

Analyzing Rollover Indices for Critical Truck Maneuvers

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ABSTRACT

Rollover has for long been a major safety concern for trucks, and will be even more so as automated driving is envisaged to becoming a key element of future mobility. A natural way to address rollover is to extend the capabilities of current active-safety systems with a system that intervenes by steering or braking actuation when there is a risk of rollover. Assessing and predicting the rollover is usually performed using rollover indices calculated either from lateral acceleration or lateral load transfer. Since these indices are evaluated based on different physical observations it is not obvious how they can be compared or how well they reflect rollover events in different situations.

In this paper we investigate the implication of the above mentioned rollover indices in different critical maneuvers for a heavy 8×4 twin-steer truck. The analysis is based on optimal control applied to a five degrees of freedom chassis model with individual wheel dynamics and high-fidelity tire-force modeling. Driving scenarios prone to rollover accidents are considered, with a circular-shaped turn and a slalom maneuver being studied in-depth. The optimization objective for the considered maneuvers are formulated as minimum-time and maximum entry-speed problems, both triggering critical maneuvers and forcing the vehicle to operate on the limit of its physical capabilities. The implication of the rollover indices on the optimal trajectories is investigated by constraining the optimal maneuvers with different rollover indices, thus limiting the vehicle's maneuvering envelope with respect to each rollover index. The resulting optimal trajectories constrained by different rollover indices are compared and analyzed in detail. Additionally, the conservativeness of the indices for assessing the risk of rollovers are discussed.

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1. INTRODUCTION

With the growing demand for goods and transportation, the numbers of on-road trucks are on the rise. Judging from the trend of global truck ownership shown in Figure 1, it is clear that the number of trucks on road will continue to increase for years to come.

One notable difference between trucks and light duty vehicles is the discrepancy of their respective center of gravity height. A higher center of gravity increases the chance of rollover accidents [14].

According to the 1995 U.S. truck statistics, rollover occurred in 53.8% of the first harmful event of non collision-type crash that involved trucks and buses [4]. Although rollover accidents only occur in 3.6% of all truck accidents, they account for 7.9% of tow-away accidents, 9.9% of injury accidents, and 12.3% of fatal accidents involving trucks. Moreover, 58% of fatal injuries to the truck driver occurred in the rollover crashes [14]. Additionally, rollover accidents pose danger to the public safety, where 95% of all accidents involved bulk spillage of hazardous commodities started with rollover [12]. These statistics show the severity of rollover events can cause significant damage disproportional to the incidence rate [5; 8].

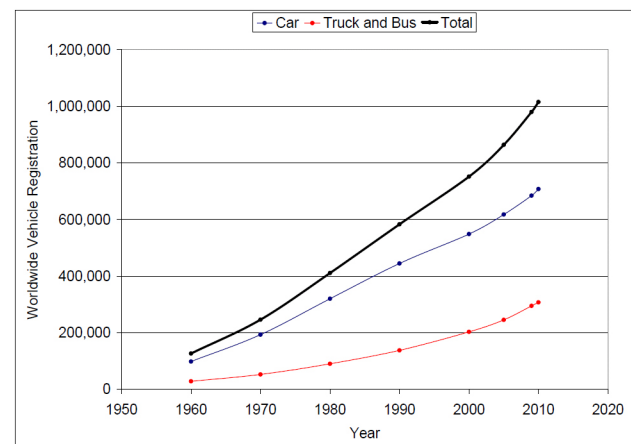


Figure 1. The trend of worldwide vehicle registration [6; 7].

Apart from setting legislation demands on road design and limiting the driving speed, two potential approaches to deal with rollover prevention is to offer an early warning to the driver whenever there is a risk of rollover using a rollover indicator [13], and to intervene by active steering or braking [3]. These systems require quantitative measures to assess the risk of rollover, which are commonly referred to as rollover indices.

There are several rollover indices that have been proposed in the past, where the two most commonly adopted indices are presented and analyzed in this paper. The first one is based on the lateral acceleration [14], followed by the lateral load transfer [9]. Ideally, a good rollover index should give a non-conservative risk assessment on the rollover based on the current vehicle state while having fewer requirements on the measurement and knowledge of vehicle parameters. To balance these contradicting criteria, a systematic analysis is required to assess factors that affect the accuracy of indices and to establish a consensus for a good choice among these indices.

To support the analysis, a method for generating driving scenarios prone to rollover is needed. To this end, an optimization approach is proposed, where optimal trajectories are calculated based on the minimum-time and maximum entry-speed problems. Additionally, a five degrees of freedom chassis model that incorporates the longitudinal and lateral translations, together with roll, pitch and yaw rotations is derived to facilitate the optimizations. Furthermore, non-linear tire dynamics are also included in the chassis model to handle maneuvers with large, and combined, longitudinal and lateral slip. The conservativeness of the rollover are assessed by constraining the optimal solutions with different rollover indices. Moreover, analytical methods based on the vehicle dynamics are sought to analyze the relationships between the rollover indices.

The paper is outlined as follows. In [Section 2](#), the vehicle chassis and tire modeling is described, followed by an overview and analysis of the rollover indices in [Section 3](#). In [Section 4](#), the optimal-control problem is defined and the solution procedure described. The results from the optimizations are subsequently presented and discussed in [Section 5](#). Finally, conclusions regarding the main results and aspects on future work are given in [Section 6](#).

2. MODELING OF A TWIN-STEER TRUCK

The vehicle is modeled with eight individual wheels, divided on four axles here denoted as a , b , c , and d as depicted in [Figure 2](#). The two foremost axles (i.e. axles a and b) are steered with the steer input δ , controlling the steering angles according to

$$\delta_a = \delta, \quad \delta_b = k_\delta \delta,$$

where δ_a and δ_b are the steering angles for the wheels on axle a and b , and k_δ represents a fixed steering ratio between these axles. The chassis is modeled with mass m and moment of inertia I_{xx} , I_{yy} , and I_{zz} about the coordinate axes. The suspended chassis mass has five degrees of freedom consisting of the planar motions (longitudinal and lateral translation, and yaw rotation), roll and pitch. The equations of motion for the chassis are derived based on the Euler-Lagrange equations. The full set of equations are omitted here for space and readability reasons.

The suspension system is modeled as a rotational spring-and-damper system, where the chassis rotates about the roll center (rotations about the x -axis) and pitch center (rotations about the y -axis). The roll center is assumed to be located in the middle of the car in the longitudinal and lateral direction, at a height of h_{rc} . The pitch center is

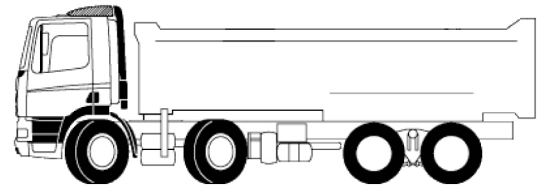
also located in the middle of the car, but is vertically positioned in the ground plane. The roll moment τ_ϕ produced by the suspension system can be described by

$$\tau_\phi = (K_{\phi,a} + K_{\phi,b} + K_{\phi,c} + K_{\phi,d})\phi + (D_{\phi,a} + D_{\phi,b} + D_{\phi,c} + D_{\phi,d})\dot{\phi}, \quad (1)$$

where ϕ is the roll angle, and K_ϕ and D_ϕ are stiffness and damping parameters for the respective axles. Similarly, the pitch moment τ_θ can be described by

$$\tau_\theta = K_\theta \theta + D_\theta \dot{\theta}, \quad (2)$$

where θ is the pitch angle, and K_θ and D_θ are the lumped stiffness and damping contributions from all axles.



Axle:	a	b	c	d
Left Wheel:	1	3	5	7
Right Wheel:	2	4	6	8

Figure 2. An 8x4 twin-steer tipper truck.

The longitudinal load transfer between the axles is determined from

$$(F_{z,1} + F_{z,2} + F_{z,3} + F_{z,4}) \frac{l_a + l_b}{2} - (F_{z,5} + F_{z,6} + F_{z,7} + F_{z,8}) \frac{l_c + l_d}{2} = K_\theta \theta + D_\theta \dot{\theta}, \quad (3)$$

$$\sum_{i=1}^8 F_{z,i} = mg, \quad (4)$$

where F_z is the normal force for each wheel, l is the longitudinal distance from each axle to the center of gravity, and the left and right wheels are indexed by odd and even numbers respectively. The lateral load transfer for each axle is described by

$$-w(F_{z,1} - F_{z,2}) = (F_{y,1} + F_{y,2})h_{rc} + K_{\phi,a}\phi + D_{\phi,a}\dot{\phi}, \quad (5)$$

$$-w(F_{z,3} - F_{z,4}) = (F_{y,3} + F_{y,4})h_{rc} + K_{\phi,b}\phi + D_{\phi,b}\dot{\phi}, \quad (6)$$

$$-w(F_{z,5} - F_{z,6}) = (F_{y,5} + F_{y,6})h_{rc} + K_{\phi,c}\phi + D_{\phi,c}\dot{\phi}, \quad (7)$$

$$-w(F_{z,7} - F_{z,8}) = (F_{y,7} + F_{y,8})h_{rc} + K_{\phi,d}\phi + D_{\phi,d}\dot{\phi}, \quad (8)$$

where w is the half-track width.

2.1. Wheel and Tire Modeling

Slip angle α and slip ratio κ are described following the definition in [11],

$$\dot{\alpha}_i \frac{\sigma}{v_{x,i}} + \alpha_i = -\arctan\left(\frac{v_{y,i}}{v_{x,i}}\right), \quad (9)$$

$$\kappa_i = \frac{R_w \omega_i - v_{x,i}}{v_{x,i}}, \quad i \in \{1, 2, 3, 4, 5, 6, 7, 8\}, \quad (10)$$

where σ is the relaxation length, R_w is the wheel radius, and $v_{x,i}$ and $v_{y,i}$ are the longitudinal and lateral velocities resolved in the frame of wheel i . The rotational wheel dynamics for each wheel is described by

$$T_i - I_w \dot{\omega}_i - F_{x,i} R_w = 0, \quad i \in \{1, 2, 3, 4, 5, 6, 7, 8\}. \quad (11)$$

where ω is the wheel velocity, I_w the wheel inertia, and F_x the longitudinal tire force. The wheel torque T is controlled with a braking signal u_{brake} , applying braking moments with a fixed brake-force distribution between the wheels, and an acceleration signal u_{acc} , generating a driving moment on axle c and d (i.e., on wheel 5, 6, 7, and 8).

The nominal tire forces, describing the tire forces for pure slip, are determined using the *Magic Formula* tire model [11], defined by

$$F_{x0,i} = \mu_{x,i} F_{z,i} \sin(C_{x,i} \arctan(B_{x,i} \kappa_i - E_{x,i}(B_{x,i} \kappa_i - \arctan B_{x,i} \kappa_i))), \quad (12)$$

$$F_{y0,i} = \mu_{y,i} F_{z,i} \sin(C_{y,i} \arctan(B_{y,i} \alpha_i - E_{y,i}(B_{y,i} \alpha_i - \arctan B_{y,i} \alpha_i))), \quad i \in \{1, 2, 3, 4, 5, 6, 7, 8\}. \quad (13)$$

Here μ_x and μ_y represent the friction coefficients, and B , C , and E are model parameters. Combined slip is modeled using *weighting functions* [11], G_{xo} and $G_{y\kappa}$, scaling the nominal tire forces. Consequently, the longitudinal and lateral tire forces, F_x and F_y , are given by

$$H_{x\alpha,i} = B_{x1,i} \cos(\arctan(B_{x2,i} \kappa_i)), \quad (14)$$

$$G_{x\alpha,i} = \cos(C_{x\alpha,i} \arctan(H_{x\alpha,i} \alpha_i)), \quad (15)$$

$$F_{x,i} = F_{x0,i} G_{x\alpha,i}, \quad i \in \{1, 2, 3, 4\}, \quad (16)$$

$$H_{y\kappa,i} = B_{y1,i} \cos(\arctan(B_{y2,i} \alpha_i)), \quad (17)$$

$$G_{y\kappa,i} = \cos(C_{y\kappa,i} \arctan(H_{y\kappa,i} \kappa_i)), \quad (18)$$

$$F_{y,i} = F_{y0,i} G_{y\kappa,i}, \quad i \in \{1, 2, 3, 4, 5, 6, 7, 8\}, \quad (19)$$

where B and C are model parameters. The vehicle model parameters and tire model parameters are presented in Table 1 and 2 respectively.

Table 1. Vehicle model parameters.

Notation	Value	Unit
l_a	2.75	m
l_b	0.81	m
l_c	2.15	m
l_d	3.51	m
w	1.03	m
m	16 930	kg
I_{xx}	6 167	kgm ²
I_{yy}	31 440	kgm ²
I_{zz}	28 030	kgm ²
R_w	0.5	m
I_w	4.0	kgm ²
σ	0.3	m
g	9.82	m/s ²
h_{cg}	1.4	m
h_{rc}	0.6	m
h_{pc}	0	m
k_δ	0.5	
$K_{\phi,a}, K_{\phi,b}, K_{\phi,c}, K_{\phi,d}$	359	kNm/rad
$D_{\phi,a}, D_{\phi,b}, D_{\phi,c}, D_{\phi,d}$	32.3	kNms/rad
K_θ	2 930	kNm/rad
D_θ	250	kNms/rad

Table 2. Tire model parameters.

Notation	Value
μ_x	1.20
B_x	11.7
C_x	1.69
E_x	0.377
μ_y	0.935
B_y	8.86
C_y	1.19
E_y	-1.21
B_{x1}	12.4
B_{x2}	-10.8
$C_{x\alpha}$	1.09
B_{y1}	6.46
B_{y2}	4.20
$C_{y\kappa}$	1.08