

Development of a Multibody Dynamics Based Simulation of a Lift-Truck in a High Speed Turn to Evaluate Dynamic Lateral Stability

Joshua A. Hogg
NACCO Materials Handling Group, Inc.

ABSTRACT

This paper discusses the development and preliminary validation of a dynamic stability test simulation to fulfill a new dynamic lateral stability requirement for lift trucks sold in the European market.

A representative lift truck was chosen and instrumented to capture vehicle path, travel speed, and lateral acceleration during a high speed turn. The vehicle was simulated using a Multi-Body Dynamics (MBD) approach in a commercially available software package.

Experimental data showed significant variation in a human operator's vehicle path, which had a strong effect on the measured lateral acceleration during a high speed turn. Preliminary comparisons showed reasonable agreement between the simulation model and physical test data. Additionally, it was found that the simulation provided a consistent measure of comparison to evaluate the effect of design changes to observed lateral stability.

CITATION: Hogg, J., "Development of a Multibody Dynamics Based Simulation of a Lift-Truck in a High Speed Turn to Evaluate Dynamic Lateral Stability," *SAE Int. J. Commer. Veh.* 8(2):2015, doi:10.4271/2015-01-2752.

INTRODUCTION

Effective in 2014, a newly drafted standard [1] requires all conventional counterbalanced lift trucks with capacities up to 5000kg sold into the European market meet new dynamic test criteria. The new test involves navigating a 90 degree turn within a specified lane at 90% of the vehicle's maximum travel speed. Failure of a trial is determined by visible lift-off of the inside rear steer tire (for conventional 4 wheel trucks). The standard also permits the use of simulation, provided the results of such computer modeling are validated based on physical testing.

The dynamic behavior of a vehicle in a high speed turning event is highly sensitive to multiple variables, including human factors (operator experience, skill) as well as environmental factors (temperature, ground surface, etc.) which are independent from the characteristic stability of the test vehicle. The rear wheel steering mechanism common to counter balanced lift-trucks makes repeatability of a turning event particularly sensitive to the skill of the test operator. As of this writing, EN16203:2014 [1] requires only 3 "successful" attempts (defined as ones where the test vehicle completes the course within the specified boundaries) out of 20. The objective of this study was to evaluate the effectiveness of simulation to characterize the general physics of a turning event similar to that described in [1] and to provide a preliminary validation with physical testing.

EXPERIMENTAL TESTING

In order to capture the general dynamic behavior of a particular test vehicle, a test plan was defined to perform multiple test trials varying both entry speed and turning radius. Turning radius was established by reducing the exit corridor width of the test course. Previous work presented in [2] took a similar approach. Figure 1 shows the test course used for all physical test trials presented. For the first set of trials, the target exit corridor width was reduced until a lift-off of the inside steer tire was observed. For the second set of trials, entry speed was decreased until drive tire lift-off did not occur. These conditions for lift-off were used as additional points of comparison.

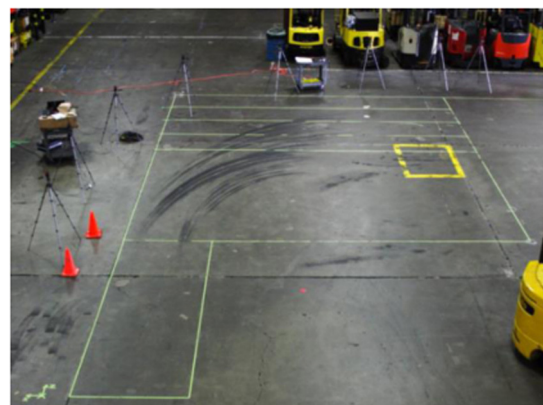


Figure 1. Test course used for physical dynamic stability testing

The testing consisted of 3 trials at each of the conditions presented in Table 1. In addition to physical instrumentation, a test witness would indicate whether lift-off of either drive or steer tire occurred for each trial. The truck was instrumented with an accelerometer to measure lateral acceleration during the turning event. To monitor vehicle path and speed, a motion capture (Mo-Cap) system was used to track a point mounted at the front of the vehicle.

Table 1. Physical test parameters. Target exit corridor width is used as an identifier for specific test trials, though actual exit corridor distance for each trial varied.

	Entry Speed (m/s)	Target Exit Corridor Width (m)
Constant Entry Speed, Varying Radius	5.0	8.368
		7.368
		6.368
		5.368
		4.368
Constant Radius, Varying Entry Speed	5.0	6.368
	4.8	
	4.4	
	3.9	

Measured Vehicle Path

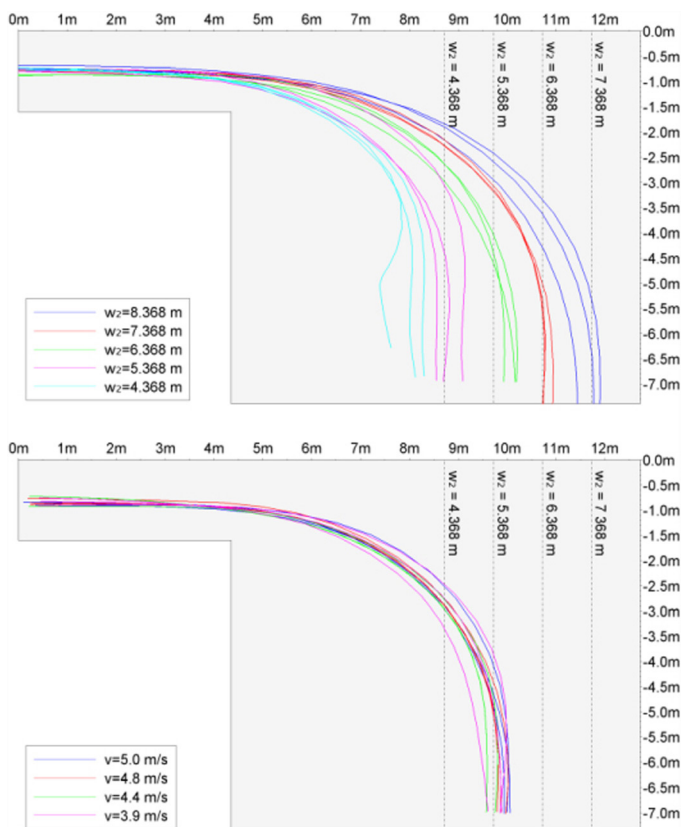


Figure 2. Measured vehicle path for constant entry speed with variable turning radius trials (above) and constant turning radius with variable entry speed trials (below).

A single point measurement was made of the test vehicle using an array of Mo-Cap cameras. Using a single point did not allow for measurement of rigid body rotations; however, it did provide a tangential point position trajectory and instantaneous speed during the test event. Figure 2 presents the measured vehicle trajectory for each test trial presented in Table 1. The Mo-Cap system recorded vehicle motion at 120 fps with a mean calibration error of less than 1mm.

Capturing Vehicle Speed

Instantaneous vehicle speed was calculated numerically using a finite difference method for the forward and lateral component position data. The tangential velocity was determined as the vector sum of the component velocities.

Due to signal loss issues with the Mo-Cap system, the position measurements often resulted in a non-uniform time increment of the measured data which made frequency based filtering of the raw position data a challenge. Numeric differentiation of the raw position data resulted in large spikes due to amplification of noise. To eliminate these spikes, velocities were determined from a spline fit of the raw position data. Figures 3 and 4 present a typical example of the calculated velocity for a given trial. Calculated entry speeds based on spline fit data were within 5% of independent radar gun speed measurements.

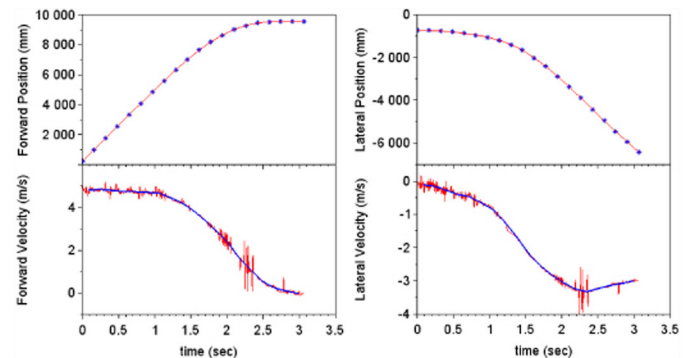


Figure 3. Typical calculated velocities based on measured position data. Red lines represent unprocessed position data and resulting calculated velocities. Blue lines represent spline fit position data and corresponding calculated velocities.

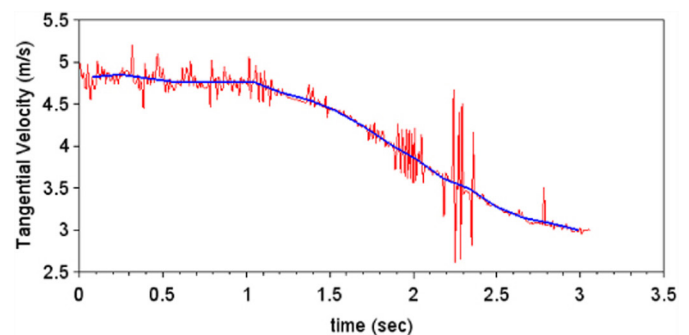


Figure 4. Calculated tangential velocity for both the raw measured and spline fit component position data. Spikes in the velocity calculation based on raw position data are due to noise amplification in the finite difference calculation. Calculated entry speeds based on spline fit data were within 5% of independent radar speed measurements.

A summary of the observed instantaneous velocity is presented in Figure 5. As can be seen, there is a significant reduction in speed during the turn that increases as the turning radius is reduced for a given entry speed. The deceleration also increases with increasing entry speed. This is believed to be due primarily to 2 reasons:

1. Due to the effect of the open differential, in the event of a lift-off of the front inside drive tire there is an immediate loss of tractive force. In this condition the vehicle is essentially coasting to a stop until both drive tires are again in contact with the ground. In the trials tested, almost every run involved a lift-off of the inside drive tire.
2. There was a significant amount of tire scrub of the rear steer tires observed during a typical test trial. This is evidenced by the tire markings in Figure 1. In the event of tire scrub, there is a significant amount of drag experienced by the vehicle, which acts as a braking force.

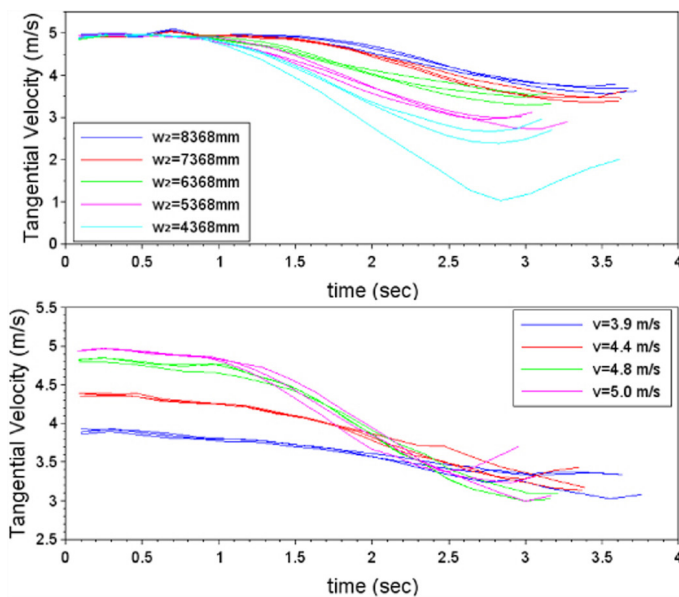


Figure 5. Calculated total velocity for constant entry speed with variable turning radius trials (above) and constant turning radius with variable entry speed trials (below).

Measured Lateral Acceleration

Considering the test event analogous to the quasi-static condition depicted in Figure 6, lift off of the inside tires would occur when the product of the lateral acceleration and the vehicle mass exceed the restoring moment due to the vehicle weight. There is an additional dynamic component involved in tip-over [3,4], however, such quasi-static roll stability has been found to be a dominant quality affecting vehicle rollover [3].

The overturning moment, given by Equation 1, is directly proportional to the lateral acceleration of the vehicle during the turn. Because of this, lateral acceleration was chosen as the primary focus for comparison. From Equation 2, we would expect the observed lateral acceleration to be proportional to the square of the entry speed of the truck and inversely proportional to the turning radius.

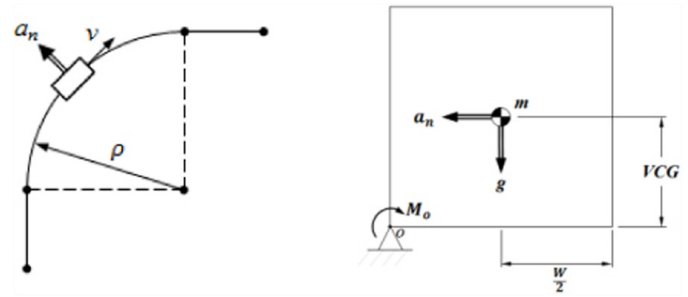


Figure 6. Simplified kinematic representation of turning event.

$$M_o = \frac{mgW}{2} - m \cdot a_n \cdot VCG \quad (1)$$

$$a_n = \frac{v^2}{\rho} \quad (2)$$

The lateral acceleration is dictated by the kinematics of the turn event, and is proportional to the overturning moment. The test requirement defined in [1], however, defines failure as lift-off of an inside steer tire. Therefore, an additional item used for qualitative comparison was visible tire lift off for each test trial.

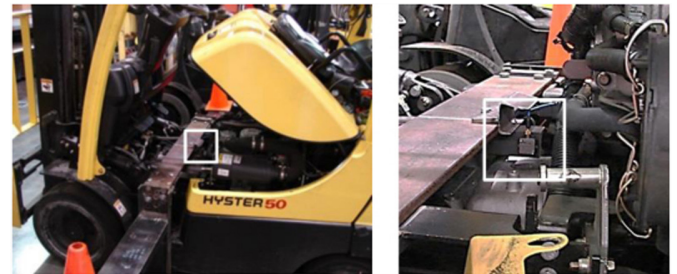


Figure 7. Accelerometer location of test vehicle.

To measure the lateral acceleration of the vehicle, a uniaxial DC accelerometer was mounted transversely at a location near the CG of the test vehicle (shown above in Figure 7). Figures 8 and 9 present the observed lateral acceleration for the test trials listed in Table 1. The direction of measured acceleration was fixed in the truck coordinates; therefore any roll experienced during the turn added a gravitational affect to measured values. This was later accounted for in comparisons to simulated data.

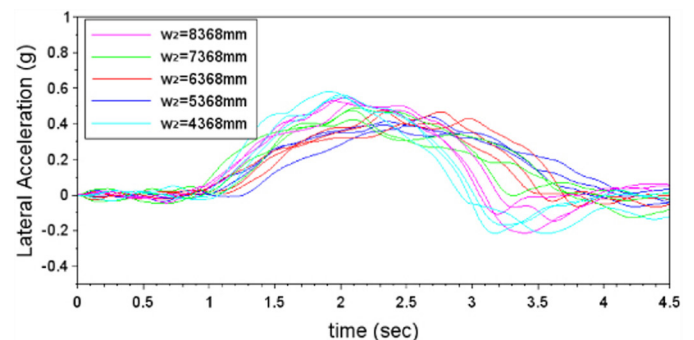


Figure 8. Observed lateral acceleration for constant entry speed with variable turning radius trials.

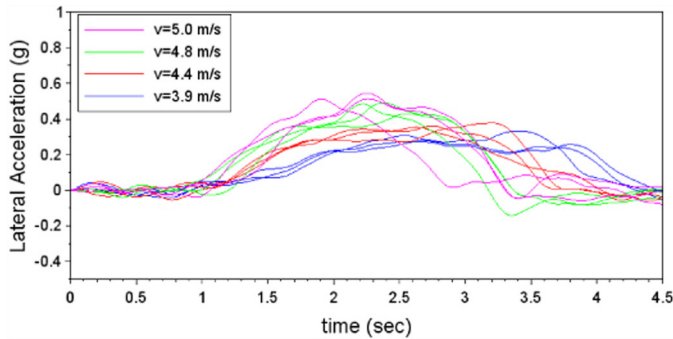


Figure 9. Observed lateral acceleration for constant turn radius with variable entry speed trials.

To evaluate the validity of the simple kinematic representation shown in Figure 6, the average peak acceleration of 3 trials was taken for each test condition, as well as the average (actual) exit corridor distance (w_2). As shown in the figures below, the measured lateral acceleration was indeed found to be proportional to the square of the measured entry speed, and inversely proportional to the exit corridor distance.

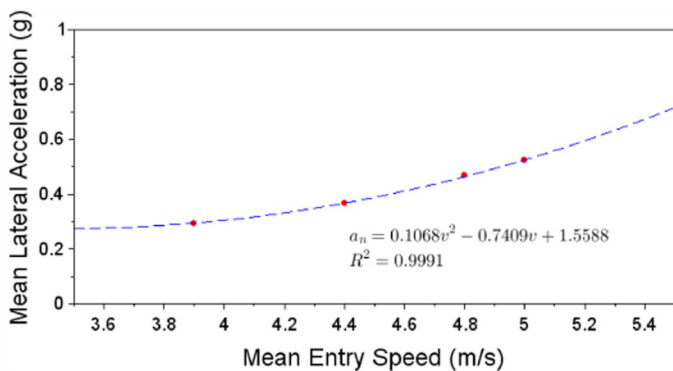


Figure 10. Least squares second order regression of average measured peak lateral acceleration vs. mean entry speed.

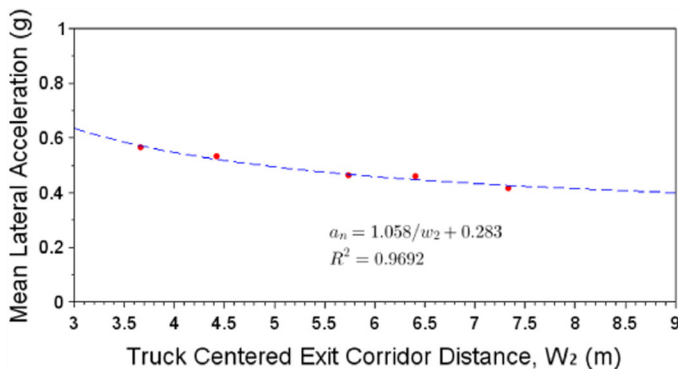


Figure 11. Least squares regression of average measured peak lateral acceleration vs. exit corridor width.

Development of the Dynamic Model

A multi-body dynamic (MBD) simulation was constructed using the commercial software LMS Virtual.Lab Motion [5] to simulate the various test conditions presented in Table 1. The simulation model consisted of discrete rigid bodies constrained to each other using a combination of kinematic joints and dynamic force elements. A summary of the MBD model used for this analysis is presented in Figure 12. The mass and inertia of each discrete body was accounted for by the corresponding CAD geometry (presented in Figure 13) and specified material density. The complete mechanical assembly consisted of the truck body (including safety outriggers), front end equipment, pivoting rear axle and steering mechanism, tires, and a simplified differential assembly.

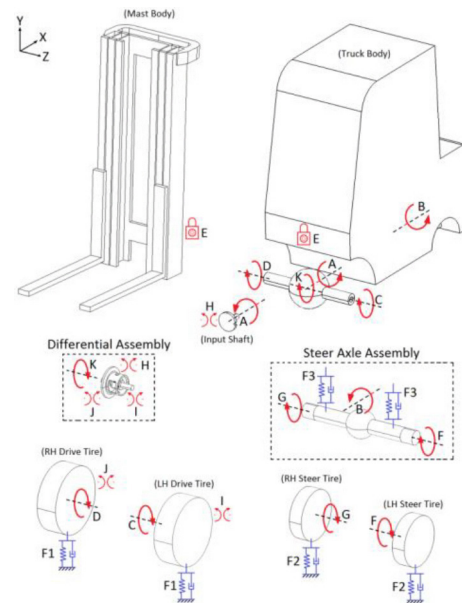


Figure 12. DOF schematic of dynamic model. Connection details are provided in Table 5 of the Appendix.



Figure 13. CAD representation of test truck geometry.