

Balance of Static and Dynamic Rollover Thresholds for a Three-Axle Vehicle

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ABSTRACT

In this study, a three-axle vehicle model established with ADAMS/Car is first correlated with field test data from quasi-static tilt table and highly dynamic NATO double lane change maneuver tests, respectively. It is then applied to predict the vehicle static rollover threshold (SRT) and dynamic rollover threshold (DRT). With the optimization approach proposed in this study it is possible to efficiently tune the anti-roll bar stiffness at each axle, to either maximize SRT or DRT, or balance both. The sensitivity results derived from the optimization iteration process can be applied to effectively size the three anti-roll bars that balance the static and dynamic roll stability performances. The proposed method can be potentially applied to include other parameters to address the roll stability issues and beyond.

INTRODUCTION

Highway safety related to vehicle operation on the road is a complex function of dynamic interactions between the vehicle, the driver and the environment. The safety dynamics of a vehicle directly relates to its rollover stability and control limits. Based on [1], the probability of single vehicle rollover accidents is directly related to their static rollover thresholds (SRT), as shown in Figure 1. As the basic measure of roll stability, the SRT will be expressed in this study as steady-state lateral acceleration in gravitational units (g) when the first tire lifts off the ground. Most passenger cars have rollover thresholds greater than 1.0 g, while light trucks, vans, and SUVs range from 0.8 to 1.2 g [1]. However, the rollover threshold of a loaded heavy truck often lies well below 0.5 g.

There are several factors affecting the vehicle SRT, which are summarized in Table 1. Among all these factors, only the

vehicle track width always has positive influence on its SRT, i.e., the wider the track width, the greater the SRT. All other factors listed in Table 1 have negative or positive impacts on SRT. Depending upon individual vehicle design, each of these factors may have different levels of influence on SRT, but generally the vehicle CG height has the most significant negative impact.

Assuming a vehicle's compliances due to suspension, tire, and structure flexibility are negligible, its SRT on flat surface can be simply calculated as shown in Equation (1). The resultant value from this simple calculation yields the upper limit of the vehicle SRT, which is usually called as static stability factor (SSF).

$$Max_SRT = SSF = \frac{T}{2h_{cg}} \quad (1)$$

Where T and h_{cg} are the vehicle track width and CG height, respectively.

When a vehicle experiences a dynamic event such as double lane change maneuver, the influences on SRT from its compliances, damping forces and mass moment of inertia cannot be ignored. The dynamic rollover threshold (DRT) of a vehicle is thus defined in Equation (2) [2]:

$$DRT = \frac{a_y|_{F_z=0}}{g} \quad (2)$$

where a_y is the vehicle CG lateral acceleration when the first tire lifts off the ground (i.e. the tire vertical force, F_z , becomes zero), and g is the gravitational constant. This definition may lead to various values of DRT associated with

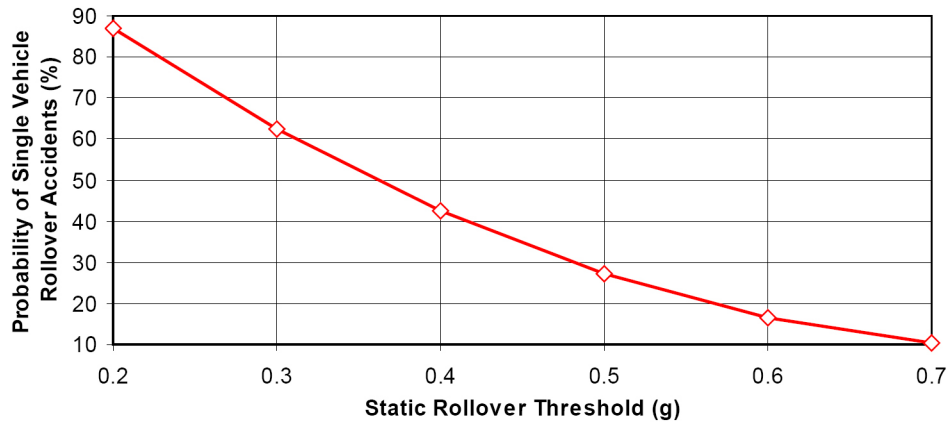


Figure 1. The chance of rollover is strongly influenced by the roll stability of the vehicle [1].

Table 1. Various factors affecting vehicle static roll stability.

Influence Factors	Track Width	CG Height	Tire Stiffness	Compliant Springs	Roll Stiffness Distribution	Suspension Lateral Compliance	Frame/Body Flexibility	CG Lateral Off-Center
Positive/Negative Influence on Static Roll Stability	Positive	Negative	Negative	Negative	Negative or Positive	Negative	Negative or Positive	Negative

different tires lifting-off ground at different dynamic maneuvers. It is very complicated to express DRT as an explicit function of all factors affecting it. Therefore in this study, a simulation with a multi-body system modeling approach is applied to estimate the vehicle's DRT.

In this study, a three-axle vehicle is chosen to demonstrate the SRT and DRT sensitivity to the anti-roll bar stiffness at each axle. The three-axle full vehicle model is first validated with static and dynamic test events, including tilt table and NATO double lane change maneuvers, and then applied to predict the SRT and DRT. Then an optimization approach is applied to efficiently tune the anti-roll bar stiffness at each axle to either maximize SRT or DRT, or balance both.

TEST EVENTS USED IN THIS STUDY

In this section, the two test events used for correlation and rollover threshold prediction are briefly introduced. These two events include tilt table and NATO double lane change maneuver.

TILT TABLE TEST

A tilt table, as shown in [Figure 2](#), is a device used to determine the vehicle's SRT [1]. It is essentially a pivoting platform on which vehicles are installed and tilted until one or more uphill tires lift off the platform. This test facility is designed to simulate the roll plane behavior of a vehicle in a steady-state turn. Therefore the tilt table test should be conducted at a very low rate of change in tilt angle.



Figure 2. Tilt table testing.

The SRT result from the tilt table test [3] is derived from the measured tilt angle (ϕ) when the first tire lifts off the table, given by [Equation \(3\)](#).

$$SRT = \tan(\phi) \quad (3)$$

Since the whole vehicle is included during the tilt table test, the test includes the contributions of all factors listed in [Table 1](#). The contributions from tire side slip, vehicle mass moment of inertia and damping forces, however, are not included due to the slowly tilting process. It should be noted about [Equation \(3\)](#) that in the tilt table method, one component of gravity, $g\sin\phi$, acts laterally while the other component, $g\cos\phi$, acts perpendicular to the tilt table surface. Assuming $g\cos\phi$ simulates gravity, then simulated lateral acceleration is $(g\sin\phi)/(g\cos\phi)$ or $\tan\phi$. Thus, if the table angle is slowly increased, the tangent of the tilt angle at which the vehicle rolls over can estimate the lateral acceleration (in g's) at

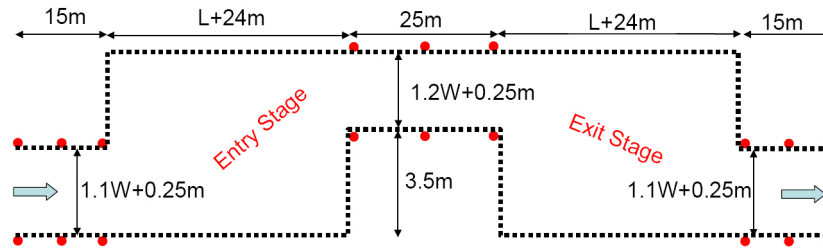


Figure 3. NATO (AVTP 03-160W) Lane Change Test Course (W : vehicle width, L : vehicle length).

which the vehicle reaches its static roll stability limit [3]. If the tilt angle is too large, the tilt table measurement may not be representative to the actual vehicle roll plane behavior of a steady-state turn. This may indicate that a vehicle with a large SRT should not be tested on a tilt table, since it may yield a significant estimation error.

NATO DOUBLE LANE CHANGE

The double lane-change test simulates an emergency maneuver to determine vehicle transient handling dynamics. The test is important because the more controllable and secure a vehicle is when pushed to its handling limits, the better chance the driver has of avoiding an accident. In situations where an obstacle is in the way, steering around it can cause the vehicle to go out of control and result in a vehicle rollover or obstacle impact. This can happen due to the compromising nature of vehicles in general (such as tradeoffs between performance handling and ride comfort). In the double lane-change maneuver, rollovers are more common on vehicles with high center of gravity (CG) especially if swerving abruptly. An inexperienced driver may not know how fast the vehicle he/she is driving can negotiate a lane change without rollover. Alternatively, a driver at a proving ground may not know how to control the steering wheel effectively so that he/she does not violate the double lane change boundary specified with the red-color pylon as shown in [Figure 3](#).

In an early stage of vehicle development, modeling and simulation engineers often employ double-lane change maneuvers to evaluate a vehicle's transient handling dynamics [4]. In such events, the lane boundaries are given with a few specified pylons. The locations of these pylons are dependent upon the vehicle length and width (NATO double lane change AVTP-03-160W for example). Evaluation of the vehicle's dynamic capabilities involves finding (and demonstrating) the maximum speed at which the vehicle can navigate through the course 1) without violating the lane boundaries and 2) without any tire normal force becoming zero. A course boundary violation is defined as any part of the tire sidewall or vehicle corners that crosses over a boundary. The higher the speed at which the vehicle can pass the event successfully (no boundary violations and no zero

tire normal forces), the more stable the vehicle's transient dynamics are.

VEHICLE MODEL AND ITS CORRELATIONS

The objective of this study is to investigate the rollover threshold as function of anti-roll bar stiffness at three axles in both the static and dynamic domains. Therefore it is essential to establish an accurate vehicle model that can be well correlated with test data from both static and dynamic events, before using it to perform any simulation and analysis.

In this study, a prototype 6×6 vehicle is chosen such that majority of the key vehicle model parameters are available. The vehicle model is illustrated in [Figure 4](#), where the front and rear suspensions are typical short-long arm (SLA) with coil springs and anti-roll bars. The two rear axles are independent. The vehicle body and powertrain (not shown) are modeled as rigid parts and its masses and moments of inertia are derived from the CATIA model. All of the mounts or bushings and their forces are assumed to be nonlinear functions of their deflections and linear functions of velocities in the three translational and three rotational directions. All suspension components are represented with rigid bodies (their flexible mode frequencies being much higher than the concerned road and steering input frequencies). The PAC2002 Tire model [5] is applied to calculate the tire patch forces. The total kinematic degrees of freedom of the full vehicle model are 143. The model is first well correlated with the target kinematics and compliance properties before using it for full vehicle simulations.

For correlation, the corner weights of the model are first compared with those measured, as shown in [Table 2](#). The maximum corner weight difference between the model and test vehicle is 2.02%. For the axle weight, the maximum difference is 0.64%. This shows that the total vehicle CG location in horizontal plane is very close to that of test vehicle.

Based on [Equation \(1\)](#), the SSF of the concerned vehicle can be calculated as 0.79. The tilt table simulated SRT is 0.57 which is the same as the measured value. It is also important to mention that both simulation and test show that the right