

The effect of velocity on the ability to rollover of the tractor semi-trailer when turning

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ABSTRACT

This paper aims to investigate the impact of vehicle velocity on the rollover stability of a fully loaded tractor semi-trailer during lane changes and turning. Specifically, the study focuses on velocities ranging from 30 km/h to 60 km/h. The investigation found that vehicle velocity is a critical parameter that can affect the potential for rollover during lane changes or turning. The results showed that to ensure the vehicle moves steadily and does not roll over during these maneuvers, the steering angle must be controlled and kept within certain limits. The study provided specific recommendations for maximum steering angles at different velocities. For instance, to ensure stability when the vehicle is moving at 60 km/h, the maximum steering angle should be less than 4 degrees. Similarly, at 50 km/h, the maximum steering angle is recommended to be less than 6 degrees, and at 44 km/h, it should be less than 8 degrees. At lower speeds, the recommended maximum steering angle increases, with the maximum recommended angle at 36 km/h being 12 degrees. These findings highlight the importance of carefully controlling vehicle velocity and steering angle to minimize the risk of rollover accidents. By providing specific recommendations for different velocities, this study can inform the design and safety testing of vehicles to improve their stability and safety during lane changes and turning.

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1. Introduction

Tractor semi-trailers are an integral part of the global transportation industry, with millions of these vehicles transporting goods across vast distances each year. However, rollover accidents involving tractor semi-trailers can have catastrophic consequences, leading to loss of life, property damage, and significant economic costs (Sampson 2000). Research on tractor semi-trailer rollover accidents has identified several factors that can contribute to these incidents, including vehicle design, cargo weight distribution, driver behavior, and environmental conditions. Vehicle speed and maneuvering are also critical factors in rollover accidents. Previous studies have shown that high vehicle speeds, sudden lane changes, and sharp turns increase the likelihood of rollover accidents (Abdulwahab, 2018; McKnight & Bahouth, 2009).

Several researchers have used computer simulations to investigate the stability of tractor semi-trailers during different maneuvers, providing insights into the relationship between vehicle speed, steering angle, and rollover risk. For example, Lewis & El-Gindy (2003) studied the effect of lateral acceleration on the rollover stability of a semi-trailer and found that increased lateral acceleration increased the rollover risk. Meanwhile, García et al. (2003) analyzed the effects of different load distributions on the stability of tractor semi-trailers during turning. Goldman and coworkers (2001), surveyed the rollover dynamics of vehicles passing from the road. In other similar work, Liu et al. (2001) performed the same study for articulated freight vehicles. Some researchers such as (Tan et al. 2021; Zheng et al. 2023) investigated the stability and control system of heavy vehicles including tractor semi-trailer. In the aforementioned works, the dynamic models and simulation results were validated using track test data for different maneuvers such as ramp-steer and sine-with-dwell. Wang et al. analyzed and tested

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experimentally the roll stability of liquid tank semi-trailer based on the wheel force when the vehicle turns or experiences the emergency conditions. The effectiveness of the roll stability control system for commercial double-truck long vehicles was evaluated experimentally by Van Kat (2022). He obtained the ranges of speed-induced rollovers by sensing the lateral acceleration and roll angle. Zhang et al. (2020) also performed numerical research to study the stability of tractor semi-trailer under cross and lateral wind. They simulated the aerodynamics of the heavy vehicle using CFD (computational fluid dynamics) method for different crosswinds. They also explained the stability conditions in terms of the lateral acceleration/displacements and yaw rate. Zhou et al. (2020) also surveyed the lateral stability of articulated heavy vehicles (AHV) with central axle trailer. They considered and modeled three degrees of freedom (3-DOF) yaw-plane AHV and obtained its dynamic behavior under evasive maneuver. According to their findings, trailer axle positions related to its center of gravity had the most important role on the stability of AHV. Chen et al. (2019) presented a comparative study on lateral stability of vehicles with 28-ft and 33-ft doubles and 28-ft triple trailers. According to their findings, the trucks with triple trailers showed greater rearward amplification, larger likelihood of rollovers, and greater off-tracking compared to the double trailers vehicles. By considering the liquid sloshing effects, Sun et al. (2021) studied the rollover control of a semi-trailer tanker. Sahin & Akalin (2020) also worked on active rear-wheel steering control of heavy tractors using fuzzy logic models. Liu and coworkers (2021) modeled the path tracking control of large vehicles such as tractor semi-trailer. Tung (2017), Hung (2017) and Kiet (2018) also investigated the stability of semi-trailer tractors. Despite the considerable research on tractor semi-trailer rollover accidents and dynamics of such vehicles, few studies have investigated the effect of vehicle velocity on the rollover stability of these vehicles during turns and lane changes. This study aims to fill this gap by examining the relationship between vehicle velocity, steering angle, and the potential for rollover accidents during turns and lane changes.

Research on tractor semi-trailer rollover accidents has identified several factors that contribute to these incidents, including vehicle design, cargo weight distribution, driver behavior, and environmental conditions. Previous studies have used computer simulations and mathematical models to investigate the stability of these vehicles during different maneuvers. However, few studies have examined the effect of vehicle velocity on rollover stability during turns and lane changes. This study aims to address this gap in the literature and provide specific recommendations for safe steering angles at different velocities to improve the stability and safety of tractor semi-trailers during these maneuvers.

2. Mathematical model

The tractor semi-trailer is a large and complex vehicle that consists of two bodies connected by a turntable. The semi-trailer's bulky and heavy structure is supported by the turntable and the tractor. The tractor has three axles, with the front axle using a dependent suspension and fitted with single wheels, while the two rear axles use a balanced suspension and fitted with dual wheels. The semi-trailer has three rear axles that use a continuously balanced suspension system and are fitted with dual wheels. To model the tractor semi-trailer, the method of splitting the structure of the many-body system is used. Accordingly, the survey model of this problem is shown in Fig. 1.

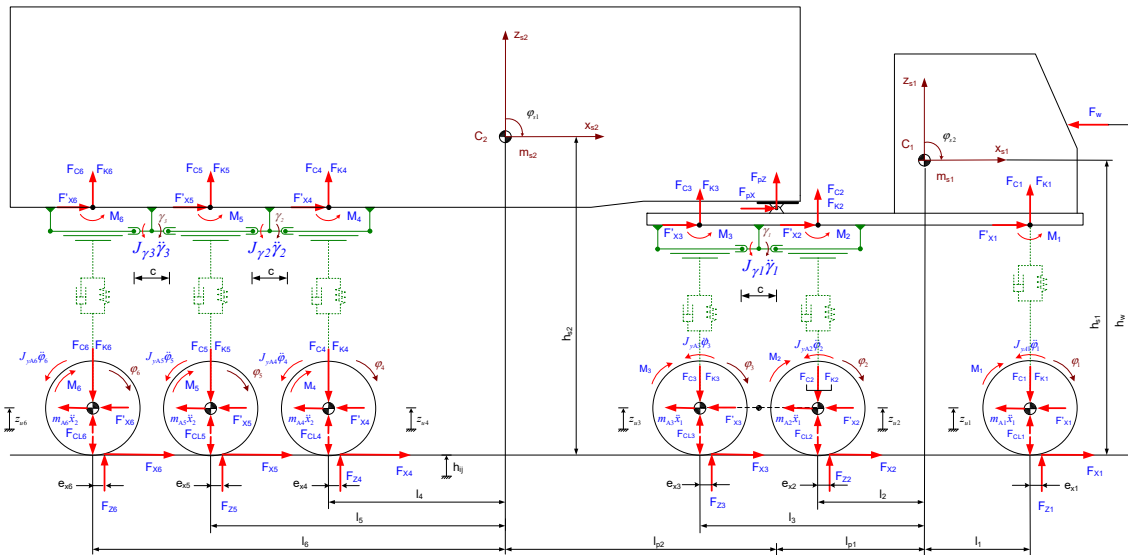


Fig.1. Mathematical model of the vehicle

To describe the non-linear motion of the sprung mass and unsprung mass of a 6-axis vehicle, Newton's method is applied using differential equations in the x, y, and z directions. These equations are developed through research and can provide valuable insights into the behavior and performance of the tractor semi-trailer. By analyzing these equations, researchers can better understand the dynamic response of the vehicle under different conditions, which can help to optimize its design and improve its performance. The use of Newton's method in this context is an example of how mathematical tools can be applied

to solve real-world problems and advance our understanding of complex systems (Tung et al. 2020; Tung & Van Huong 2021a,b).

$$(m_{s1} + m_{u1} + m_{u2} + m_{u3})\ddot{x}_{s1} = F_{x1r}\cos\phi_{1r} + F_{x1l}\cos\phi_{1l} - F_{y1r}\sin\phi_{1r} - F_{y1l}\sin\phi_{1l} + F_{x2l} + F_{x2r} + F_{x3l} + F_{x3r} - F_w - F_{px1} \quad (1)$$

$$(m_{s2} + m_{u4} + m_{u5} + m_{u6})\ddot{x}_{s2} = F_{xj1} + F_{xjr} + F_{px2} \quad (j = 4 : 1 : 6) \quad (2)$$

$$(m_{s1} + m_{u1} + m_{u2} + m_{u3})\ddot{y}_{s1} = F_{x1r}\sin\phi_{1r} + F_{x1l}\sin\phi_{1l} + F_{y1r}\cos\phi_{1r} + F_{y1l}\cos\phi_{1l} + F_{y2r} + F_{y3r} + F_{y2l} + F_{y3l} - F_{py1} \quad (3)$$

$$(m_{s2} + m_{u4} + m_{u5} + m_{u6})\ddot{y}_{s2} = F_{yjr} + F_{yjl} + F_{py2} \quad (j = 4 : 1 : 6) \quad (4)$$

$$m_{s1}\ddot{z}_{s1} = F_{Cjr} + F_{Kjr} + F_{Cjl} + F_{Kjl} - F_{pz1} \quad (j = 1 : 1 : 3) \quad (5)$$

$$m_{s2}\ddot{z}_{s2} = F_{Cjr} + F_{Kjr} + F_{Cjl} + F_{Kjl} - F_{pz2} \quad (j = 4 : 1 : 6) \quad (6)$$

$$m_{uj}\ddot{z}_{uj} = F_{CLjl} + F_{CLjr} - F_{Cjl} - F_{Kjl} - F_{Cjr} - F_{Kjr} \quad (j = 1 : 1 : 6) \quad (7)$$

$$m_{ij}\ddot{y}_{ij} = F_j + F_{yjl} + F_{yjr} \quad (j = 1 : 1 : 6) \quad (8)$$

Euler's method is applied to build differential equations describing the rotation of the sprung mass and unsprung mass of a 6-axis vehicle around three axes x, y, z as follows (Tung 2021; Thanh Tung 2021; Tung & Van Huong 2021):

$$J_{ss1}\ddot{\beta}_{s1} = (F_{Cjl} + F_{Kjl} - F_{Cjr} - F_{Kjr})b_j + F_j(h_{s1} - h_{Bj}) - M_{px1} \quad (j = 1 : 1 : 3) \quad (9)$$

$$J_{ss2}\ddot{\beta}_{s2} = (F_{Cjl} + F_{Kjl} - F_{Cjr} - F_{Kjr})b_j + F_j(h_{s2} - h_{Bj}) + M_{px2} \quad (j = 4 : 1 : 6) \quad (10)$$

$$J_{ys1}\ddot{\phi}_{s1} = F'_{xj}(h_{s1} - r_j) - (F_{Cjl} + F_{Kjl} + F_{Cjr} + F_{Kjr})l_j + F_{px1}(h_{s1} - h_{p1}) + F_{pz1}l_{p1} + M_{jl} + M_{jr} \quad (j = 1 : 1 : 3) \quad (11)$$

$$J_{ys2}\ddot{\phi}_{s2} = F'_{xj}(h_{s2} - r_j) - (F_{Cjl} + F_{Kjl} + F_{Cjr} + F_{Kjr})l_j + F_{px2}(h_{s2} - h_{p2}) + F_{pz2}l_{p2} + M_{jl} + M_{jr} \quad (j = 4 : 1 : 6) \quad (12)$$

$$J_{zs1}\ddot{\psi}_{s1} = (F_{x1l}\sin\phi_{1l} + F_{y1l}\cos\phi_{1l} + F_{x1r}\sin\phi_{1r} + F_{y1r}\cos\phi_{1r})l_1 + (F_{x1l}\cos\phi_{1l} - F_{x1r}\cos\phi_{1r})a_1 + F_{py1}l_{p1} \quad (13)$$

$$+ (F_{y1r}\sin\phi_{1r} - F_{y1l}\sin\phi_{1l})a_1 - (F_{y2l} + F_{y2r})l_2 - (F_{y3l} + F_{y3r})l_3 + (F_{x2l} - F_{x2r})a_2 + (F_{x3l} - F_{x3r})a_3 \quad (14)$$

$$J_{zs2}\ddot{\psi}_{s2} = (F_{xj1} - F_{xjr})a_j + F_{py1}l_{p1} - (F_{yjl} + F_{yjr})l_j \quad (j = 4 : 1 : 6) \quad (14)$$

$$J_{xuj}\ddot{\beta}_{uj} = (F_{Cjr} + F_{Kjr} - F_{Cjl} - F_{Kjl})b_j + (F_{CLjl} - F_{CLjr})b_j - F_{yj}(r_j + z_{uj}) + F_j(h_{Bj} - r_j) \quad (j = 1 : 1 : 6) \quad (15)$$

$$J_{wyj}\ddot{\phi}_{wj} = M_{Aj} - M_{Bj} - F_{xj}r_{dj} \quad (j = 1 : 1 : 6) \quad (16)$$

The damping coefficient K is a parameter that relates to the dissipation of energy and resistance to motion in a mechanical system. It describes the relationship between the displacement and relative displacement velocity between the sprung mass and the unsprung mass. In other words, it represents the amount of force that opposes the relative motion between these two masses. In practical terms, the damping coefficient is an essential factor in the design of mechanical systems, such as vehicle suspensions. It helps to control the response of the system to external forces and influences, such as road irregularities or changes in the speed. A higher damping coefficient will result in a greater resistance to motion, which can improve stability and handling but may also lead to a harsher ride. Conversely, a lower damping coefficient may provide a smoother ride but could compromise stability and control. In order to determine the optimal damping coefficient for a given system, engineers and researchers use mathematical models and simulations that take into account various factors such as vehicle weight, speed, and road conditions. By analyzing the relationship between displacement and relative displacement velocity, they can refine the design and make adjustments to the damping coefficient as needed. This iterative process can help to improve the performance and safety of mechanical systems, and can highlight the importance of understanding the relationship between these physical quantities (Tung & Van Van, 2022; Tung & Van, 2022a,b).

$$F_{Kj} = K_j (\dot{z}_{uj} - \dot{z}_{sj}) \quad (17)$$

$$F_{Cj} = \begin{cases} C_\infty (z_{uj} - z_{sj} + f_{dj}^n) & \text{if } f_{dj}^n < z_{uj} - z_{sj} \\ C_j (z_{uj} - z_{sj}) & \text{if } f_{dj}^t \leq z_{uj} - z_{sj} \leq f_{dj}^n \\ -C_\infty (z_{uj} - z_{sj} - f_{dj}^t) & \text{if } z_{uj} - z_{sj} < f_{dj}^t \end{cases} \quad (18)$$

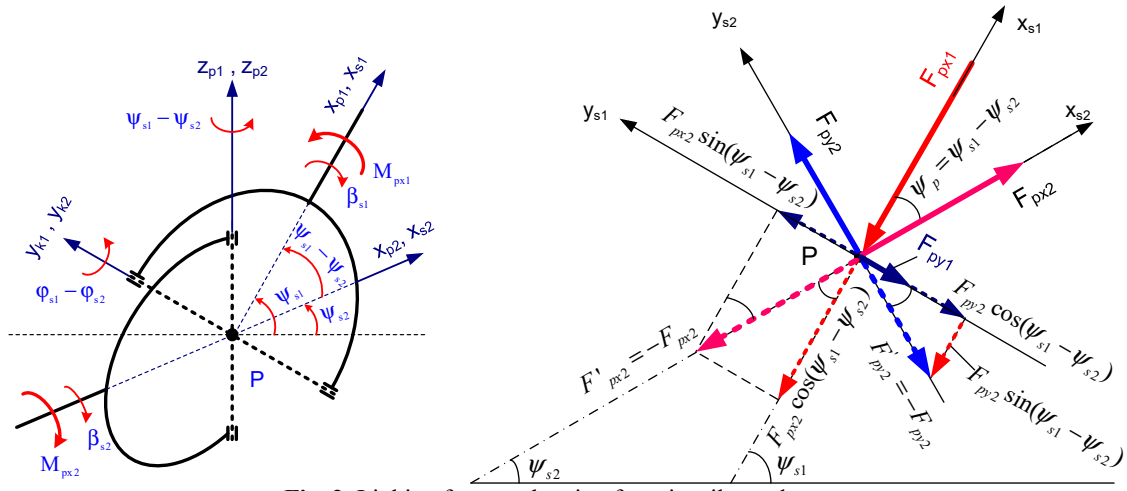


Fig. 2. Linking force at the pin of semi-trailer and tractor

The forces and moments at the link pin acting on the tractor and the semi-trailer (as shown and defined in Fig. 2) are calculated as follows (Van Van et al., 2022; Rajamani & Rajamani, 2012):

$$\begin{cases} F_{px1} = F_{px2} \cos(\psi_{s1} - \psi_{s2}) + F_{py2} \sin(\psi_{s1} - \psi_{s2}) \\ F_{py1} = F_{py2} \cos(\psi_{s1} - \psi_{s2}) - F_{px2} \sin(\psi_{s1} - \psi_{s2}) \\ M_{px1} = M_{px2} \frac{\cos(\psi_{s2} - \psi_{s1})}{1 - \sin^2(\psi_{s2} - \psi_{s1}) \cos^2(\varphi_{s2} - \varphi_{s1})} \end{cases} \quad (19)$$

$$\begin{cases} F_{px2} = F_{px1} \cos(\psi_{s1} - \psi_{s2}) - F_{py1} \sin(\psi_{s1} - \psi_{s2}) \\ F_{py2} = F_{px1} \sin(\psi_{s1} - \psi_{s2}) + F_{py1} \cos(\psi_{s1} - \psi_{s2}) \\ M_{px2} = M_{px1} \frac{\cos(\psi_{s1} - \psi_{s2})}{1 - \sin^2(\psi_{s1} - \psi_{s2}) \cos^2(\varphi_{s1} - \varphi_{s2})} \end{cases} \quad (20)$$

Table 1 illustrates the symbols and their definitions used in the aforementioned formulas.

Table 1. Symbols used for deriving the mathematical model and their explanations.

Symbols	Explain
$j = 1:1:6$	The order of the vehicle's axles from front to rear
x_s, y_s, z_s	Displacement of the sprung mass in the x, y, z directions
x_u, y_u, z_u	Displacement of the unsprung mass in the x, y, z directions
r_{dj}	Tyre dynamic radius
ϕ_l, ϕ_r	Left steer angle (LSA) and right steer angle (RSA)
l_j	Length from j-th axle to center of gravity
h_{s1}, h_{s2}	Center height of sprung and un-sprung mass
a_i, b_i	Distance between tire and suspension of left and right side
φ_j	The angle of rotation of j-th wheel
$\beta_s, \varphi_s, \psi_s$	The angle of rotation of vehicle about axes x, y, z
m_{s1}, m_{s2}, m_{uj}	Sprung and un-sprung mass of vehicle
$J_{xs}, J_{ys}, J_{zs}, J_{xu}, J_{yu}$	Moment of inertia of the vehicle about axes x, y, z
F_{Cj}, F_{Kj}, F_{CLj}	Elastic force and damping of suspension and tires
F_{xj}, F_{yj}, F_{zj}	Force acting on the wheel in the directions x, y, z
F_{px}, F_{py}, M_{pz}	Forces and moments at the link pin in the directions x, y, z
M_{Aj}, M_{Bj}	Active torque and braking

3. Rollover evaluation criteria

Our study focused on evaluating the criteria for vehicle rollover, which is a significant safety concern that can occur during lane changes, cornering, or turning, especially at high speeds. To determine the likelihood of instability and rollover, the authors utilized two evaluation criteria: the Load Distribution Coefficient (LDC) and the Safety Rollover Coefficient (SRC).

The Load Distribution Coefficient (LDC) measures the separation of the wheels from the road surface and helps to determine the likelihood of rollover. It is calculated using a formula (17) that takes into account the weight distribution of the vehicle and the position of the center of gravity. Additionally, the maximum Load Distribution Coefficient “(LDC)max” is calculated using formula (18), which provides information of the maximum load distribution on the vehicle.

The Safety Rollover Coefficient (SRC) is another evaluation criterion that can help to determine the possibility of vehicle instability and rollover. It takes into account the lateral acceleration and height of the center of gravity, which can affect the vehicle's stability during turning or cornering. By analyzing both the Load Distribution Coefficient and the Safety Rollover Coefficient, one can determine the potential risks of instability and rollover for the vehicle under different conditions.

This study highlights the importance of evaluating vehicle rollover and instability as part of the design and safety testing process. By utilizing criteria such as the Load Distribution Coefficient and the Safety Rollover Coefficient, designers and engineers can identify potential safety concerns and make adjustments to the vehicle design to mitigate these risks. Ultimately, this can lead to safer and more reliable vehicles on the road. The sign that the wheel is not in contact with the road surface occurs when $(LDC)_{max} = \pm 1$ (Chen et al., 2016; Jazar, 2017, 2019).

$$(LDC)_j = \frac{F_{jr} - F_{jl}}{F_{jr} + F_{jl}} \tag{17}$$

$$(LDC)_{max} = \sum (LDC)_{jmax} \quad (j=1:1:6) \tag{18}$$

The maximum Safety Rollover Coefficient $(SRC)_{max}$ on the vehicle is calculated by the formula (19). The tractor semi-trailer with 6 axles shows signs of motion instability and will rollover if $(SRC)_{max} = 1$ (Chen, 2016; Jazar, 2017,2019).

$$(SRC)_{max} = \sum \max(LDC)_j \quad (j = 2:1:6) \tag{19}$$

4. Simulations and comments

Matlab-Simulink software is used to solve a system of differential equations and simulate the effects of vehicle velocity on signs of instability and rollover when the vehicle is undergoing lane changes, cornering or turning with different steering angles. Specifically, we have investigated the impact of velocity and steering angle on the rollover stability of a fully loaded tractor semi-trailer. We have established the following conditions for our study: the vehicle has a center of gravity height of 2.3m, it is moving on a road with a coefficient of friction of 0.8, and the steering wheel is rotated according to the Ramp Steer Maneuver (RSM) with a changing steering angle of $LSA=\phi_r=[2:2:12]$ deg.

The simulation is conducted at different velocities, ranging from 30 km/h to 60 km/h, to explore the influence of velocity on the vehicle's stability and potential for rollover during lane changes, cornering, and turning.

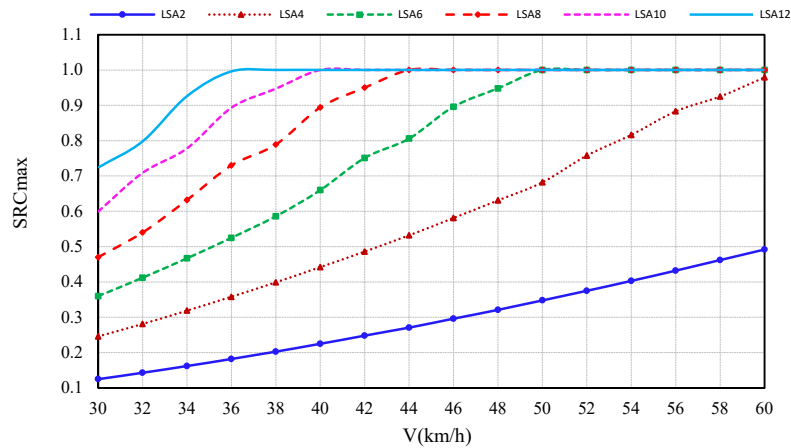


Fig. 3. The maximum Safety Rollover Coefficient $(SRC)_{max}$

Fig. 3 shows the change of the maximum Safety Rollover Coefficient $(SRC)_{max}$ of the vehicle when the vehicle is moved at different velocities, $V=[30:2:60]$ km/h, corresponding to the steering angle LSA, $\phi_r=[2:2:12]$ deg. When the vehicle is moving at $V=[30:2:60]$ km/h and the steering wheel is turned with $\phi_r < 4$ deg, then $(SRC)_{max} < 1$, the vehicle moves very safely. Similarly, in order for the vehicle to change lanes, corner, turn around to ensure safety and not roll over, the driver must comply with the following: (1) if the vehicle velocity is 50km/h, the steering angle must be less than 6deg; (2) if the vehicle velocity is $V=44$ km/h, the steering angle must be less than 8deg; (3) if the vehicle velocity is $V=40$ km/h, the steering angle must be less than 10deg and (4) if the vehicle velocity is $V=36$ km/h, the steering angle must be less than 12deg.

The maximum Load Distribution Coefficient $(LDC)_{max}$ of the vehicle when the vehicle is moved at different velocities, $V=[30:2:60]$ km/h, corresponding to the steering angle $\phi_r=[2:2:12]$ deg is shown in Fig. 4. The vehicle loses its stability and rollover $LDC_{max}=1$ when the vehicle velocity is $V=38$ km/h and the steering angle $\phi_r > 12$ deg; or the vehicle velocity is $V=40$ km/h and the steering angle $\phi_r > 10$ deg; or the vehicle velocity is $V=44$ km/h and the steering angle $\phi_r > 8$ deg; or the vehicle velocity is $V=50$ km/h and the steering angle $\phi_r > 6$ deg.

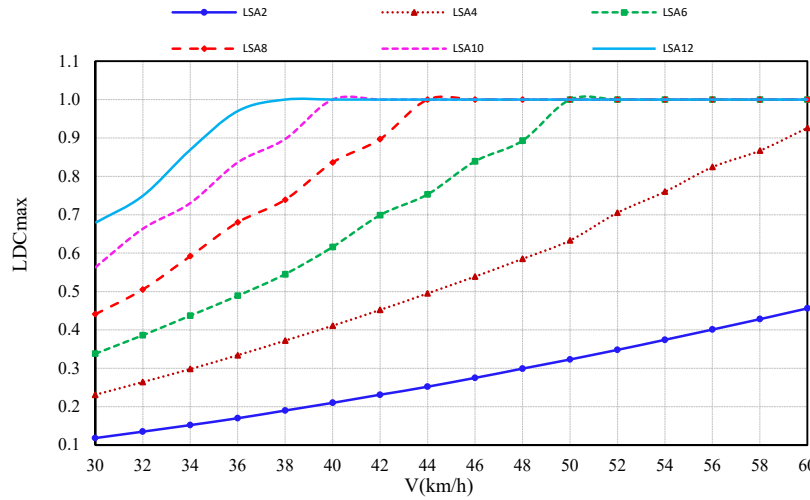
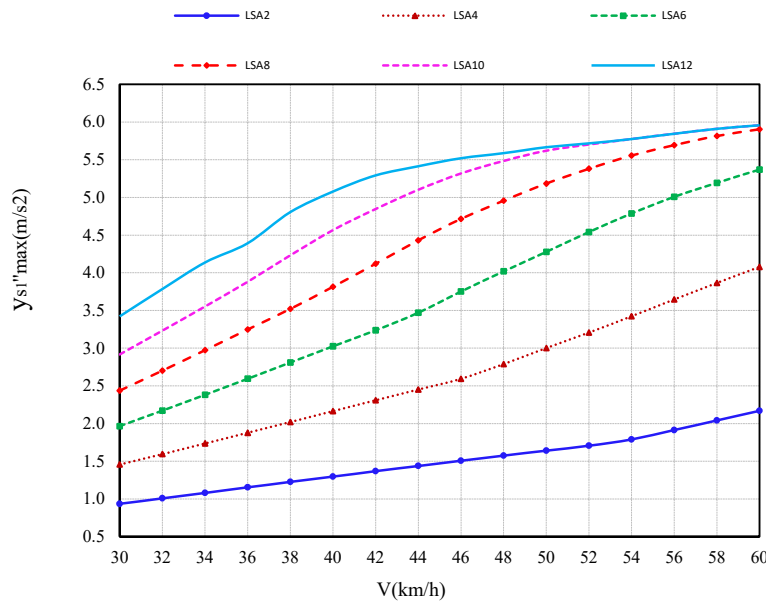


Fig. 4. The maximum Load Distribution Coefficient $(LDC)_{max}$



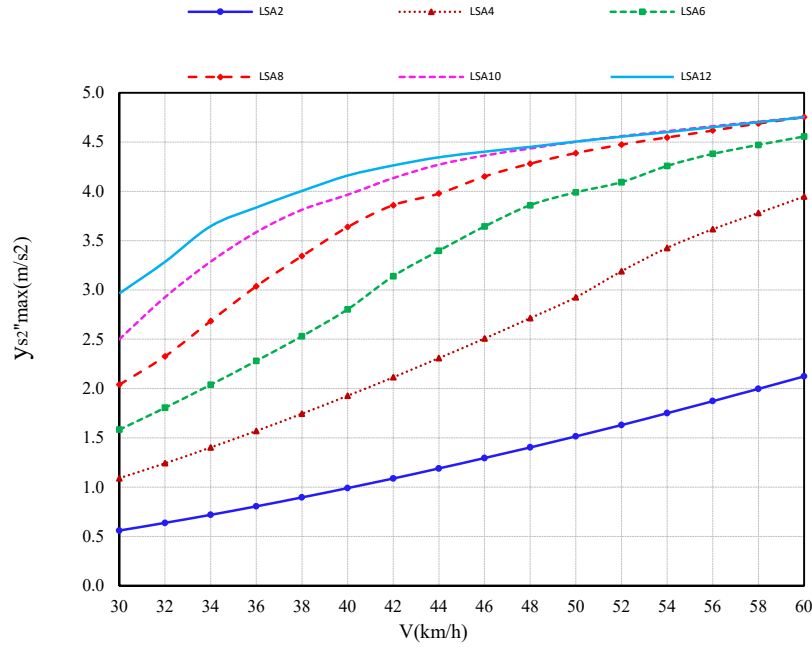
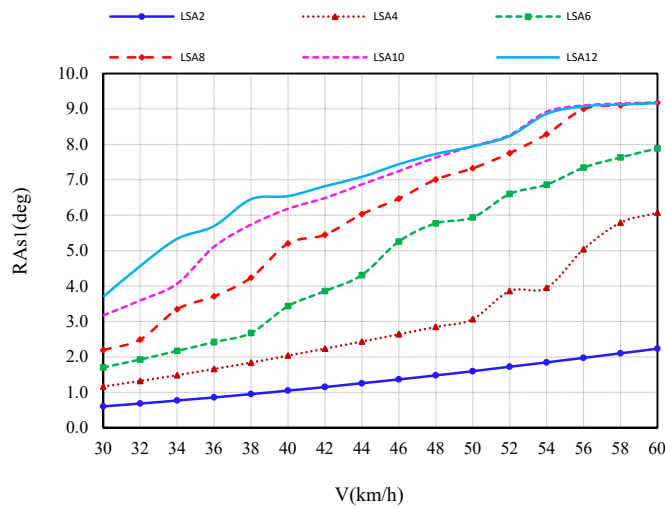


Fig. 5. The maximum lateral acceleration of the tractor and semi-trailer.

Fig. 5 shows the change of the maximum lateral acceleration of the tractor (\ddot{y}_{s1}) and semi-trailer (\ddot{y}_{s2}) when the vehicle is moved at different velocities, $V=[30:2:60]$ km/h, corresponding to the steering angle $\phi_l=[2:2:12]$ deg. The higher the vehicle speed, the greater the maximum lateral acceleration of the vehicle. At the same time, the larger the steering angle of the vehicle, the greater the maximum lateral acceleration of the vehicle. When the steering angle is 2deg and the vehicle speed is $V=[30:2:60]$ km/h, the lateral acceleration of the vehicle are $\ddot{y}_{s1max} = (0.94 \div 2.17)m/s^2$ and $\ddot{y}_{s2max} = (0.56 \div 2.12)m/s^2$; . Similarly, if $\phi_l=4$ deg and $V=[30:2:60]$ km/h, then $\ddot{y}_{s1max} = (1.46 \div 4.08)m/s^2$ and $\ddot{y}_{s2max} = (1.09 \div 3.95)m/s^2$; if $\phi_l=6$ deg and $V=[30:2:60]$ km/h, then $\ddot{y}_{s1max} = (1.97 \div 5.37)m/s^2$ and $\ddot{y}_{s2max} = (1.58 \div 4.56)m/s^2$; if $\phi_l=8$ deg and $V=[30:2:60]$ km/h, then $\ddot{y}_{s1max} = (2.44 \div 5.9)m/s^2$ and $\ddot{y}_{s2max} = (2.04 \div 4.75)m/s^2$; if $\phi_l=10$ deg and $V=[30:2:60]$ km/h, then $\ddot{y}_{s1max} = (2.92 \div 5.96)m/s^2$ and $\ddot{y}_{s2max} = (2.5 \div 4.76)m/s^2$; if $\phi_l=12$ deg and $V=[30:2:60]$ km/h, then $\ddot{y}_{s1max} = (3.42 \div 5.96)m/s^2$ and $\ddot{y}_{s2max} = (2.96 \div 4.75)m/s^2$.



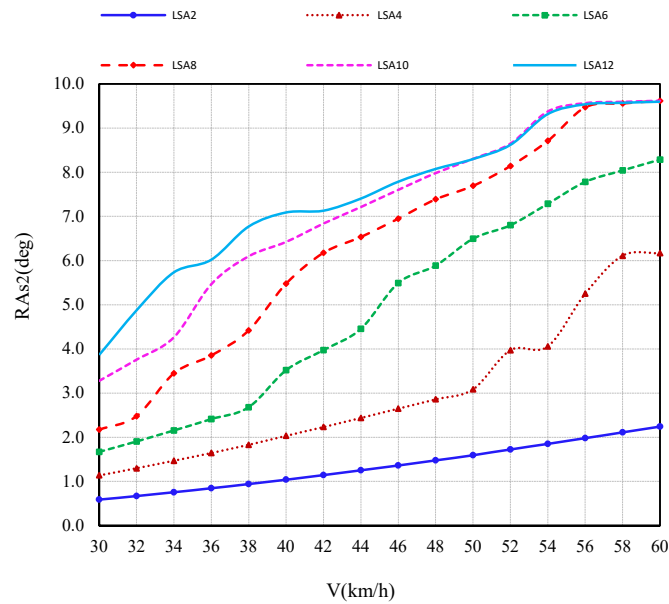


Fig. 6. The roll angle of the tractor and semi-trailer

The roll angle of the tractor (β_{s1}) and semi-trailer (β_{s2}) when the vehicle is moved at different velocities, $V=[30:2:60]$ km/h, corresponding to the steering angle $\phi_f=[2:2:12]$ deg is shown in Fig. 6. The higher the vehicle speed, the greater the roll angle of the vehicle. Simultaneously, the larger the steering angle of the vehicle, the greater the roll angle of the vehicle. When the steering angle is $\phi_f=2$ deg and the vehicle speed is $V=[30:2:60]$ km/h, the roll angle of the tractor $\beta_{s1}=[0.6\div 2.23]$ deg and the roll angle of the semi-trailer $\beta_{s2}=[0.59\div 2.24]$ deg. Similar, if $\phi_f=4$ deg and $V=[30:2:60]$ km/h, then $\beta_{s1}=[1.16\div 6.07]$ deg and $\beta_{s2}=[1.14\div 6.17]$ deg; if $\phi_f=6$ deg and $V=[30:2:60]$ km/h, then $\beta_{s1}=[1.7\div 7.88]$ deg and $\beta_{s2}=[1.67\div 8.29]$ deg; if $\phi_f=8$ deg and $V=[30:2:60]$ km/h, then $\beta_{s1}=[2.19\div 9.18]$ deg and $\beta_{s2}=[2.17\div 9.62]$ deg; if $\phi_f=10$ deg and $V=[30:2:60]$ km/h, then $\beta_{s1}=[3.17\div 9.19]$ deg and $\beta_{s2}=[3.28\div 9.62]$ deg; if $\phi_f=12$ deg and $V=[30:2:60]$ km/h, then $\beta_{s1}=[3.7\div 9.17]$ deg and $\beta_{s2}=[3.86\div 9.6]$ deg.

5. Conclusions

In conclusion, this study aimed to investigate the influence of vehicle velocity and steering angle on the rollover stability of a fully loaded tractor semi-trailer. The simulations were conducted with a vehicle that had a center of gravity height of 2.3m, was moving on a road with a coefficient of friction of 0.8, and the steering angle changed between 2 to 12 degrees. The study found that in order to ensure that the vehicle remains stable and does not roll over during lane changes, cornering, and turning, the maximum safety rollover coefficient and the maximum load distribution coefficient must be less than 1.

Based on the simulations, the study provided specific recommendations for the maximum steering angle at different velocities to prevent rollover. For instance, when the vehicle is moving at 60 km/h, the steering angle must be less than 4 degrees. Similarly, if the vehicle is moving at 50 km/h, the recommended maximum steering angle is 6 degrees, while at 44 km/h, the steering angle should be less than 8 degrees. At lower speeds, the recommended steering angle increases to ensure the vehicle remains stable.

These findings demonstrate the importance of carefully considering vehicle design and safety testing to minimize the risk of rollover accidents. By utilizing simulations and calculations based on factors such as velocity, steering angle, and center of gravity height, designers and engineers can make informed decisions to ensure that vehicles remain stable and safe under various conditions.

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