Sensing Proximity to Trailer Rollover: Theoretical and Experimental Analysis

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ABSTRACT

This research is in regards to the development of an inexpensive anti-rollover system for use in the trailer axles of a heavy commercial truck. It includes the design and experimental results for a real-time rollover warning system to inform the driver prior to rollover or to initiate stability control algorithms in the vehicle. The developed system in this research uses suspension displacement, lateral acceleration, and wheel speed measurements. These are used to continuously update the calculation of centre of gravity (CG) height which is compared to the lateral acceleration to determine the proximity of rollover.

INTRODUCTION

This research improves the public safety on interstate highways through the development of an anti-rollover system for heavy commercial trucks. It includes the design of an inexpensive real-time rollover warning system to inform the driver prior to rollover or else initiate stability control algorithms in the vehicle. Heavy commercial vehicles have a greater risk of rollover due to their large size, higher center of gravity and highway width limitations. In addition to this, driver’s awareness of rollover depends on vehicle type, load position and torsional stiffness of the trailer, among many other vehicle parameters. The result is that the rear end of the trailer can act quite independently from the tractor and can initiate rollover before the driver is even aware of his/her imminent crash.

Recent statistics on truck and bus crashes show that rollover occurred in 53.8% of the first harmful event of non-collision type crashes. Furthermore, the fatality rate of rollover accidents was found to be significantly higher than non-rollover cases. There are about 15000 rollovers of commercial trucks every year, rollover accidents are about 4% of the total accidents but when it comes to fatality rate they are higher in rollover when compared to other accidents. Estimated damage costs because of rollover ranges from $91,112 to $289,549 depending on the number of trailer it has, crashes with straight trucks which have no trailers had the lowest cost of $56,296 per crash. The average cost of property damage due to rollover accidents is $15,114. From the above data we can say that rollover accidents are dangerous and cost lot of money for repairs.

Therefore, active counter measures for rollover crashes are crucial for highway safety for trucks (and other vehicles). The developed system in this research uses suspension displacement measurements, a lateral accelerometer, and wheel speed sensors to calculate the centre of gravity (CG) height and the proximity of rollover.

Driver’s awareness of the situation depends on various other factors such as speed, curvature of the road and truck load. Many times drivers misjudge the situation which end up in rollover crashes because every time they unload and load the truck with different type of load it becomes difficult for the driver to predict the exact vehicle response and drive accordingly. They may not be aware that the truck is on the verge of rollover and by the time he/she notices the situation it is too late to control it.

Taking all these factors into account the best method for rollover prevention is to install a warning system on the
truck providing enough notice to take anti-rollover counter measures. The system works on the principal of rollover threshold. The threshold is the lateral acceleration at which rollover occurs and the system monitors the ratio of lateral acceleration to the acceleration for rollover. When the ratio reaches a predetermined amount, a warning is issued.

Further studies by the author’s include a variety of driver warning (haptic feedback) systems. With the application of a reliable sensor and feedback system, an accurate and timely warning is issued regarding the impending rollover threat and the driver can take corrective measures in a timely fashion to avoid rollover accidents.

THEORETICAL VEHICLE MODEL

In order to develop model for rollover prevention we need to take into consideration its dynamic as well as static conditions. The axle spring rate is calculated when the vehicle is in rest position (static) by a simple load displacement test. This is done prior to installation of the rollover threshold program but does use the mobile computer spring displacement sensors. The center of gravity height is calculated when the vehicle is in motion (dynamic).

The dynamic condition is studied for deriving the equations of motion for the vehicle model which include rollover analysis. The model should be able to relate to the rollover conditions with suspension displacement and results in an equation for the center of gravity height. Figure 1 shows a model of a heavy vehicle taking a turn in which there is a change in suspension displacement and shift in the center of gravity position. The nomenclature for the figure is

\[ W = \text{Weight of the vehicle} \]
\[ a_y = \text{Lateral acceleration} \]
\[ T = \text{Track Width} \]
\[ \Delta y = \text{Shift in Center of Gravity relative to track} \]
\[ F_1, F_2 = \text{Reaction at Right & Left Tires} \]
\[ h = \text{Center of Gravity height} \]

The generalized equation of roll moment about a point on the ground at the center of the track is:

\[ W \cdot h \cdot a_y = (F_2 - F_1) \cdot \frac{T}{2} - W \cdot \Delta y \]

Figure 1: Simplified free-body diagram of a heavy vehicle in a steady turn.

In the above equation there is the stabilizing moment due to the vehicle tires, \((F_2 - F_1) \cdot \frac{T}{2}\). There are two destabilizing (overturning) moments acting in response.

- A moment due to the weight of the vehicle acting at a position that is laterally offset from the center of the track, \(W \cdot \Delta y\).
- A moment due to lateral force acting at the center of gravity (CG) height of the vehicle, \(W \cdot h \cdot a_y\) as a result of the external imposition of lateral acceleration.

These two destabilizing moments are opposed by one stabilizing (restoring) moment due to the side-to-side transfer of vertical load on the tires, \((F_2 - F_1) \cdot \frac{T}{2}\). This moment is also due to the internal, compliant responses of the vehicle. The maximum possible value of this moment is \(W \cdot \frac{T}{2}\), which occurs when all load is transferred to one side of the vehicle, i.e., when \(F_2 = W\) and \(F_1 = 0\).

CENTER OF GRAVITY HEIGHT DETERMINATION

While calculating the center of gravity height, the nomenclature of the truck changes and we consider different parameters and viewpoints in our analysis.
\( m = \text{Mass of the vehicle} \)

\( \alpha = \text{Roll angle} \)

\( T_s = \text{Distance between springs} \)

\( T = \text{Track width} \)

\( F_{ZR}, F_{ZL} = \text{Reaction at right & left springs} \)

\( l_3 = \text{Distance from center of axle to center of gravity} \)

\( l_4 = \text{Shift in center of gravity relative to the track length} \)

\( l_2 = \text{Distance from ground to center of the axle} \)

\( h_{cg} = \text{Center of Gravity Height} = l_2 + l_3 \)

Taking a moment around the point on the axle where the left spring makes contact with it gives us the following equation,

\[
F_{ZR} * T_s - m g \left( \frac{t_s}{2} - l_3 * \tan \alpha \right) + m * a_y * l_3 = 0
\]

Here: \( \tan \alpha = \frac{\Delta X_R - \Delta X_L}{T_s} \)

\( \Delta X_R \& \Delta X_L \) are changes in displacement of the right and left spring (which is measured using linear potentiometers mounted in parallel with the spring).

\( F_{ZR} \) is the reaction force at the right spring suspension which is:

\[
F_{ZR} = K_R * \Delta X_R + \frac{mg}{2}
\]

Upon solving:

\[
\left( K_R * \Delta X_R + \frac{mg}{2} \right) * T_s - \frac{m * g * T_s}{2} + m * g * l_3
\]

\[
* \left( \frac{\Delta X_R - \Delta X_L}{T_s} \right) + m * a_y * l_3 = 0
\]

Cancelling:

\[
l_3 m \left( a_y + g \left( \frac{\Delta X_R - \Delta X_L}{T_s} \right) \right)
\]

\[
= \frac{m * g * T_s}{2} - T_s \left( K_R * \Delta X_R + \frac{mg}{2} \right)
\]

Solving for \( l_3 \):

\[
l_3 = \frac{-K_R * \Delta X_R * T_s}{m * a_y + \left[ m * g * \left( \frac{\Delta X_R - \Delta X_L}{T_s} \right) \right]}\]

\[
l_3 = \frac{K_R * \Delta X_R * T_s^2}{m * g \left( \Delta X_L - \Delta X_R - \frac{a_y}{g} * T_s \right)}
\]

Since center of gravity height \( h_{cg} = l_2 + l_3 \),

\( l_2 \) is measured using a measuring tape or caliper during the axle spring rate measurements before the program is loaded (during the initial computer installation).

With these relations in the rollover threshold program, the computer can calculate the Center of Gravity Height continuously. The variability in signals and the errors at very low lateral accelerations require large numbers of calculations and several numerical and temporal filters. However, the result is an updated center of gravity height determination with a 60 second response. Although this seems very slow, the reality is it needs to update only fast enough for a change in load before the first rollover condition. That is, it must calculate a new CG height after loading or unloading of the trailer before entering a highway ramp.
EXPERIMENTAL SETUP

The experimental setup consists of a collection of electronic devices connected to the dual tire axle of a truck. For reasons of application, it is a mobile system that operates on 12 VDC and the majority of components are located inside the cab (see figure 3); however, the suspension displacement devices were located outdoors and only marginally protected from road debris. A permanent system would need to be adapted to a harsh environment.

Figure 3. CDS data acquisition inside vehicle.

The developed system setup is simple and can be used on any trailer (or vehicle). Linear potentiometers are used for the measurements of spring displacements and an accelerometer is used to measure lateral acceleration. For air suspension trailers, an air pressure sensor would also be needed; however, this research was only done on leaf spring suspension systems. The system is powered by the vehicle electrical power and does not require any power converter.

Figure 4. Linear potentiometer attached in parallel to leaf spring.

The experimental setup consists of linear potentiometers (see figure 4), lateral and longitudinal accelerometers (see figure 5), a mobile data acquisition (CDS shown in figure 3), and wheel speed sensor (not shown). The potentiometers are mounted on both left and right suspension springs and are hard wired to the data acquisition system using a data loom. The accelerometer is connected to the data acquisition system and placed at the center position of the vehicle. A Hall Effect sensor was used for calculating wheel speed in the evaluation of dynamic conditions of the vehicle and also for data analysis in order to locate and strip out stationary vehicle readings.

Figure 5. X-Y Accelerometer mounted inside the vehicle.

EXPERIMENTAL TESTS

With the vehicle wired up and the system powered, the experiment starts in a stationary mode by taking spring rate measurements. As can be seen in the stair steps at the right end of the displacement data in figure 7, loads were applied to the suspension and the spring deflection recorded. The loads were then removed and the result was a collection of spring displacements corresponding to changes in axle load.

For experimental convenience reasons, the tests were done on the rear axle of a ‘dually’ pickup truck (a Chevrolet 3500 crew cab). The rear suspension is identical to most heavy commercial vehicles with the main differences being this being a driven beam axle, the size is smaller, and uses leaf springs whereas many heavy commercial vehicles are using air bags.

DATA ANALYSIS

Consider the following data for a pair of experimental runs. In review of the lateral acceleration data, it can be seen that in each of the two sets, there is a strong left turn followed by a strong right turn. In reality, these were sweeps around a skid pad counter-clockwise and then reversed running clockwise. At the center of the data, the vehicle is brought to a stop and then returned to the skid pad for a second set.
Figure 6. Lateral acceleration versus time.

Figure 7 shows the spring displacements for the same experimental runs. Notice that during the left hand turns, the left side spring drops and the right side spring is compressed upwards (with respect to the vehicle). In addition, the course included two large “Speed Bumps” which are easily seen on the displacement plots leading up to the skid pad and returning on both sets of runs.

Using the calculations described in the theoretical vehicle dynamics model above, a continuous calculation can be made of the Center of Gravity Height. It was found that this calculation is highly sensitive to the lateral acceleration values and any noise in the data. For example, as the lateral acceleration approaches zero, the calculated center of gravity height would experience an unreasonable amount of variability.

To handle this issue, the data was filtered in a number of ways that resulted in an effective 0.014 Hz filter. This meant that the center of gravity height could only be updated every 70 seconds or so. Despite this slow response, a new center of gravity height in a real world application is only needed after a load change and prior to the first hard turn.

It was determined that an update rate of 70 seconds would be sufficient for calculating the center of gravity height because of two things:

- The expectation is that the most frequently a heavy truck is loaded and unloaded is once every hour or so.
- The fastest a truck would leave the loading dock and reach the on ramp onto a highway is on the order of minutes.

The result is that the fastest the system would need to update after a load change is on the order of minutes and by this time, the computer would have already obtained a new CG height.

Figure 8. Filtered CG Height estimation is around 60”.

Using the data from the two runs an analysis was made of the center of gravity height. It required several filtering schemes including data averaging. It also employed a requirement for volume of data within a reasonable range. That is, if the vehicle did not reach a threshold lateral acceleration for a period of time, then there was no updated estimation of CG height. This is why the estimated value around point 6400 returns to zero.

Figure 9. Calculation of proximity to rollover with the second set of data.
Figure 9 uses the individual lateral acceleration values in a comparison with the vehicle roll angle to determine proximity of rollover. The vehicle roll angle is estimated from the mechanical displacements of the spring on the suspension.

APPLICATION OF SYSTEM

The value of this system is that it can provide a real time analysis of a truck approaching rollover. The application of this is a warning system for a driver or automatic stability control system. As an example, consider the scenario where a heavy truck spends 99.99% of its time at very low lateral acceleration values. Even going around turns tends to be at very low lateral acceleration values. Even on the high risk entrance ramps onto a highway, trucks require several seconds to reach full cornering speeds.

What this system provides is a warning mechanism during these high risk events and in a quick enough way to help the driver avoid rollover. In figure 10, the same plot as in figure 9 is made; however, at 50% of the rollover condition an alarm is triggered. These values are shown in red.

CONCLUSION

This research is in regards to the development of an inexpensive anti-rollover system for use in the trailer axles of a heavy commercial truck. It includes the design and experimental results for a real-time rollover warning system to inform the driver prior to rollover or to initiate stability control algorithms in the vehicle. The developed system in this research uses suspension displacement, lateral acceleration, and wheel speed measurements. These are used to continuously update the calculation of centre of gravity (CG) height which is compared to the lateral acceleration to determine the proximity of rollover.

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