaw instability, cause by either braking or combined braking and steering maneuvers on the low adhesion coefficient of roads.

This paper focuses on analysis rollover conditions of the tractor semitrailer when turning maneuvers on the high forward speed. A full dynamics model for tractor semitrailer vehicle is established with the multibody system analysis to determine the rollover conditions. Rollover conditions of the tractor semitrailer evaluation is based on the rollover indicators, namely the Load Transfer Ratio (LTR), Roll Safety Factor (RFS). The results of paper can be used to determine the Dynamic Rollover Threshold (DRT).

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2. Tractor Semitrailer Model

2.1. Coordinate systems

The framework of this study is focused on the tractor semitrailer vehicle which is composed of a tractor vehicle with 3 axles and a semitrailer vehicle with 3 axles. Sprung and unsprung masses are connected together by the suspension system (leaf springs for steering axles, walking beams for rear axles of the tractor vehicle, and six spring tandems for the semitrailer vehicle). The tractor and the semitrailer vehicle are connected at the fifth wheel hitch as shown in Figure 2.

To established the dynamics equation of this model, we consider the motion of the two sprung masses *k* (*k*=1: sprung mass of tractor vehicle; *k*=2: sprung mass of semitrailer vehicle); 6 axles *i* (*i*=1÷6) and wheels in the coordinate system (see Figure 2).

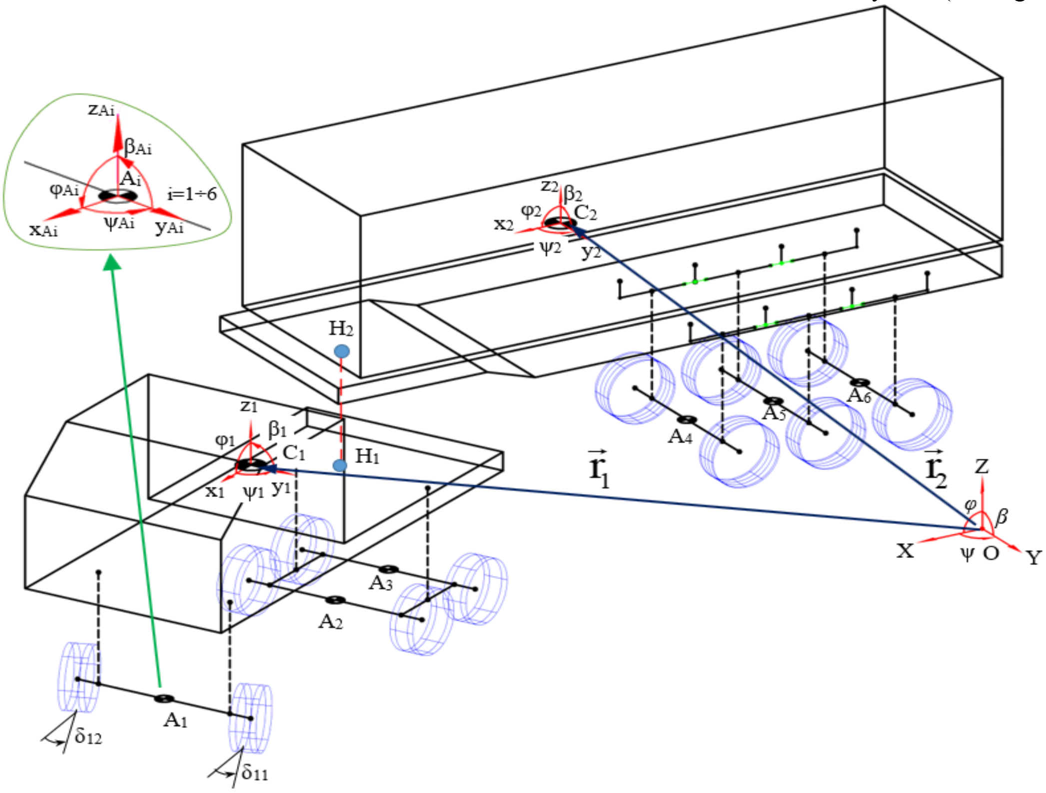

Fig. 2. Tractor Semitrailer Coordinate Systems

$\vec{r}_k = [X_k, Y_k, Z_k]^T$ is the vector which determines the location of the sprung mass k in the earth-fixed coordinate system. (OXYZ) is the earth-fixed coordinate system. ($C_k x_k y_k z_k$) are the sprung mass coordinate systems which are fixed at the center of gravity (CoG) of each body. ($A_i x_{Ai} y_{Ai} z_{Ai}$) are the axle coordinate systems defined at the center of each axle. The relative motion of sprung mass coordinate systems with respect to the earth-fixed coordinate system are described by the rotation matrices ${}^{O}_{Ck}R$. These rotation matrices are based on a set of body (X-Y-Z) rotations with β_k-φ_k-ψ_k angles [4] as follows:

$$ {}^{O}_{C_k}R = \begin{bmatrix} c\psi_k\, c\varphi_k & c\psi_k\, s\varphi_k\, s\beta_k - s\psi_k c\beta_k & c\psi_k\, s\varphi_k c\beta_k + s\psi_k s\beta_k \\ s\psi_k c\varphi_k & s\psi_k\, s\varphi_k\, s\beta_k + c\psi_k c\beta_k & s\psi_k\, s\varphi_k c\beta_k - c\psi_k s\beta_k \\ -s\varphi_k & c\varphi_k\, s\beta_k & c\varphi_k c\beta_k \end{bmatrix} \quad (1) $$

$c = \cos \qquad s = \sin$

2.2. Sprung Masses Model

In this paper, the motion of the two sprung masses in the earth-fixed coordinate system is considered (tractor and semitrailer). For each sprung mass, the mathematics model is formulated six equations of motion corresponding with the six degrees of freedom resulting from unconstrained motion. The six equations associated with the translational motion of the body are known as Newton Euler equations of motion:

$$ \begin{cases} m_k(\dot{v}_{xk} - v_{yk}\omega_{zk} + v_{zk}\omega_{yk}) = F_{xk} \\ m_k(\dot{v}_{yk} - v_{zk}\omega_{xk} + v_{xk}\omega_{zk}) = F_{yk} \\ m_k(\dot{v}_{zk} - v_{xk}\omega_{yk} + v_{yk}\omega_{xk}) = F_{zk} \\ I_{xk}\dot{\omega}_{xk} + (I_{zk} - I_{yk})\omega_{zk}\omega_{yk} = M_{xk} \\ I_{yk}\dot{\omega}_{yk} + (I_{xk} - I_{zk})\omega_{xk}\omega_{zk} = M_{yk} \\ I_{zk}\dot{\omega}_{zk} + (I_{yk} - I_{xk})\omega_{yk}\omega_{xk} = M_{zk} \end{cases} \quad (2) $$

Where v_{xk}, v_{yk}, v_{zk} are the translational velocities of sprung mass k; ω_{xk}, ω_{yk}, ω_{zk} are the rotational velocities of sprung mass k; m_k are the mass of the sprung mass k; I_{xk}, I_{yk}, I_{zk} are moments of inertia of the sprung mass k; F_{xk}, F_{yk}, F_{zk} are the total applied forces acting on the sprung mass k resolved parallel to $C_kx_ky_kz_k$; M_{xk}, M_{yk}, M_{zk} are the total applied moments acting on the sprung mass k resolved parallel to $C_kx_ky_kz_k$.

2.3. Unsprung Masses Model

Each of the axles is thus characterized as a rigid beam with 2 DOFs (vertical z_{Ai} and roll motion ω_{xAi}) (Fig.3). The Newton's and Euler's Equations of the axles in the axle coordinate systems are as follows:

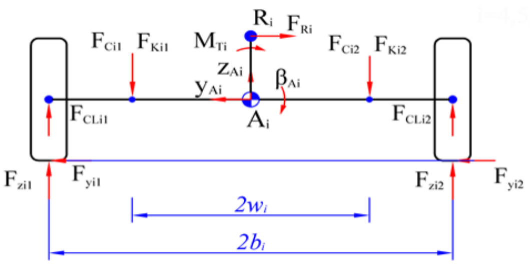

Fig. 3. Unsprung masses model

$$\begin{cases} m_{Ai}(\dot{v}_{zAi} - v_{xAi}\omega_{yAi} + v_{yAi}\omega_{xAi}) = F_{zAi} \\ I_{xAi}\dot{\omega}_{xAi} + (I_{zAi} - I_{yAi})\omega_{zAi}\omega_{yAi} = M_{xAi} \end{cases} \quad (3)$$

where m_{Ai} and I_{Axi}, I_{Ayi}, I_{Azi} are the mass and the moment of inertia of the axle i, respectively; F_{AZi}, M_{Axi} are the total applied forces and moments acting on the axle i resolved parallel to $A_ix_{Ai}y_{Ai}z_{Ai}$;

The total applied forces and moments acting on sprung mass k are calculated from the suspension systems with the spring and damper forces and auxiliary roll moments [5]; aerodynamic forces [4] and fifth wheel hitch forces and moments. The total applied forces and moments acting on the axle i are calculated from the suspension systems and tire-road interaction. The tire forces are longitudinal, lateral and vertical forces. These forces are present in the following.

2.4. Modeling of Tires

Vehicle motions are primarily caused by forces and moments developed at the tire-road interface. In this paper, assuming that the overturning moment and other moments are neglect. The longitudinal and lateral forces are computed based on Ammon Tire Model [6].

$$\begin{cases} F_{xij} = \dfrac{s_{ij}}{\sqrt{s_{ij}^2 + \alpha_{ij}^2}}\varphi_{x\max}F_{zij}(t)f\left(\dfrac{\sqrt{s_{xij}^2 + \alpha_{ij}^2}}{s_{x,\max}\varphi_{x,\max}}\right) \\ F_{yij} = \dfrac{\alpha_{ij}}{\sqrt{s_{ij}^2 + \alpha_{ij}^2}}\varphi_{y\max}F_{zij}(t)f\left(\dfrac{\sqrt{s_{x}^2 + \alpha_{ij}^2}}{\alpha_{\max}\varphi_{y,\max}}\right) \end{cases} \quad (4)$$

The inputs of the tire model are tire vertical loads F_{zij}, lateral slip angles α_{ij} and longitudinal slip ratios s_{ij}...

2.5. Modeling of Fifth Wheel Hitch

In this paper, the fifth wheel hitch is assumed to be relatively rigid in translation at H_1 and H_2 as shown in Figure 2. The forces transmitted through the hitch are calculated from kinematic constraints, stating that the acceleration at the hitch point is same for both the sprung mass of the tractor and that of the semitrailer vehicle [5].

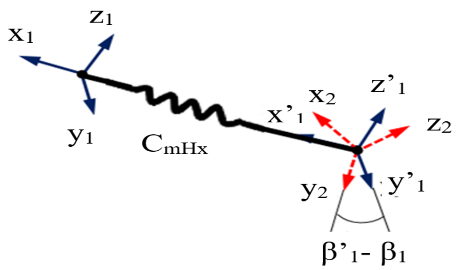

Fig. 4. Representation of the conventional fifth wheel hitch connection.

The fifth wheel hitch allows relative motions by yaw and pitch angles when the friction is skipped at the hitch surface. The fifth wheel is represented by the roll stiffness coefficient C_{mHx} at the hitch [7] as shown in Figure 3. The roll moment M_{Hx1}, M_{Hx2} acting on fifth wheel can be expressed as:

$$\begin{cases} M_{Hx1} = C_{mHx}(\beta'_1 - \beta_1) \\ M_{Hx2} = -\cos(\psi_2 - \psi_1)M_{Hx1} \end{cases} \quad (5)$$

where C_{mHx} is roll angle stiffness of the fifth wheel hitch β'_1 is calculated as:

$$\beta'_1 = \arctan\frac{\sin\beta_2\cos(\psi_2 - \psi_1) - \varphi_2\cos\beta_2\sin(\psi_2 - \psi_1)}{\varphi_1\sin(\psi_2 - \psi_1)\sin\beta_2 + \cos\beta_2} \quad (6)$$

2.6. Rollover Risk Indicator

The rollover risk evaluation is based on the load transfer ratio. LTR corresponds to the load transfer between the left and right sides of all tires. RSF is the load transfer ratio between the left and the right sides of all tires without the tires of the 1st axle [7]. The formula for the 6-axle tractor semitrailer vehicle is as follows:

$$LTR=\frac{\left|\sum_{i=1}^{6}(F_{zi2}-F_{zi1})\right|}{\sum_{i=1}^{6}(F_{zi2}+F_{zi1})} \tag{7}$$

$$RSF=\frac{\left|\sum_{i=2}^{6}(F_{zi2}-F_{zi1})\right|}{\sum_{i=2}^{6}(F_{zi2}+F_{zi1})} \tag{8}$$

where the vertical tire force F_{zij} (i=1÷6; j=1: left wheels, j=2: right wheels) at each wheel is calculated from the vertical deflection of tire.

The tractor semitrailer vehicle model is simulated with the software Matlab-Simulink and structural parameters of the 6-axle tractor semitrailer vehicle which compose of the tractor HOWO A7-375 tractor vehicle and CIMC 40FT semitrailer vehicle [1]. All parameters of the tractor semitrailer vehicle are defined in Table 1.

Table 1. Simulation parameters of a 6-axle tractor semitrailer vehicle.

Parameter	Symbol (Unit)	Value
Sprung mass of the tractor	m_1(kg)	7620
Sprung mass of the semitrailer	m_2(kg)	34715
Unsprung masses of the axles	$m_{A1}/m_{A2,3}/m_{A4,5,6}$(kg)	640;1150;780
Wheel base of the tractor	L_1+c(m)	3.24+1.34
Wheel base of the semitrailer	L_2+d+d(m)	6.945+1.31+1.31
Half track width of the axles	b_1; $b_{2,3}$; $b_{4,5,6}$ (m)	1.025; 0.93; 0.925
Half spring spacing of the axles	w_1; $w_{2,3}$; $w_{4,5,6}$ (m)	0.6; 0.5; 0.5
Height of the fifth wheel hitch	h_H (m)	1.33
Height of tractor's CoG	h_1 (m)	1.2
Height of semitrailer's CoG	h_2 (m)	2.2
Roll moment of inertia of tractor's sprung mass	I_{x1}(kgm^2)	11494.3
Roll moment of inertia of semitrailer's sprung mass	I_{x2}(kgm^2)	52828.7
Pitch moment of inertia of tractor's sprung mass	I_{y1}(kgm^2)	38399.2
Pitch moment of inertia of semitrailer's sprung mass	I_{y2}(kgm^2)	484022.2
Yaw moment of inertia of tractor's sprung mass	I_{z1}(kgm^2)	34969.9
Yaw moment of inertia of semitrailer's sprung mass	I_{z2}(kgm^2)	467066.4
Suspension stiffness of the axles	C_{1j}, C_{23j}, $C_{4,5,6j}$(kN/m)	250; 1400; 2500
Suspension damping ratio of the axles	K_{1j}, $K_{2,3j}$, $K_{4,5,6j}$(kNs/m)	15; 30; 30
Tire vertical stiffness of single tire	C_L (kN/m)	980
Fifth wheel roll stiffness	C_{mHx} (kNm/rad)	6000
Maximum friction coefficient	$\varphi_{max,0}$	0.8

3. Results and Discussions

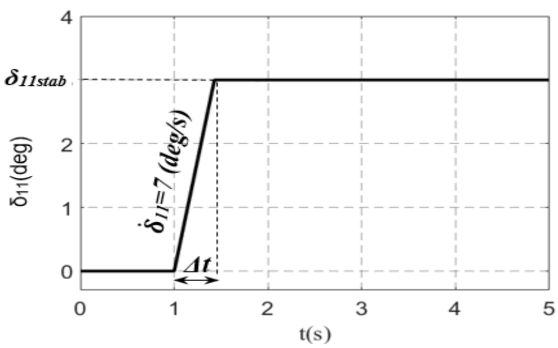

Fig. 5. Left road wheel steering angle.

The turning maneuver in an open-loop mode are often characterized by a Ramp Steer Maneuver [8] with some amplitude of steering angle δ_{11stab} at 70km/h of forward speed (Fig.5).

For the δ_{11stab} of 2;2,5;3(deg), the tractor semitrailer vehicle is in stable condition. For the δ_{11stab} of 3,5;4;4,5(deg), the tractor semitrailer vehicle suffers from rollover condition. These are shown by the increase rapidly in the roll angle of semitrailer β_2 (see Fig.6).

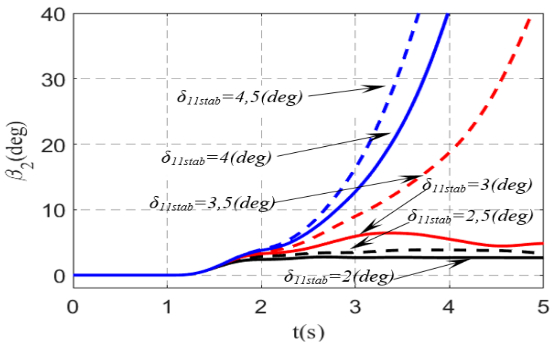

Fig. 6. Roll angle of the sprung mass of the semitrailer vehicle.

In the stable conditions, the signal rollover (LTR, RSF) is not reach to 1 (see Fig.6) when the road wheel steer angle is kept in δ_{11stab} (see Fig.7). Fig.8 illustrates the roll performance signature of a tractor semitrailer vehicle in δ_{11stab}=4(deg). In this condition, the vehicle is suffered from rollover, the roll angle of semitrailer is increases rapidly and the lateral acceleration a_{y1}, a_{y2} are decreased after reaches the peak value. When the left tire of 2nd axle of tractor is lost contact from ground, RSF equals 1. Later, the LTR reaches to 1 when all the left tires of vehicle are lost contact from ground. There are signals of rollover condition. For example, when the velocity is 70 (km/h) and the magnitude of road wheel steering angle is 4 deg, LTR is equal to 1 at 13.012 (deg) of β_2, a_{y1max} is 4.996 (m/s^2) at 4.425 (deg) of β_2 and a_{y2max} is 4.52 (m/s^2) at 7.659 (deg) of β_2 (see Fig. 8).

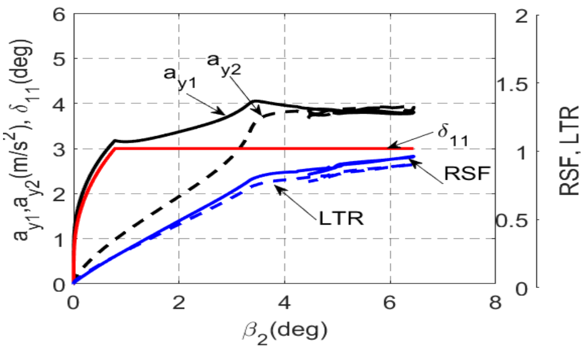

Fig. 7. Roll performance signature of a tractor semitrailer vehicle in δ_{11stab}=3deg.

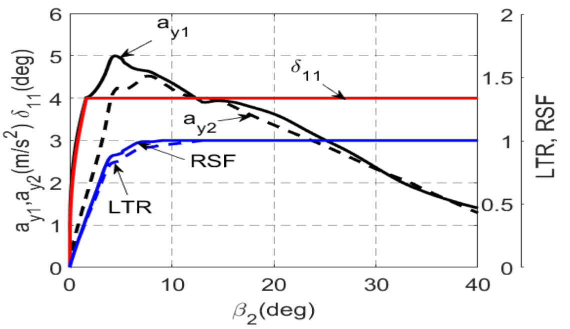

Fig. 8. Roll performance signature of a tractor semitrailer vehicle in δ_{11stab}=4deg.

Graphed the peaks of a_{y1}, a_{y2}, LTR, RSF with the δ_{11stab} is from 0,5 to 12 (deg) in the Fig.9. This figure illustrates the effect of road wheel steering angle on rollover condition of tractor semitrailer vehicle. The DRT of semitrailer is equal 4,12 (m/s^2) when the reaching to 1 of LTR at δ_{11stab}=3,5 (deg) and 70km/h of forward speed. With the method of survey, corresponding to each velocity during turning maneuver at the left road wheel steering angle input δ_{11}, the max values of a_{y1} and a_{y2} will be obtained. It is possible to detect DRT of tractor and semitrailer according to many different parameters.

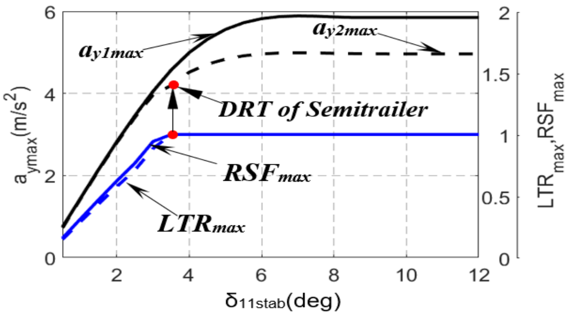

Fig. 9. Effect of road wheel steering angle on rollover condition of tractor semitrailer.

4. Conclusion

In this paper, the rollover condition of a tractor semitrailer vehicle is examined. This paper presents the full dynamics model for the tractor semitrailer vehicle which is developed on the basis of the multibody system analysis with 6 DOF for each sprung mass. The model includes the details of vehicle dynamics as well as fifth wheel model, tire model, etc. The evaluation results are shown in the stable and rollover condition. The model is applied to detect the rollover conditions of the tractor semitrailer and the DRT of semitrailer while turning maneuver on the high forward speed.

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