

Cost Effective Rollover Mitigation Strategy

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(ABSTRACT)

A cost effective method of rollover mitigation in vehicles is presented. The method was designed so that some of the system states were measured by sensors that are already available on most vehicles and so that other states could be measured with relatively low cost sensors. Also, the control algorithm was designed to be implementable using a series of look up tables and computationally efficient equations to enable the use of low-cost controller platforms. These look up tables and equations can be modified to change the conservativeness of the method as well as to configure the method for use on almost any 4-wheeled vehicle. Lastly, the proposed mitigation technique was designed to be directly implementable with existing vehicle hardware.

To develop this method, a vehicle model was created using several advanced computer packages including SolidWorks 2008™, MATLAB ®, Simulink ®, and SimMechanics™. Once created, the model was outfitted with virtual sensors that represent data from realistic sensor types. A detection algorithm was designed around the hypothesis of a stability boundary utilizing the sensor data to detect impending rollover. Finally, a mitigation algorithm was designed to limit throttle and braking upon impending rollover. This algorithm was defined using the basic principles of end-stop control, but was adapted to work appropriately with this scenario. To conclude this research, two simple maneuvers were used to verify the effectiveness of this system to mitigate vehicle rollover.

This research was government sponsored and in some instances utilized secured data. Due to the nature of this material, some data has been omitted from this document.

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Chapter 1

Introduction

With the substantial presence of vehicles with high centers of gravity and an increasing emphasis on vehicle safety, there is great need for cost effective methods of rollover mitigation. This is seen in the commercial market with trucks and sport utility vehicles where manufacturers are being held to increasingly higher standards of safety. This is also seen in the military market where vehicles are being equipped with more armor and are being forced to operate at higher ground clearances for sub-terranean mine protection. These systems must be able to accurately detect impending rollover using realistic and cost effective sensors and controllers. Also, they must detect this impending rollover soon enough to activate mitigation hardware without misinterpreting safe vehicle maneuvers as rollover situations.

This work begins with an analysis of vehicle mitigation systems. These systems can be separated into three subsystems including detection sensing, detection and mitigation algorithms, and mitigation hardware. There are many options for fulfillment of each of these subsystems and in some cases these options can be interchangeable. Some of these options exist as systems designed to perform tasks other than rollover mitigation but can be easily adapted to incorporate this field. Several options are discussed in this work and one, a stability boundary based algorithm using end-stop control of the vehicle throttle, is presented. An overview of this system can be seen in Figure 1.1.

Due to the nature of these rollover events, it is important to be able to accurately simulate rollover maneuvers within a safe and controlled setting. This setting needs to be both accurate and repeatable, thus making the world of computerized vehicle modeling an obvious choice [1]. Computer models range in complexity and accuracy and it is important to determine an

appropriate level of detail needed for development. Two different vehicle models of differing complexity were developed and the tradeoffs involved with each have been analyzed and are discussed in this work.

A series of vehicle maneuvers have been developed to manipulate the vehicle models such that rollover can be studied. These models include a quasi-static test, called the constant radius maneuver, as well as a dynamic test, called the Fishhook 1B, that has been defined by the NHTSA as the most effective maneuver for vehicle rollover [2].

It was hypothesized that a boundary exists in the space of a number of key vehicle states that represents rollover stability. A surface was developed in the space of vehicle velocity, steering angle, and ground plane angle that verifies this hypothesis. An algorithm was developed to interpret the potential for vehicle rollover at any given vehicle state relative to this boundary.

Another algorithm was developed using modified end-stop control to create a control signal based on the findings of the stability boundary interpreter. This control signal was utilized in a throttle controller to prevent vehicle rollover.

After the mitigation strategy had been created, it was implemented into the intermediate vehicle model and was used to perform a series of tests. The first test verified the conservatism of the stability boundary in quasi-static situations. The second test verified the effectiveness of the system in dynamic situations.

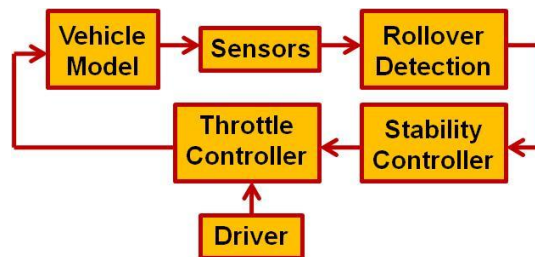


Figure 1.1. Top Level Diagram of Model Used to Detect and Mitigate Vehicle Rollover

Chapter 2

Literature Review

The topic of rollover mitigation has been widely researched in recent years; however the findings and systems that have been developed often remain proprietary to the automotive manufacturers or parts distributors that funded the research. Although documentation of these complete systems is not readily available, research on individual components including sensors, strategies and mitigation hardware is available. The sections within this chapter will describe the available knowledge on each of these topics.

Beyond the private sector, the US Department of Transportation has sponsored a large number of research projects on vehicle rollover and has subsequently published standards for testing rollover mitigation systems. One of the methods for testing the effectiveness of these systems is the NHTSA Fishhook 1B maneuver, defined by Forkenbrock [2]. This maneuver is defined by the required driver inputs (steering and throttle) and can be seen in Figure 2.1. The premise of this maneuver is to utilize the release of potential energy stored in the vehicle suspension along with the inertial forces associated with cornering to successfully induce enough vehicle body roll to produce two-wheel liftoff which is often indicative of impending vehicle rollover. This is the primary maneuver used in this study to determine the effectiveness of the rollover mitigation scheme that has been developed. Further details are discussed in subsequent sections.

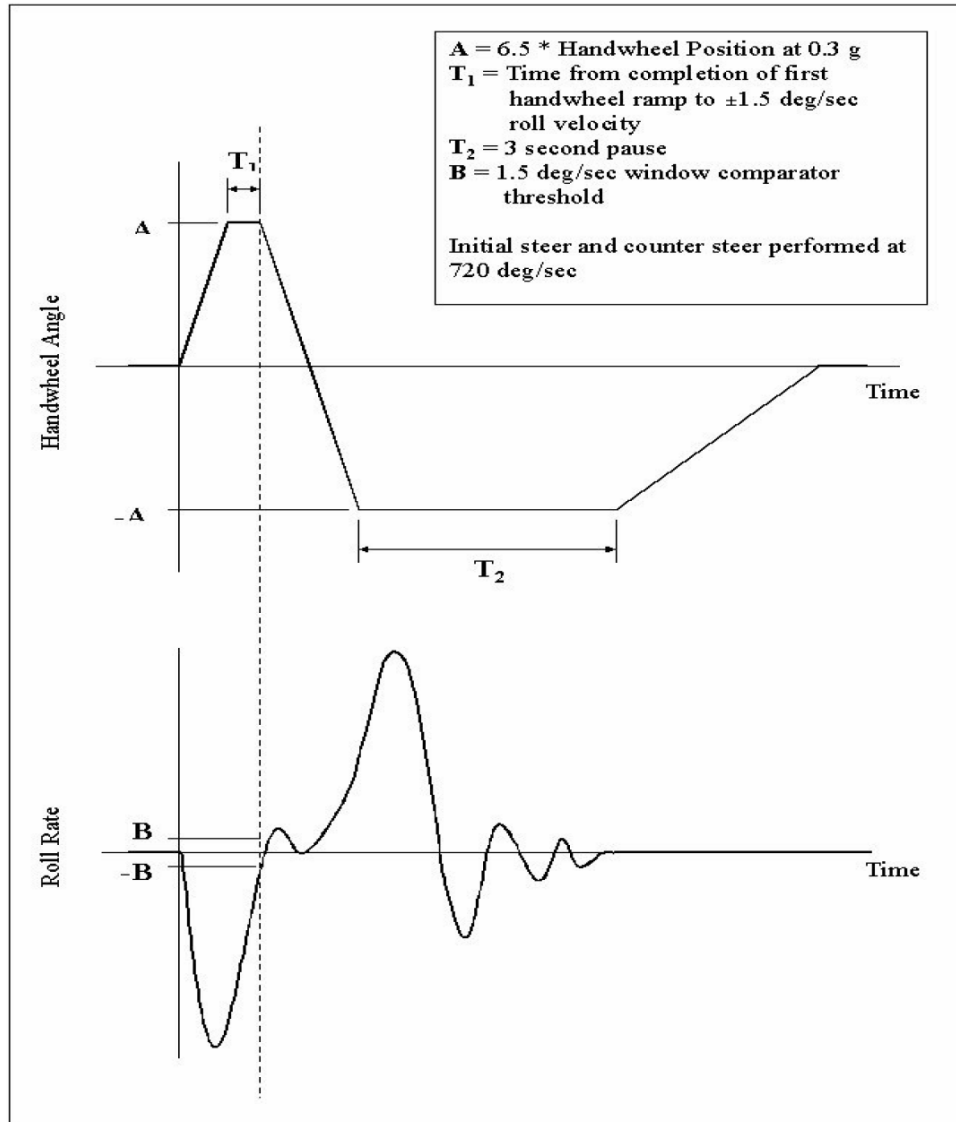


Figure 2.1. Fish Hook Maneuver as Defined by the National Highway Traffic Safety Administration [2]

2.1 Existing Rollover Detection Sensors

One key aspect of rollover mitigation is accurate detection and prediction of impending rollover. To detect this impending rollover, several sensors and combinations of sensors using simple to complex methodologies have been studied to produce accurate and reliable measurements.

Many systems require knowledge of vehicle body roll and/or vehicle roll rate. One of the simplest methods to measure these states uses a pendulum or pendulum-like object to determine the direction of the gravitational acceleration vector. This can then be compared to the vehicle body normal to determine the vehicle roll angle. This roll angle measurement can then be

utilized to determine roll rate if such a measurement is required. One such device can be seen in the patent by Cash [3]. These devices often fail to differentiate gravitational acceleration from inertial acceleration found during cornering. This lack of discrimination can be the cause for an incorrect assessment of the threat of impending rollover during high speed turns. Also, the rotational inertia of the device components imposes a natural damping and limits the measurement bandwidth of the system. Due to their simplistic nature, these devices can be produced at a relatively low cost but at the expense of accuracy and reliability.

Another means of measuring roll angle or roll rate is through the use of one or more accelerometers. There has been extensive research to determine how effectively these sensors can produce the desired measurements within a vehicle setting. Most often, these sensors are used to determine the vehicle roll angle with respect to the earth's gravitational field directly. Aleksander Hac [4] has researched the use of lateral accelerometers for this purpose and some of his findings can be seen in Figure 2.2. In this figure it can be seen that lateral acceleration data can be used to accurately estimate low roll angles on smooth roads but is less accurate at measuring relatively large angles on uneven terrain. Accelerometers can be relatively inexpensive in comparison to other roll measuring sensors; however they are unable to accurately measure roll throughout all vehicle operating ranges.

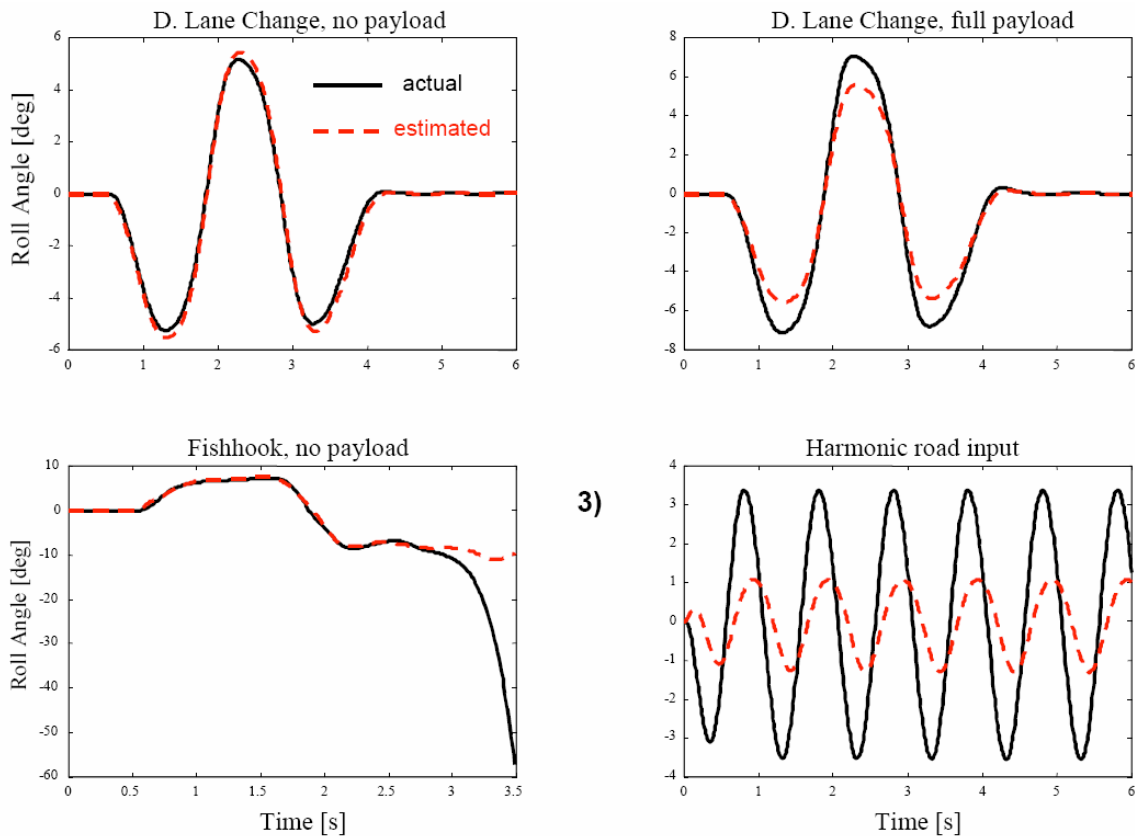


Figure 2.2. Actual Vehicle Roll versus Vehicle Roll Estimated by Lateral Acceleration in Simulation [4]

A third method for calculating body roll and roll rate utilizes a rate gyrometer. These devices are capable of accurately measuring roll rate, which can be integrated to determine roll angle. However, Hac's [4] research has also shown that pure integration of this roll rate signal with bias in either direction will cause the calculation to drift and drastically effect the error of the roll angle estimation as can be seen in Figure 2.3. As can be seen, the sensor is able to give reasonable approximations of roll during transient maneuvers; however the presence of drift in steady state maneuvers causes high levels of error. Rate gyrometers are relatively inexpensive, although they are typically more expensive than accelerometers. Like accelerometers, they are available with varying levels of accuracy. Higher accuracy sensors typically show less signs of drift; however they are typically more expensive.

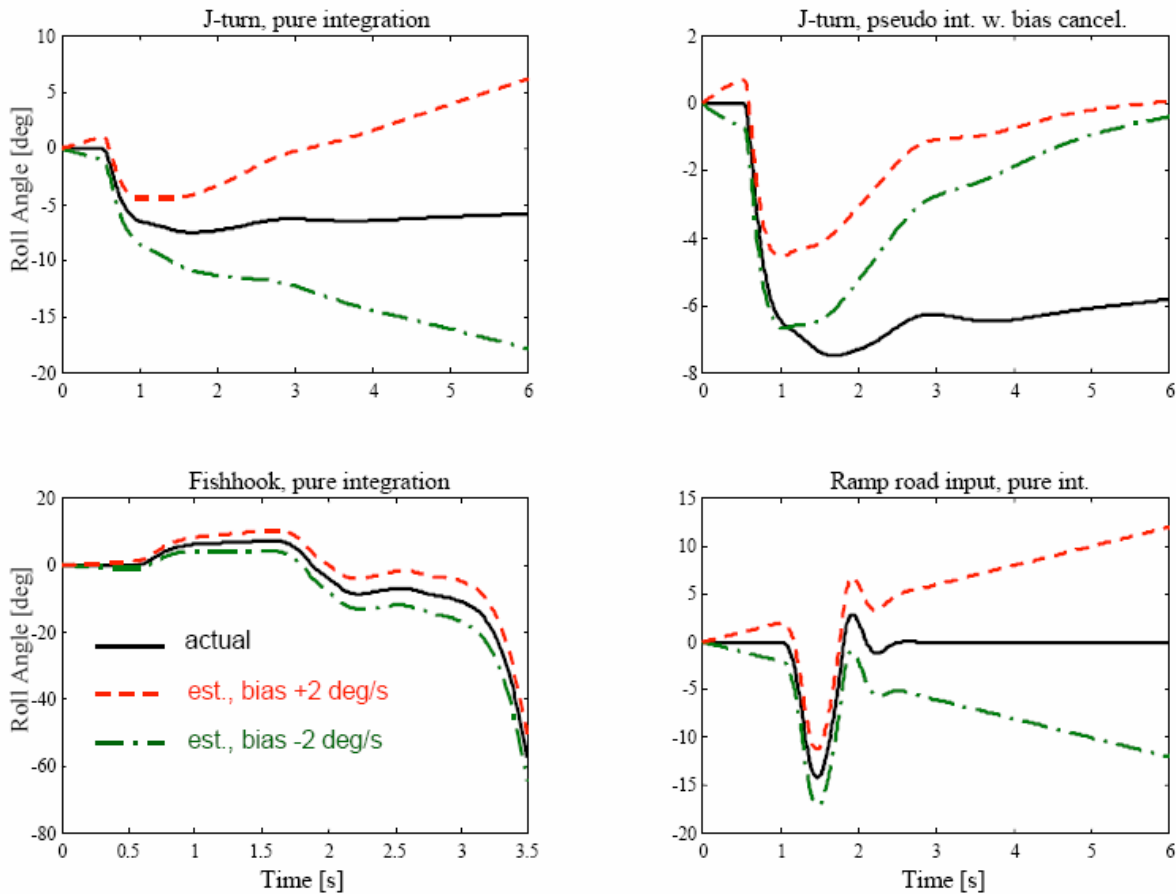


Figure 2.3. Actual Vehicle Roll versus Vehicle Roll Estimated by Roll Rate with Positive and Negative Bias in Simulation [4]

Beyond bias and integration issues, rate gyros are also commonly considered to be delicate sensors that are greatly affected by their environment. Henry Kong and his research team [5; 6; 7] have noted that it is important to consider bandwidth, noise measurement, and sensitivity to linear acceleration when selecting a rate gyro for use in a vehicular setting. While these considerations may positively impact the reliability of the measurement, they may also negatively impact the cost of the system.

More complex mitigation strategies often require additional vehicle states to be measured such as lateral velocity, side slip angle, and tire slip angle. These measurements are often determined through the use of multiple sensors working together, as seen in the works done by Beiker [8] and Willig [9]. Beiker's work utilizes the combination of GPS location data to reduce drift of an inertial measurement unit, or IMU. The IMU is capable of providing accurate information about all six degrees of freedom of a vehicle body, however due to integration errors they require additional inputs to maintain their accuracy. The concept of utilizing data from multiple sensors to increase fidelity in measurements is not uncommon. Due to the nature of the needed measurements of these systems, they are typically expensive relative to other approaches. They also require complex algorithms which can be more computationally demanding and require more expensive controller hardware. Although these systems come at a premium cost, they are able to produce accurate and reliable data to complex algorithms.

2.2 Existing Rollover Detection and Mitigation Algorithms

The next step to mitigating rollover is determining the threat of rollover based on the measured input(s) of the sensor(s). These algorithms, like the sensors, range from relatively simple to complex in nature and provide varying levels of fidelity.

Several common methods for detecting rollover use knowledge of vehicle roll angle. One such system, portrayed in a paper by Sorniotti [10], can be imagined by considering the simple vehicle model described in the next chapter. In this model, rollover is viewed in terms of opposing body roll moments. Lateral forces, caused by a combination of inertial and dynamic accelerations, act on a moment arm from the ground to the height of the center of gravity causing a rollover moment. This rollover moment is then opposed by a restoring moment. The restoring moment is naturally created by gravitational forces acting on a moment arm with components of the vehicle's track width and CG height depending on the ground plane angle. As the ratio of rollover moment vs restoring moment approaches 1, a control signal is systematically increased. This control signal is used to increase the restoring moment through the use of some hardware, like Sorniotti's active anti-roll bars [10]. While the simplistic calculations needed for this method can be implemented relatively easily and at a relatively low cost, it is shown in the next chapter that this model is incapable of accurately discriminating vehicle rollover in all situations.

Some other common approaches utilize multiple sensors and complex logic to allow complementary sensors to overcome the shortcomings of some measurement types. One example is a patent by Schubert [11] where lateral acceleration is used as an arming function to the vertical acceleration based system. Both of these sensors can be used separately as mentioned in the previous section, however neither is capable of accurately detecting rollover throughout the entire range of possible vehicle motion. Logic can be used to define the relationship of these sensors when the threat of rollover is present throughout the entire range of vehicle motion in an effort to reduce the range of inaccurate detection. To further reduce this risk, additional sensors can be integrated into the strategy. This type of algorithm allows for

sensors with shortcomings to be used, however it requires the use of multiple sensors. These sensors may be relatively inexpensive; however the number of required sensors and the complexity of the algorithm can make these systems potentially expensive to achieve high fidelity and reliability.

Similar to using multiple sensors to improve fidelity, a patent by Schniffmann [12] uses multiple algorithms to analyze and overcome the shortcomings of a single sensor's measurement. This patent combines the use of long term integration that is reset when zero roll is detected, short term integration, and advanced filtering techniques to reduce the effects of bias in roll rate sensors. This system lowers the number of needed sensors and increases the reliability and accuracy of proper detection, thus providing an accurate system at a relatively low cost.

Another algorithm that can utilize multiple sensors is based on the Lyapunov direct method. This method is designed specifically to push control systems toward stability based on a known set of stable states. Tamaddoni [13; 14] and Hopkins [15] have shown how this method can be adapted to include state estimation and to control vehicle yaw rate. Yaw is inherently coupled to roll, therefore controlling yaw will also affect roll. The proposed system also creates control signals for active steering control as a backup to the differential braking system. This design is effective at preventing vehicle roll and can minimize the number of needed sensors because of state estimation. However, this system has been designed to utilize multiple methods of mitigation which promotes an increase in the overall system cost.

Other potential algorithms already exist as complete systems that were designed to complete tasks other than rollover detection or mitigation. One example is Ahmadian's [16] use of Skyhook control, which was originally designed to control semi-active dampers in automotive suspension systems and truck seats to improve ride quality. This system was modified to accept steering angle as an input. The system utilized this knowledge to logically control damper forces to reduce body roll due to suspension deflection. This system was shown to be effective but required multiple sensors to determine steering angle and suspension deflections.

Similarly, the patented control algorithm developed by Southward [17], St. Clair [18], and Miller [19], called end-stop control, was the base algorithm used in this research. End-stop control, like Skyhook, was originally designed to increase ride quality in vehicles equipped with semi-active dampers. This system, as its name might imply, restricts a system from reaching its extremes, or end-stops. This technology and the need to adapt a new form of it will be discussed in Chapter 6. As will be shown, a system can be created using a set of relatively inexpensive sensors and this moderately simplistic end-stop control algorithm to effectively detect impending vehicle rollover.

2.3 Existing Rollover Mitigation Hardware

The last component of the rollover mitigation process is the use of hardware to actually prevent the vehicle from rolling over. Due to the complex nature of vehicles, there exist many possible

ways to prevent rollover that range from simply reducing throttle or applying brakes to changing damping or sway bar properties.

One of the most common approaches to rollover mitigation is the use of differential braking. This means that the vehicle is capable of producing different braking torques at different wheels or sets of wheels. This technology is capable of being used for many different purposes; the most common is anti-lock braking where braking torque at each wheel is controlled to prevent lock-up. As seen in Palvonics' [20] work, this idea is especially important for vehicles that are segmented or vehicles towing trailers. These systems are designed to prevent jack-knifing and trailer whip, which, along with rollover, represent the most dangerous motions for these types of vehicles. While these systems have been designed for use in a wide number of applications and have become more cost effective, they are not the most effective systems for rollover mitigation in all vehicles.

Wielenga [21] and Ding [22] have shown that reversing this concept presents one of the most common forms of rollover mitigation strategies where reduced friction and potential tire lock-up are desired. The concept of this system can be seen in Figure 2.4. In this figure, a vehicle is shown in a cornering situation where inertial forces are acting on the vehicle's CG towards the outside of the turn. This inertial force is responsible for creating the vehicle roll moment that is shown. Considering the simple model that is further discussed in the next chapter, where rollover is viewed as opposing moments associated with inertial and gravitational acceleration, it becomes apparent that reducing the inertial acceleration would reduce the rollover moment. This inertial acceleration is directly associated with the vehicle longitudinal velocity and yaw rate, therefore reducing one of these states, or in this case both, will in turn reduce the inertial acceleration. To create this reduction, the front tire that is found on the outside of the turn has a braking torque applied. This torque then changes the force vector associated with the tire in an effort to slow the vehicle. The adjusted braking vector can be seen in the figure. If this force vector exceeds the maximum tractive force of the tire, then the tire will experience lock-up and reduce both the lateral and longitudinal reaction forces thus reducing longitudinal velocity and yaw rate. This system, while effective at mitigating rollover, reduces the vehicle's ability to turn and can in effect limit the maneuverability of the vehicle. This over-ride of driver control can be perceived as less than desirable for military applications.

Another means of rollover mitigation is to actively adjust a suspension component that is specifically designed to reduce body roll, the anti-roll bar. In most vehicles the anti-roll bar is a torsion spring that is connected to the suspension system on both sides of the vehicle. Whenever uneven suspension displacements are encountered, the bar produces a reaction force to reduce this. The stiffness of this bar has significant impacts on the vehicle's roll characteristics and ride quality. For passive systems, a balance must be found between these two characteristics, where an active system is capable of providing roll rigidity while simultaneously maintaining ride quality. Weser [23] and Everett [24] have performed extensive research to show the benefit and effectiveness of these systems. Though effective at mitigating rollover, this technique is not the

most cost effective system as it requires additional hardware and controllers to be added to the vehicle.

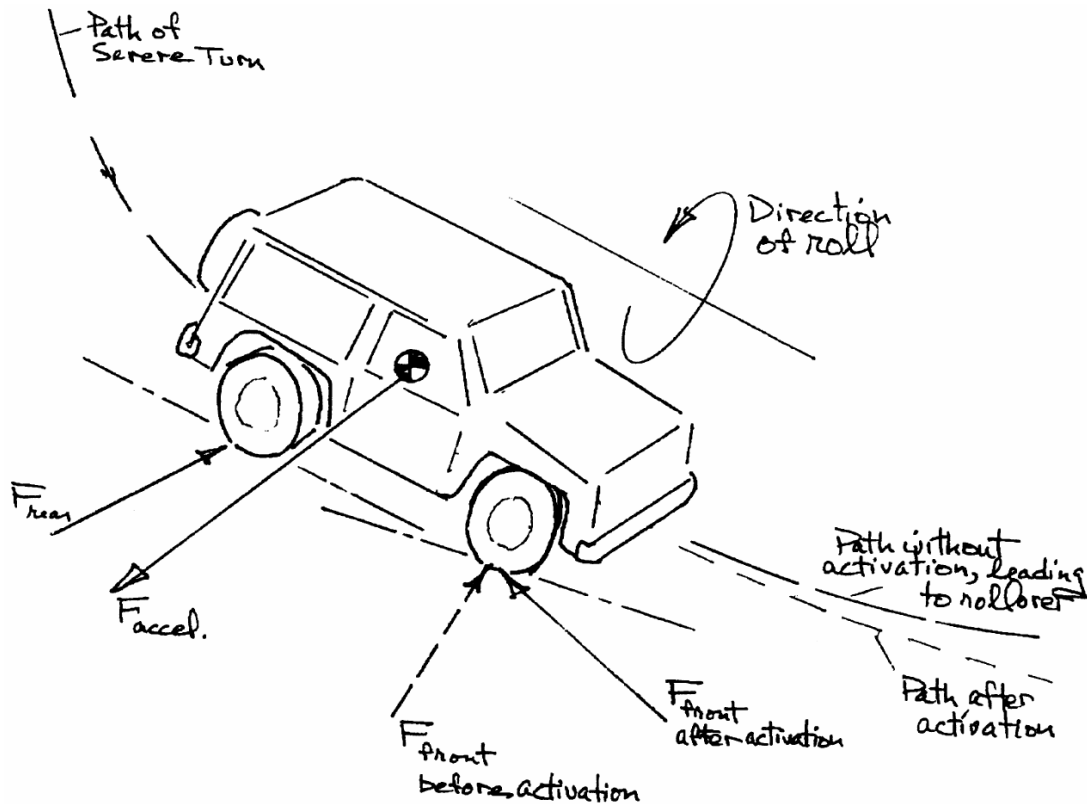


Figure 2.4. Forces on Vehicle During Hard Cornering with Differential Braking [21]

A fourth means of rollover mitigation comes in the form of active or semi-active suspension components as shown by Ahmadian [16] and Ikenaga [25]. Ahmadian's system uses Skyhook control of Magneto-Rheological Dampers. These dampers are similar to conventional dampers but are filled with magneto-rheological fluid, which can change viscosity in the presence of an applied charge. By adjusting the viscosity of the fluid in a damper, damping characteristics are greatly affected. By controlling this effect, vehicle roll can be directly influenced and as Ahmadian's research has shown, this influence is effective at reducing body roll. However, as is the case with any system that requires additional hardware, these systems can be relatively expensive to incorporate.

Another concept to consider is the use of multiple hardware systems working together. Odenthal [26] developed a system that utilized active steering control and braking control to keep a vehicle within its defined rollover stability bounds. The advantage of a system like this is the ability to tune the controller to offset the shortcomings of each system. In this case, steering control forces a vehicle to deviate from its desired course while braking control reduces velocity. The system can be adjusted to optimally serve any type of vehicle. As previously mentioned, adding additional hardware to a system is expensive and the more systems added increase both the cost and complexity.

Chapter 3

Simulated Vehicle Model Development

There are many types of vehicle models that range in complexity, each have relative pros and cons. For this research, three levels of complexity were considered and two different models were eventually created. The first model was the rigid body model and is the simplest in complexity compared to the other models. Many of the underlying concepts of vehicle rollover can be learned from this simple model, including the idea of rollover and restoring moments as well as the relationship of lateral force and longitudinal velocity and yaw rate. The second model included three-dimensional dynamics with rigid fixed tires. It was determined that this model was sufficient for the development purposes of this paper. The last type of model is the highly complex model, which can incorporate full vehicle dynamics and complex tire models. This level of complexity can be used to fine tune and adjust the final algorithm, however based on the limited knowledge of the vehicle and the scope of this research it was not deemed necessary to develop a model of this complexity. The simple and intermediate models are described in more detail in this chapter.

The majority of the modeling emphasis was placed on the intermediate model that was used to develop the mitigation system in this research. Developing this vehicle model was a multi-step process involving several software suites including MATLAB®, Simulink®, SimMechanics™, and SolidWorks 2008™. The development started with a simple CAD model to capture the geometry and kinematics of the major vehicle components. This model was then imported into SimMechanics™ as well as the Simulink Virtual Reality™ toolbox. The model was then outfitted with dynamic subsystems such as springs and dampers. The model was also outfitted with sensors and linked to the virtual model for visualization purposes. The MATLAB ® /

Simulink ® environment provided a means to develop the strategy and the Virtual Reality ™ toolbox provided a means to view the system for qualitative verification purposes. Because the geometry, kinematics, and dynamics were all included in the CAD and SimMechanics models, there was no need to develop the governing equations or equations of motion for the model.

3.1 Rigid Body Model

The rigid body model was developed as a means to produce simple analytic solutions. Model accuracy was intentionally sacrificed for simplicity. While this model produced reasonable results, it was determined, as Renfro [27] has shown, that this model is not appropriate for detailed mitigation control algorithm development.

This model is based on Gillespie's [28] simple model, where the vehicle body, suspension, and tires being a single rigid body. This body is affected by forces as seen in Figure 3.1. In this figure $M a_y$ represents inertial forces from cornering, $M g$ represents the weight of the vehicle as a result of gravity, F_{zi} and F_{zo} represent the vertical reaction forces at the tires, F_{yi} represents the lateral force applied to the outer tire, and h represents the height of the center of gravity above the ground.

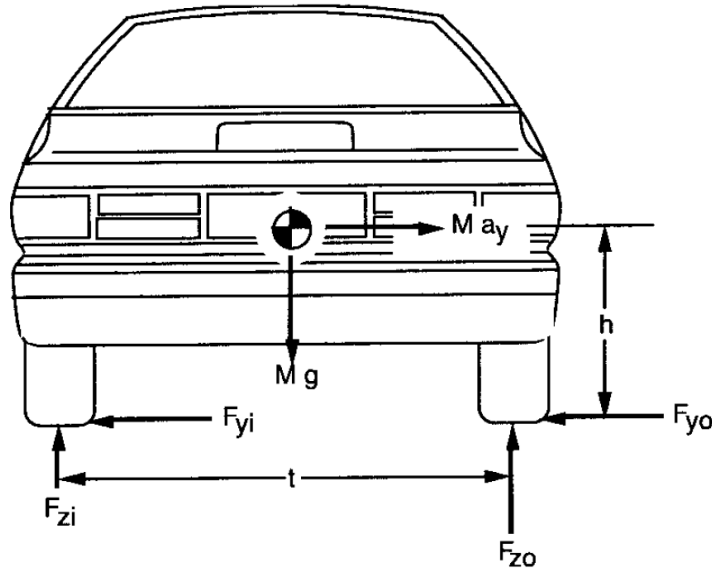


Figure 3.1. Rigid Body Model with Appropriate Reaction Forces [28]

$$\text{Static Stability Factor: } \frac{t}{2h} \quad (3.1)$$

$$M_{\text{Rollover}} = M a_y h \quad (3.2)$$

$$M_{\text{Anti-Rollover}} = M g \cos(\varphi) \left(\frac{t}{2}\right) \quad (3.3)$$

$$\text{Stability Ratio: } \frac{M_{\text{Anti-Rollover}}}{M_{\text{Rollover}}} = \frac{g \cos(\varphi) t}{2 a_y h} > 1 \cdot F_{\text{safety}} \quad (3.4)$$

One of the most common methods of determining a vehicle's stability using the simple model is the static stability factor. This value is convenient because it only requires knowledge of two vehicle parameters. However it does not include any vehicle dynamics and is very conservative. Systems using this value to predict the rollover threshold often place the threshold much lower than it really is [28].

A further simple analysis of this model rotating about the outer tire results in two opposing moments. The first is caused by the inertial force and is referred to as the rollover moment. The second is caused by the gravitational force and is known as the restoring moment or anti-rollover moment. Vehicle rollover is expected when the rollover moment exceeds the anti-rollover moment. For conservative design, a factor of safety can be easily included. The allowable inertial acceleration changes as the ground plane angle changes. Equations 3.2 to 3.4 are used to determine these moments.

3.2 CAD Model

Based on limited access to actual vehicle specifications, a very simple model was developed to represent a military type vehicle. The vehicle body was modeled as a rigid solid with appropriate located CG, wheelbase and track width values. The suspension was modeled as a double parallel a-arm to simplify geometric constraints such as camber and caster change. The system does appropriately model track width changes and spring/damper deflections. The model also included appropriate suspension and tire masses and inertias. The front and rear suspension components are identical to simplify modeling purposes. The suspension model can be seen in Figure 3.2. The suspension was modeled in full rebound to define zero suspension deflection as full rebound. The suspension assembly was then added to the main vehicle body and wheels and tires were added. The final CAD representation can be seen in Figure 3.3.

Next, a free translator offered by The Mathworks™ was used to save the CAD model from SolidWorks™ as an .xml file that can be read by SimMechanics™ [29]. The model was finally saved as a VRML (.wrl) so as to be imported into the Virtual Reality™ toolbox.

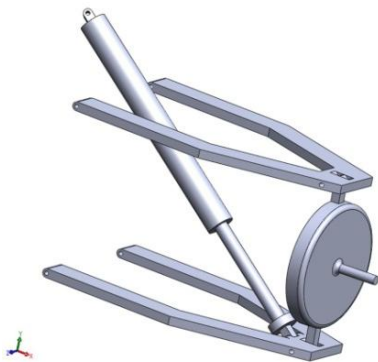


Figure 3.2. CAD Model of Vehicle Suspension

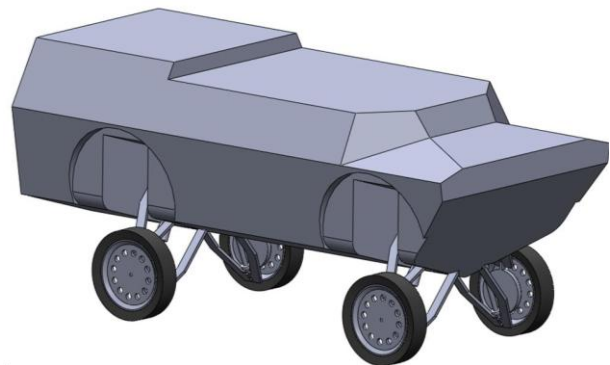


Figure 3.3. CAD Model Representative of a Military Vehicle

3.3 SimMechanics Model

Using the .xlm file from SolidWorks™, a rough vehicle model was very easily created. The .xml file defined all of the bodies of the model and defined their masses and inertias as well as CG locations and appropriate joint types.

Next, the model was divided into 5 main components; 4 suspension subassemblies and the main vehicle body. Within the suspension subassemblies, a block was added to incorporate spring and damping properties to the strut. The block can be seen in Figure 3.4.

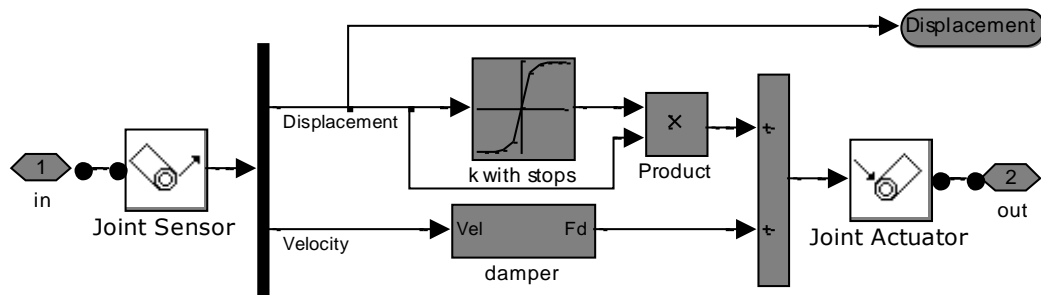


Figure 3.4. Spring and Damper Model

The block starts with an input from a joint sensor, attached to the prismatic joint in the strut. This sensor delivers data on the displacement and velocity of this joint. The displacement is utilized for the spring calculations and the velocity is used for the damping calculations. The spring is modeled as a linear spring with non-linear trends at the defined “end-stops” of the suspension’s travel. These end-stops represent the physical bump and rebound stops found in a real suspension system. They are modeled as extreme changes in spring rate and easily modeled using a look-up table. Once a spring rate is determined, it is multiplied by the displacement to determine the spring force. The damper is modeled as a simple linear damper. Therefore, the velocity is simply amplified by a constant to determine the damping force. Then, the two forces are added and sent to the Joint Actuator.

The next addition to the model was a set of joints to allow the vehicle body to be manipulated into motion. This block can be seen in Figure 3.5. The block starts with a connection to the ground at the world origin. From this origin, the body is allowed to translate in the x-, y-, and z-directions freely. Because all of the joints are separated, they have to be connected by “massless” bodies. Next, the vehicle is yawed in a controlled manner based on the vehicle’s defined heading. The heading is an output of the maneuver profile blocks that are discussed in the next chapter. Next the body is allowed to pitch and roll freely. The second connection connects this block to the CG of the vehicle body. The last set of blocks defines the vehicle’s position in the world. The position input is also defined by the maneuver profile blocks. The signals for x- and y-position are then connected to drivers that are connected to the vehicle’s CG. Thus, using inputs of heading and position, the vehicle can be driven.

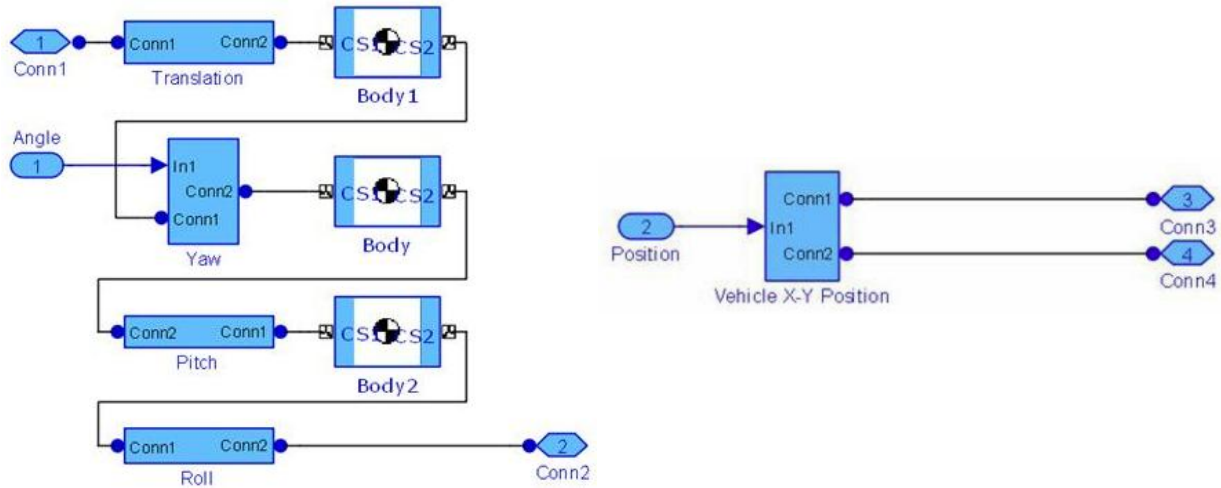


Figure 3.5. Vehicle Body Driver Block

3.4 Virtual Representation

The next modification to the vehicle model was to outfit each body with translation and rotation sensors. The signals of these sensors were then all funneled into a virtual reality block.

The virtual reality block utilized the VRML files of the vehicle and a skid pad that were previously created in SolidWorks™. Using another software package offered with MATLAB®, called V-Realm Builder 2.0©, each body of the vehicle model was modified to look for inputs of translation and rotation. Also, the skid pad was added and the overall file was saved as a world file (.wrl). This file was then used within a VR Sink block from the Virtual Reality™ toolbox. A view of the final VR simulation can be seen in Figure 3.6.

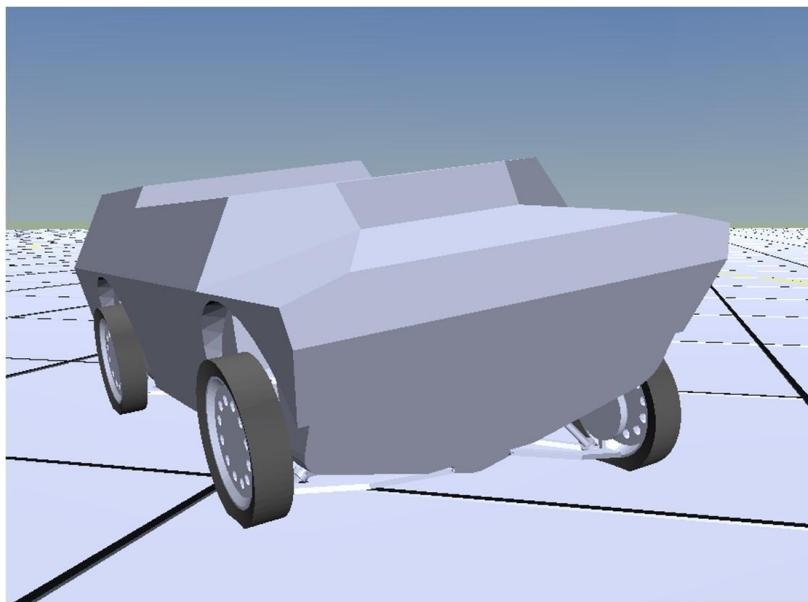


Figure 3.6. Vehicle Model Driving in Virtual World

Chapter 4

Vehicle Maneuver Profile Development

After the model was developed, a series of maneuver profiles were developed to maneuver the vehicle appropriately for the desired tests. These maneuvers included a constant radius maneuver for development of the stability boundary, a fish hook maneuver for tuning and testing the vehicle and control system, and finally a random input to fully test the capabilities of the model and controller in much more chaotic and realistic situations.

4.1 Constant Radius

The first profile developed was the constant radius maneuver. This maneuver was used in the development of the original stability boundary, which is presented in subsequent chapters. The maneuver involves the vehicle driving in a circle of a predefined constant radius and constant acceleration until the vehicle experiences rollover. The effects of ground plane angle can also be explored by changing the terrain from a flat surface to a cone or bowl. An example of the terrain can be seen in Figure 4.1.

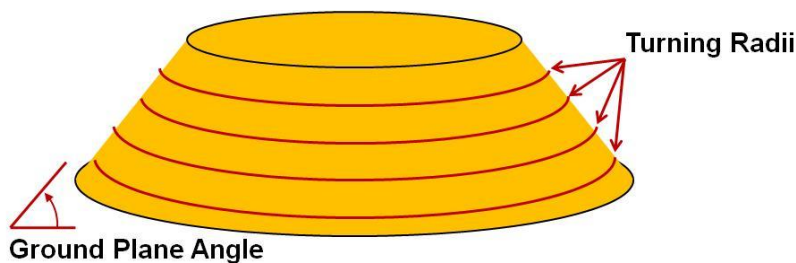


Figure 4.1. Constant Radius Maneuver

The original concept of the stability boundary was created using this maneuver. A series of runs at several defined turning radii and ground plane angles were performed and the maximum allowable speeds at each were recorded. Later a three dimensional surface of these data points was compiled and created the early concept of the stability boundary.

4.2 Fish Hook Maneuver

The next maneuver that was considered was the fish hook maneuver, based on NHTSA Fishhook 1B [2]. This maneuver is designed to instigate vehicle roll over and is commonly used for these types of tests. The maneuver starts with the vehicle steadily accelerating until it has reached a desired speed. Next, the vehicle quickly turns to pre-load the suspension on one side of the car while maintaining the desired speed. Once the maximum amount of pre-load is obtained, the steering is quickly shifted to the other direction, while still maintaining the desired speed. This quick change in direction allows the suspension to unload on the pre-loaded side while simultaneously rolling the vehicle to the other side. The amount of vehicle roll is affected by both of these actions and makes the vehicle more likely to roll than in most situations. The maneuver is repeated, until a maximum speed at which the vehicle remains stable is found. A diagram of the maneuver can be seen in Figure 4.2. The necessary inputs and corresponding results for this maneuver can be seen in Figure 4.3. The maneuver starts with the vehicle at (0,0) with zero velocity and zero steering angle. Next, the velocity is ramped up to a desired speed while still administering zero steering input. After 30 seconds, a series of abrupt steering inputs are performed. Finally, the vehicle is returned to zero steering input and allowed to stabilize. If the vehicle is able to perform the entire 60 second test, then the desired test speed is deemed safe.

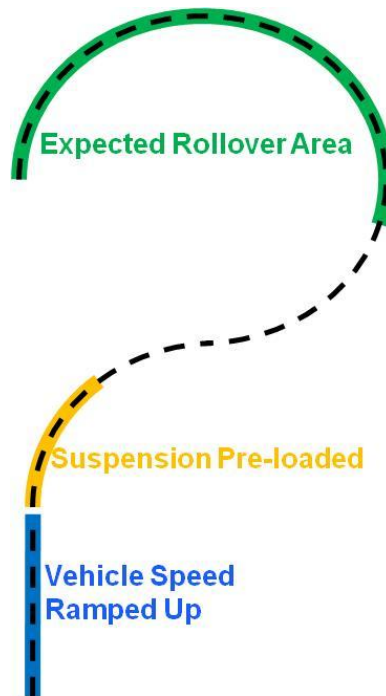


Figure 4.2. Fish Hook Maneuver

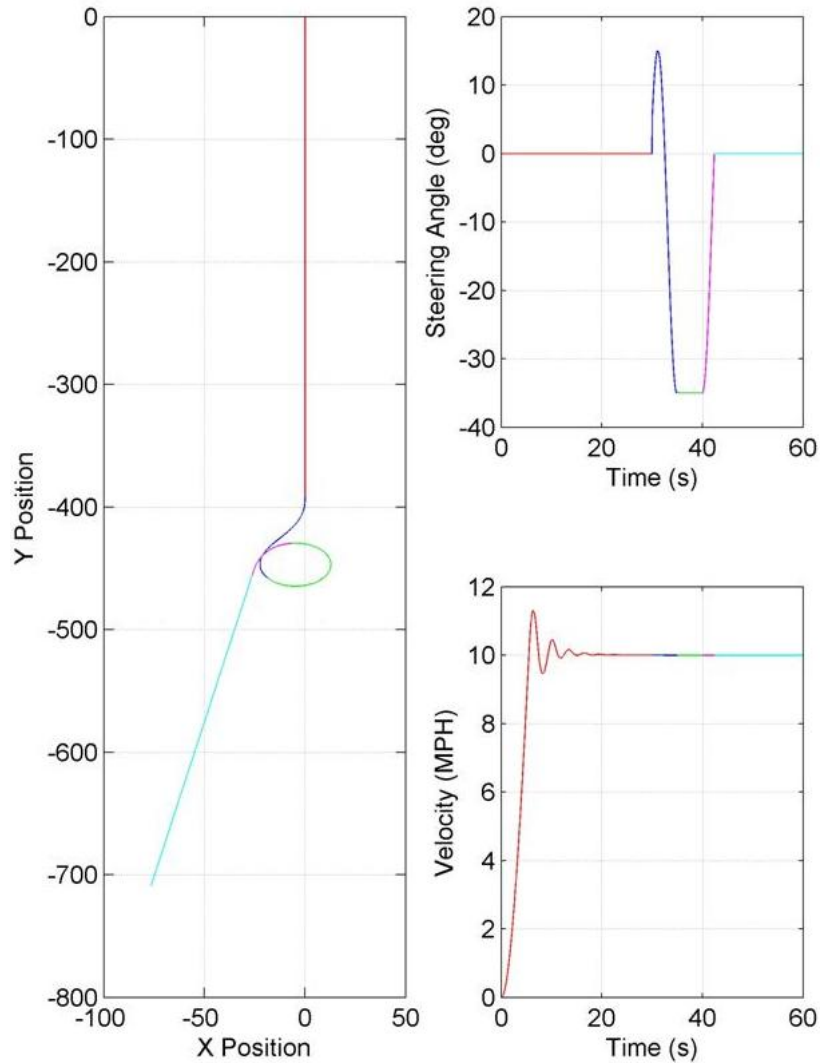


Figure 4.3. Fish Hook Maneuver Inputs and Response

4.3 Random Input

The final and most versatile input type is the random input. This maneuver was designed to accept any input for velocity and steering angle. The system was also designed to load a predefined terrain which can be a random assortment of rolling hills.

This block can also be used to simulate a shaker rig as well. If the velocity and steering angle are set to zero, the terrain inputs are capable of taking a direct signal, which would simulate a shaker rig input. It is exceptionally useful for testing and debugging the control algorithm because it puts the vehicle in the most realistic conditions.

Chapter 5

Rollover Detection

The mitigation strategy can be divided into three subparts: sensing, stability detection, and mitigation hardware control. The relationship of these three subparts can be seen in Figure 5.1. The Circles on the left represent various sensors used to gather needed vehicle data/characteristics. It is important to note that the sensors portrayed are unique to the system that is developed in this work. Other systems may require knowledge of other vehicle states and therefore utilize a different sensor set. Some of these signals will go directly to the interpreter, while others will need to be refined further. The Ground Plane Angle block represents a subsystem of the first part of the strategy. Within this block, information about the vehicle's suspension and vehicle roll are analyzed. The Stability Boundary Interpreter compares the current vehicle characteristics with a predetermined map of vehicle stability characteristics. Based on this comparison, the interpreter will determine whether the vehicle is at risk of rollover. Based on this determination, the interpreter will output a series of needed signals to the Mitigation Hardware Controller. This system was created for use with the model developed in the previous chapters.

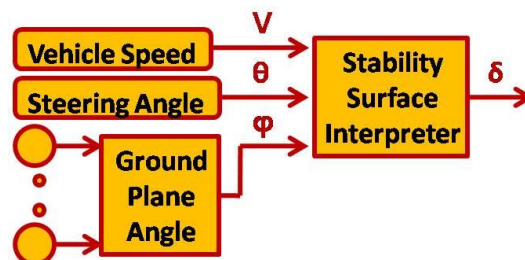


Figure 5.1. Rollover Detection Scheme

5.1 Instrumentation

As can be seen in Figure 5.1, several sensors are required to measure vehicle characteristics. The first of these sensors is for vehicle speed. This sensor is expected to have a low cost, high accuracy and low signal bandwidth relative to the other needed sensors limited by vehicle acceleration limits. This sensor will most likely already be available on most vehicles and the signal can most likely be measured from the hardware already in place, thus incurring little or no additional cost to the mitigation solution.

The second sensor shown in Figure 5.1 is used to measure steering angle. Although future research may conclude that slip angle or actual vehicle steering rate may be more useful, the current strategy uses steering input angle as the worst case scenario. This sensor is expected to have a low cost, high accuracy and moderate signal bandwidth relative to the other sensors limited by human input abilities. This sensor may already be available or can be easily added to any part of the steering linkage, thus incurring little or no additional cost to the mitigation solution.

The last sensory group, shown in Figure 5.1, is the most complex. This consists of a minimum of 5 sensors, 1 of which may be comprised of multiple sensors. A flow chart of this subsystem can be seen in Figure 5.2.

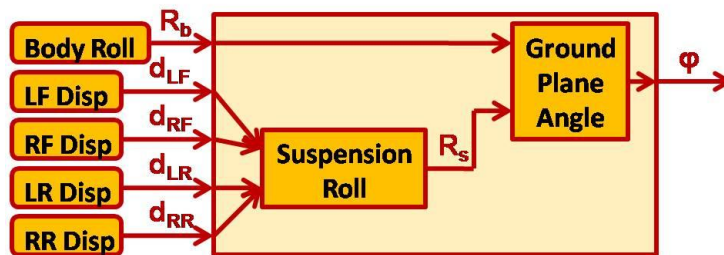


Figure 5.2. Ground Plane Angle Subsystem

As can be seen, the subsystem uses the displacement of each suspension set to determine the suspension roll. This represents the roll between the body and some imaginary “plane” involving all four wheel hubs. This technique will be discussed in a subsequent section. There are many different types of sensors that can be used to measure this deflection that offer a range of prices and accuracy. These sensors may also be readily available on the vehicle if it is equipped with a variable ride height system. Due to the nature of off-road terrain, this sensor will have the highest signal bandwidth requirements of all the sensors.

Also seen in Figure 5.2, a measurement of body roll relative to the earth is required. This is most commonly done with the use of an Inertial Measurement Unit (IMU). These units are often very complex and highly expensive, however they are capable of delivering accurate measurements of roll, yaw, pitch, heave, surge, and sway which may be useful for other systems in a vehicle. Other, possibly cheaper, solutions for measuring body roll can be considered.

5.2 Stability Boundary Interpreter

The signals from all of the instrumentation are sent into the stability boundary interpreter subsystem. A flow chart of this system can be seen in Figure 5.3. The subsystem is comprised of a series of look-up tables that eventually provide a value for maximum stable vehicle velocity based on steering angle and ground plane angle. This maximum stable velocity value is then compared to the actual vehicle velocity. The result of this comparison is a metric that is used in the hardware controller. The stability metric is a measure of how “close” the current vehicle state is to a pre-determined rollover condition. This metric can also be used to determine whether the vehicle is approaching or moving further away from a pre-determined rollover condition. In this sense, the method is very general and can be applied to a wide range of vehicles. Of course, each new vehicle type will require a unique stability boundary defining the pre-determined rollover conditions.

The first of these look-up tables contains data that is unique to each specific vehicle. This table determines the maximum lateral forces that can be applied to a vehicle based on the current ground plan angle. This data will be further discussed in section 5.6.

The second of these look-up tables, contains data that is drawn from a set of simple equations relating vehicle speed, steering angle and lateral force. Based on the nature of these equations, the data is compiled into a look-up table to make it useful. This data will be further discussed in section 5.7.

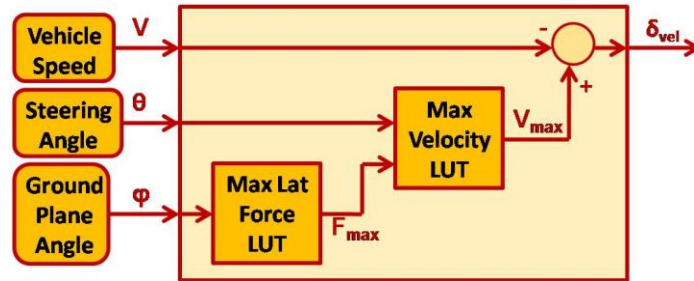


Figure 5.3. Stability Boundary Interpreter Subsystem

Once the maximum stable velocity is determined, it is compared to the current vehicle velocity and a $\delta_{velocity}$ stability metric is determined using Equation 5.1. This value is the output of the Stability Boundary Interpreter and is utilized in the mitigation strategy.

$$\delta_{velocity} = V_{max}(SteerAng, GroundPlaneAng) - Vel \quad (5.1)$$

This value is graphically represented in Figure 5.4. In this plot, the x-axis represents vehicle velocity and the y-axis represents steering angle. The blue curve is representative of a slice of the stability boundary, at a constant ground plane angle, as labeled in the legend. The red ‘X’ is representative of the vehicle’s current state with respect to velocity and steering angle. As long as the ‘X’ stays within the bounds of the blue curve, the vehicle is theoretically not at risk of rollover. The most important aspects of this plot are the dotted lines that determine the relative stability. The dotted blue (vertical) line is representative of the $\delta_{steerang}$ value, which is not used

in this system, and the dotted green (horizontal) line is representative of the $\delta_{velocity}$ value. These values, as can be seen, are the difference between the current state and the intersection points in both the vertical and horizontal directions.

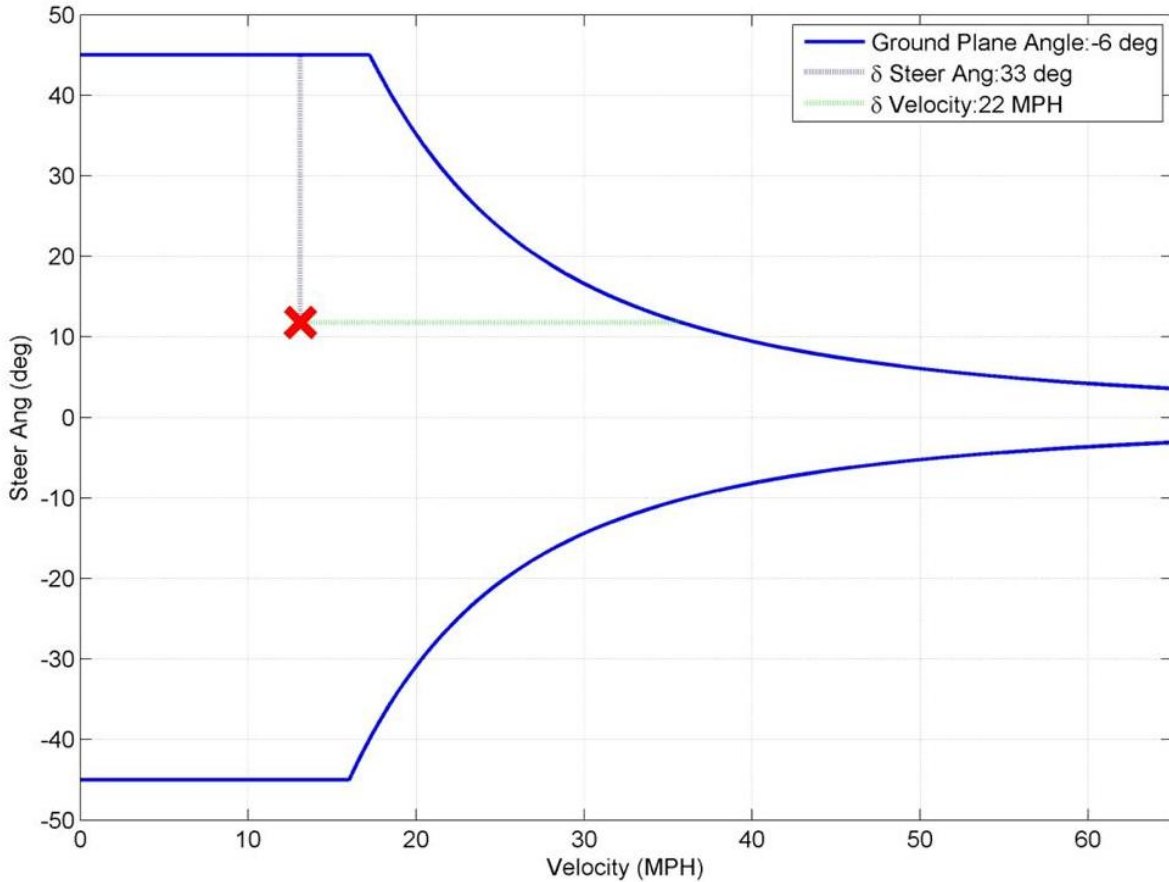


Figure 5.4. Real Time Visualization of the Stability Boundary Interpreter

5.3 Ground Plane Angle

Determining the ground plane angle is an important part of the sensing portion of the strategy. This angle is calculated using a set of simple concepts and assumptions. Figure 5.5 and Equation 5.2 detail this approach.

$$\varphi = R_{body} - R_{suspension} \quad (5.2)$$

As can be seen, the system requires knowledge of the absolute body roll and the suspension roll, both of which are measurable quantities. If the deflection of the tires is neglected, an average ground plane angle can be determined. To sense absolute body roll, an off-the-shelf unit such as an inertial measurement unit (IMU) or robust inclinometer can be used. To sense the suspension roll, potentiometers indicating each wheel's travel are needed. The proposed procedure for determining suspension roll can be seen in section 5.4.

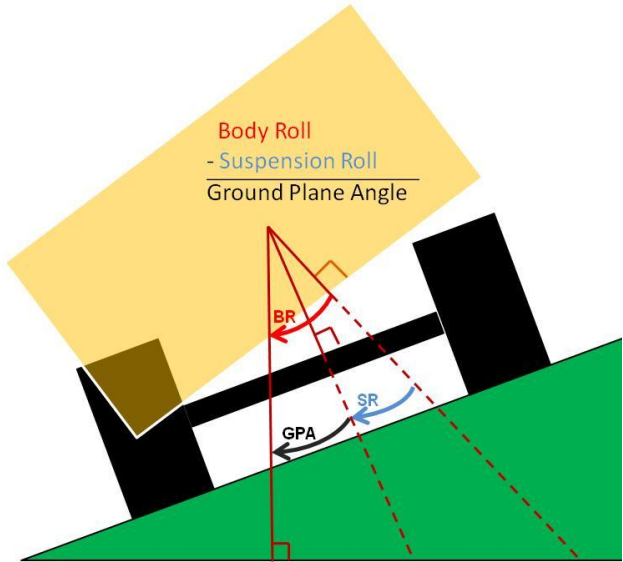


Figure 5.5. Geometric Concept for Measuring Ground Plane Angle

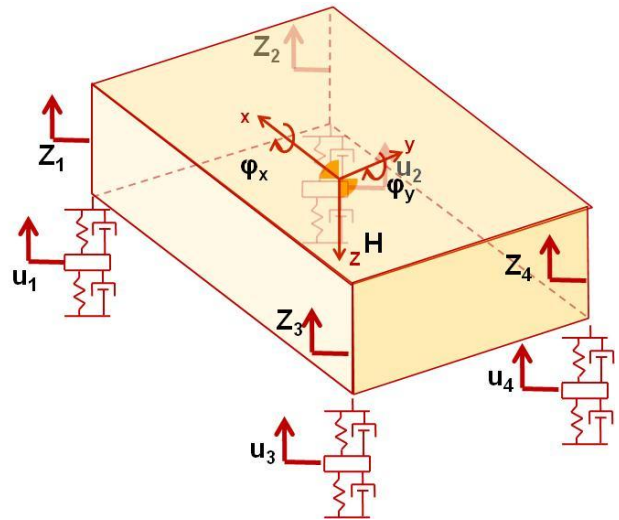


Figure 5.6. Standard 7-DOF Model

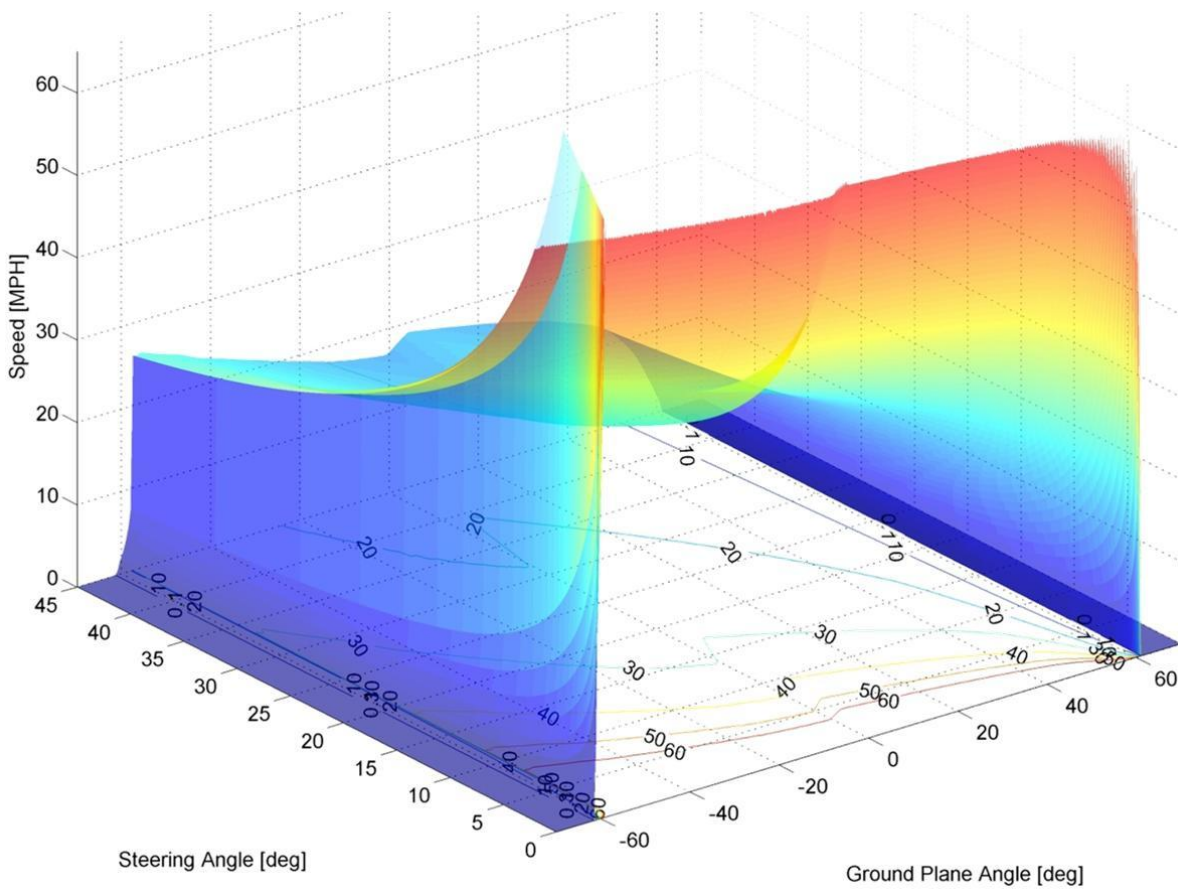


Figure 5.7. 3-Dimensional Stability boundary

5.4 Suspension Roll

The suspension roll is determined using suspension deflection measurements and basic knowledge of the vehicle geometry. When this knowledge is applied to the 7-degree-of-freedom model seen in Figure 5.6, Equation 5.3 can be derived, assuming small angle approximations. This equation is purely kinematic and does not incorporate any dynamics.

$$z_1 - u_1 = \delta_1 = H + \varphi_x y_1 + \varphi_y x_1 \quad (5.3)$$

Where, z_1 is the absolute displacement of the front left corner, u_1 is the absolute displacement of the front left wheel, δ_1 is the relative displacement of the left front suspension system, H is the suspension heave, φ_x is the suspension roll, y_1 is the distance from the CG to the front left corner along the y-axis, φ_y is the suspension pitch, and x_1 is the distance from the CG to the front left corner along the x-axis.

This calculation can be repeated for each corner of the vehicle and combined to create Equation 5.4. This equation shows the relationship between the vehicle geometry (T), main vehicle modes (α), and the suspension deflection (δ). Because the geometry is known and the suspension deflection can be measured, the suspension's heave, pitch and roll can be estimated using the pseudo-inverse of the T matrix, as seen in Equation 5.5, although we only require an estimate of the suspension roll for this analysis.

$$\underbrace{\begin{bmatrix} 1 & y_1 & x_1 \\ 1 & y_2 & x_2 \\ 1 & y_3 & x_3 \\ 1 & y_4 & x_4 \end{bmatrix}}_T \underbrace{\begin{bmatrix} H \\ \varphi_x \\ \varphi_y \end{bmatrix}}_\alpha = \underbrace{\begin{bmatrix} z_1 - u_1 \\ z_2 - u_2 \\ z_3 - u_3 \\ z_4 - u_4 \end{bmatrix}}_\delta \quad (5.4)$$

$$T^+ \delta = \begin{bmatrix} H \\ \varphi_x \\ \varphi_y \end{bmatrix} = \begin{bmatrix} \text{suspension heave} \\ \text{suspension roll} \\ \text{suspension pitch} \end{bmatrix} \quad (5.5)$$

5.5 Stability Boundary

Based on an understanding of the static stability factor from the simple model, we hypothesize that a surface exists in the axes of ground plane angle, steering angle and vehicle speed which separates the space of stable vehicle states from the space of unstable vehicle states. An example of this can be seen in Figure 5.7. This surface was created by combining the 2-D plot in section 5.6 and the function in section 5.7. This relationship of these sections can be seen in Figure 5.3.

The plot in section 5.6 relates maximum lateral force to ground plane angle. The data from this plot was gathered experimentally using the dynamic SimMechanics vehicle model. This process is discussed more in section 5.6. The function in section 5.7 defines velocity as a function of steering angle and lateral force. Because the lateral force input is the maximum stable lateral force, the output of the function is maximum stable velocity.

After compiling the surface, a safety factor was included to insure that the surface was conservative. This is necessary for the system to properly detect rollover in dynamic events.

5.6 Maximum Lateral Force Look-Up Table

The first sub-component of the Stability boundary Interpreter Subsystem is the Maximum Lateral Force Look-Up Table. This table is comprised of data retrieved from simulation and is the simple relationship between the ground plane angle and the maximum allowable forces, in both the positive and negative directions, that a vehicle can endure before rolling over. A plot of the data can be seen in Figure 5.8.

To create this plot, the vehicle model was positioned on a range of ground plane angles. At each ground plane angle, a lateral force was applied and increased until the vehicle experienced two-wheel liftoff. This considered the maximum lateral force that could be applied to the vehicle. To complete the plot, the data was rotated about the origin to account for the forces in the negative direction. This geometric relationship is the result of the relationship between positive and negative lateral force and positive and negative ground plane angle. As the ground plane angle switches from positive to negative, the effect of lateral force is the exact opposite.

The data in this plot follows SAE sign convention, where positive forces act in the positive y direction (out the passenger side) and positive ground plane angle acts along the positive roll direction (clockwise rotation from rear perspective).

5.7 Maximum Velocity Look-Up Table

The second subcomponent of the Stability Boundary Interpreter Subsystem is the Maximum Velocity Look-Up Table. This table uses knowledge of the maximum lateral force and vehicle steering angle to determine the maximum allowed vehicle velocity. This is based on a simple equation using simplified vehicle physics, and can be seen in Equations 5.6 and 5.7. These equations are represented graphically in Figure 5.9.

$$MaxForce(SteerAng, Vel) = mVel^2/R \quad (5.6)$$

$$R = \left| \frac{L}{2 \sin(SteerAng / 2N)} \right| \quad (5.7)$$

Where, L is the vehicle wheelbase, m is the vehicle mass, and N is the steering ratio.

However, the desired equation is maximum velocity as a function of steering angle and lateral force. It was found that Equation 5.7 is non-analytically-invertible and cannot be manipulated to derive the desired equation. Instead, a look-up table for maximum velocity as a function of steering angle and maximum lateral force was created. A plot of this look-up table data can be seen in Figure 5.10 and it can be noted that it resembles Figure 5.9.

Figure 5.10 also includes a maximum speed limiter. The value of 65MPH was set by the research sponsor as the maximum operating speed for a particular vehicle. It was later found that this is an effective way to limit maximum speed of the vehicle.

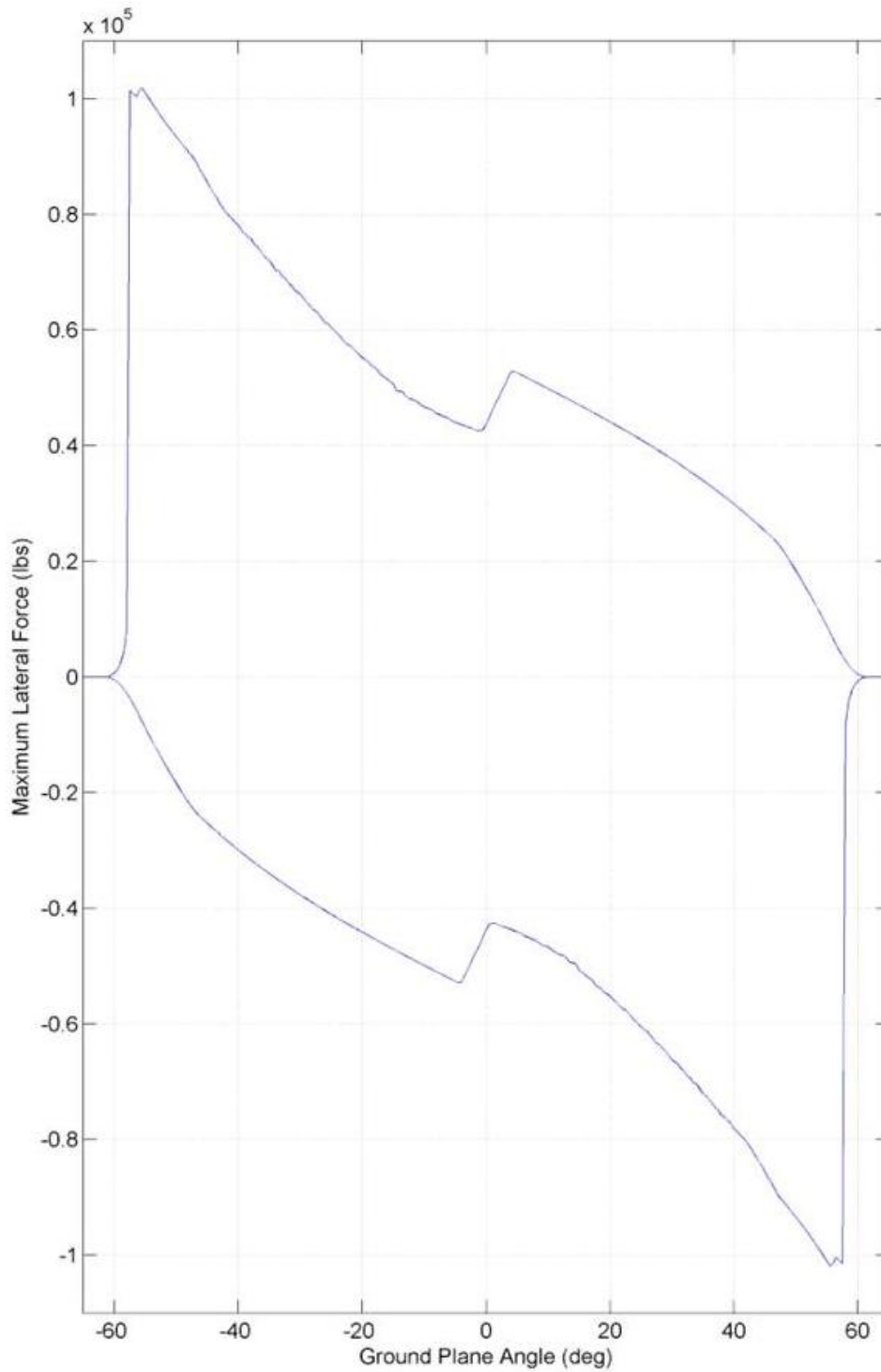


Figure 5.8. Maximum Lateral Force vs Ground Plane Angle. Data taken from simulation.

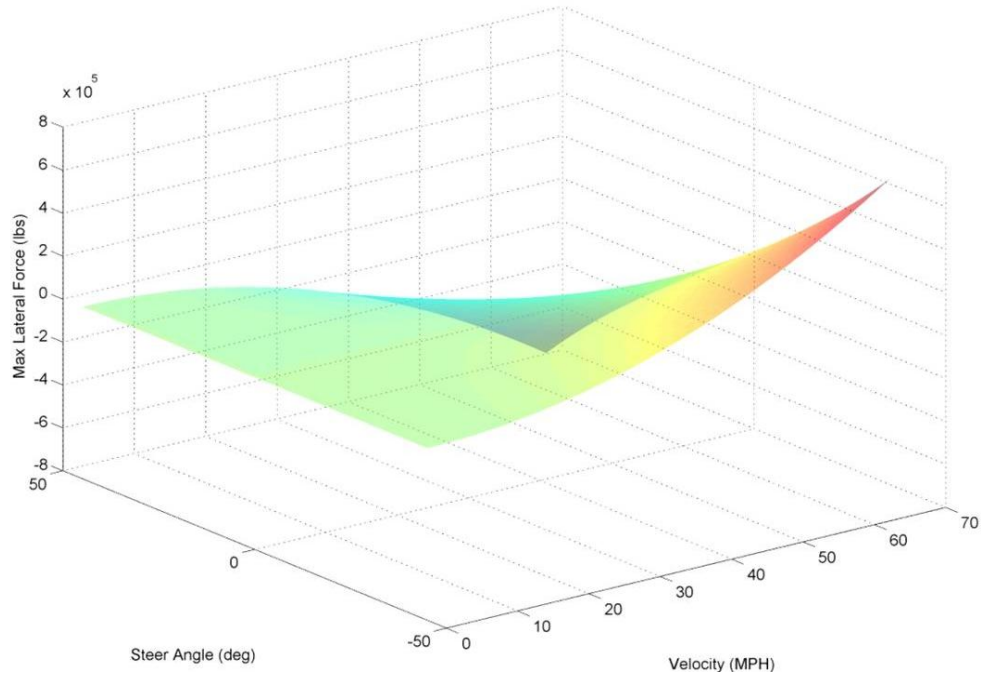


Figure 5.9. Maximum Lateral Force as a function of Steering Angle and Vehicle Velocity.

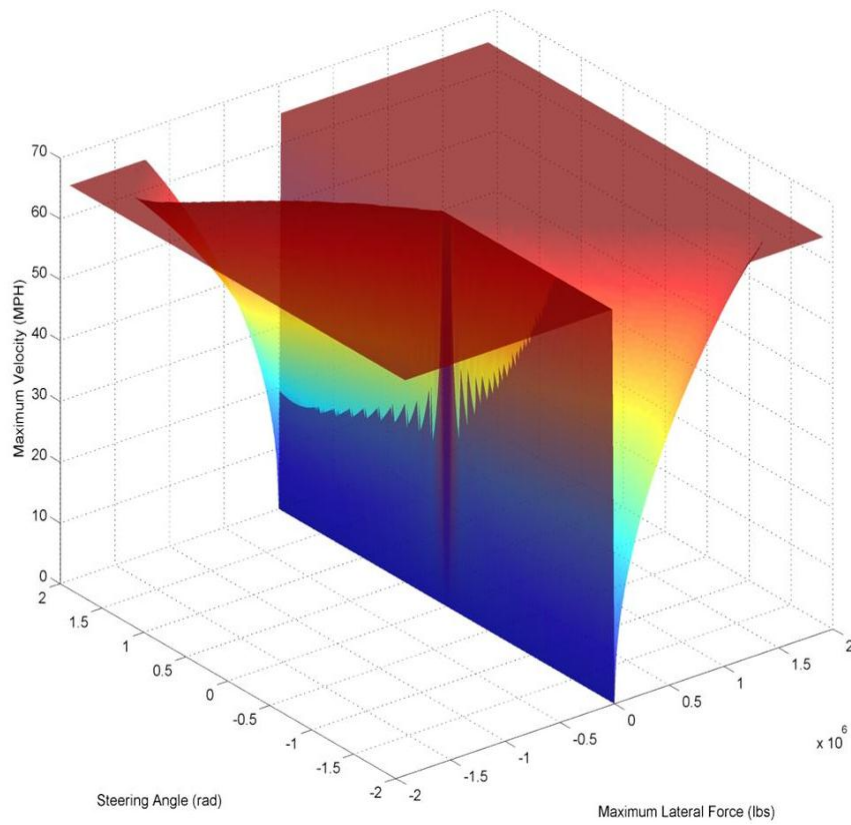


Figure 5.10. Maximum Vehicle Velocity as a function of Steering Angle and Maximum Lateral Force with Velocity Ceiling Added at 65MPH.

Chapter 6

Rollover Mitigation Strategy

As shown in chapter 2, there are numerous ways to implement rollover mitigation. For this study, a simple throttle/brake controller was implemented. This choice was made at the recommendation of the research sponsor and because of the relative simplicity of this mitigation technique. This choice was the driving factor in determining the output signal of the stability surface interpreter and for the tuning and output of the end-stop controller.

The Rollover Mitigation Strategy can be divided into several parts as seen in Figure 6.1. The Stability Boundary Interpreter, discussed in chapter 5, produces a relative stability metric called δ_{vel} . This stability metric and its first time derivative are the inputs to the End-Stop Controller, discussed in section 6.2. This controller produces a throttle control value, T_{cont} , which is designed to push the vehicle states away from the stability surface by reducing the vehicle velocity. The throttle control value and the desired throttle value are then used in the throttle controller. The desired throttle value is created by a PID controller that is discussed in section 6.3. The PID controller utilizes information of the desired throttle from the maneuver profiles, discussed in chapter 4, and the current velocity. The Throttle Controller, discussed in section 6.3, compares the two throttle values and produces the actual throttle input. The throttle value is pushed through a series of filters that represent realistic vehicle throttle response to produce velocity. This velocity signal is then used in the vehicle model discussed in chapter 3.

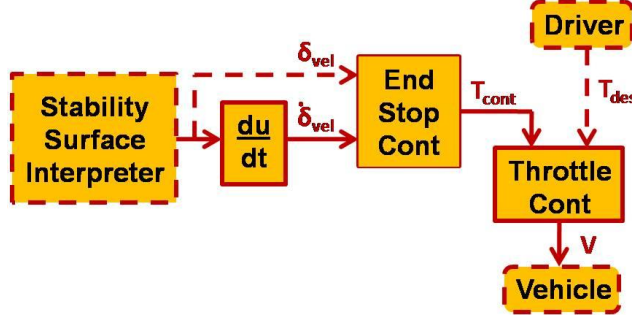


Figure 6.1. Rollover Mitigation Control Strategy

6.1 Generalized End-Stop Control

End-Stop Control is a type of control algorithm typically used to control MR dampers and similar systems. The overall concept of this type of control system is to keep a system from reaching predefined system limits. A typical strategy can be seen in Figure 6.2. The control scheme in this figure is intended to control an MR damper. The x-axis of this plot represents the relative displacement of the controlled system. This axis spans from full extension to full compression of the damper. The y-axis represents the relative velocity, or first derivative of displacement. This axis spans a predefined limit of velocities of the damper moving toward full extension and moving toward full compression. The blue sets of arrows represent the conditions of the damper. The top row of arrows represents the damper extending from full compression on the left to full extension on the right. The bottom row of arrows represents the damper compressing from full compression on the left to full extension on the right. The controller levels can be seen as contours spanning from white to yellow to red to black as noted in the legend on the right. The gains can be seen increasing in the top right and bottom left corners. These corners represent the damper extending toward full extension and compressing toward full compression respectively. These corners represent the damper moving toward its end-stops. The concept of the controller is to push the system away from the end-stops. An example path of motion is represented by the green plot. This plot shows the system oscillating within a mid-range set of values and never fully reaching the end-stops. The shape of the contours is defined by a set of constants that can be manipulated to adjust the impact of the controller. These gains can be seen in Equation 6.1.

$$Output_{ESController} = \beta(\alpha \cdot V^2 - (\delta_{max} - \delta)) \quad (6.1)$$

The constants α and β are tuning parameters and δ_{max} is the maximum displacement, based on which end-stop is closest to the current value of δ . The variables δ and V are the current values of displacement and velocity respectively [14].

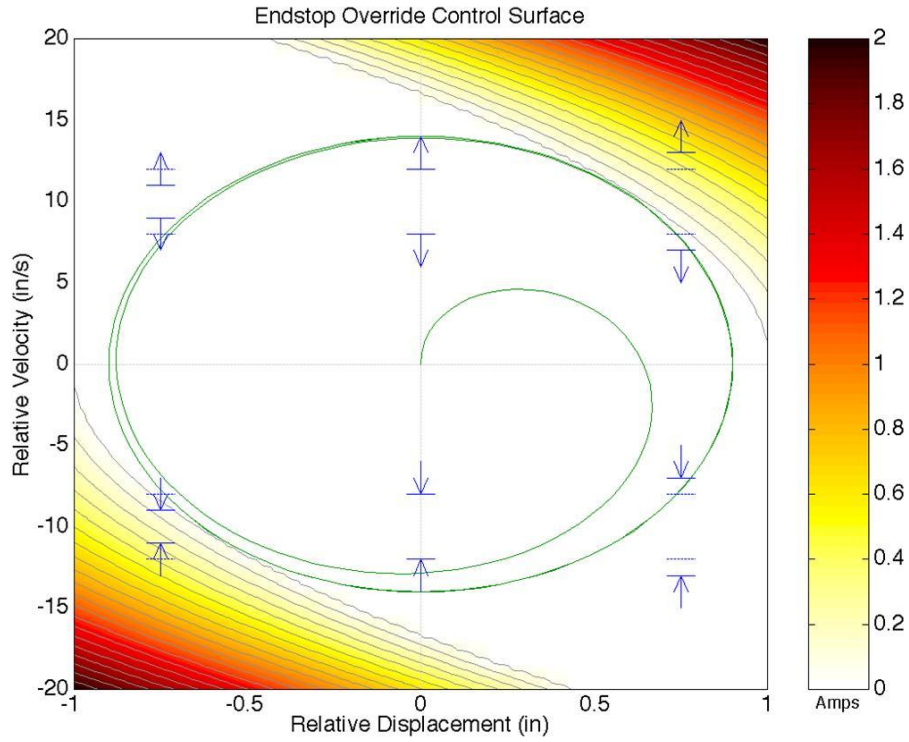


Figure 6.2. General End-Stop Control Scheme

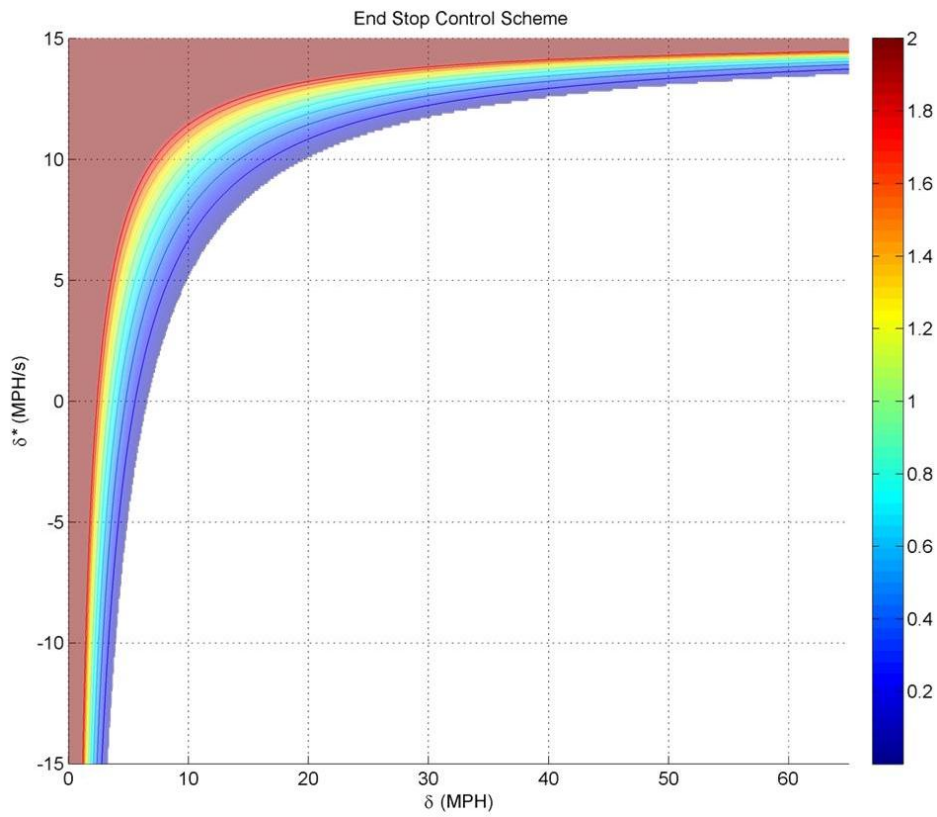


Figure 6.3. End-Stop Controller for Rollover Mitigation

6.2 Adapted End-Stop Control

The control algorithm needed to successfully mitigate rollover will have a similar structure as the generalized end-stop controller, but with a slightly different goal. The generalized end-stop controller prevents the system from reaching the end-stops; however, based on Figure 5.4, it is apparent that the controller should prevent the vehicle state from reaching the stability boundary. Also, the generalized end-stop algorithm uses relative displacement and velocity where the adapted end-stop algorithm uses the $\delta_{Velocity}$ value and a $\delta^*_{Velocity}$, calculated using Equation 6.2. A visual representation of this new end-stop controller can be seen in Figure 6.3. As can be seen in this diagram, the goals of the controller are to drive the system away from zero δ and positive δ^* . This plot is made by using the simple control algorithm seen in Equation 6.3.

$$\delta^*_{Velocity} = \frac{d\delta_{Velocity}}{dt} \quad (6.2)$$

$$T_{cont} = \alpha / \left[\left(\delta / \varepsilon \right) (-\dot{\delta} + \gamma) \right] - \beta \quad (6.3)$$

Similar to Equation 6.1, the constants α , β , γ , and ε are all tuning parameters, which are dependent on the particular vehicle and particular application. The variables δ and δ^* are the current value of $\delta_{Velocity}$ and its first derivative respectively. The output of this equation is then limited between 0 and 2, the reasons for which are discussed in section 6.3.

6.3 Throttle Control

The focus of this control scheme is to limit velocity through throttle control. In this context, throttle is a term used loosely to represent a signal between -1 and 1, where -1 represents full braking and 1 represents full acceleration. The throttle control scheme can be seen in Figure 6.4. This control scheme utilizes inputs from the maneuver profile models and the stability controller. The three major components of this system are the PID controller, stability control signal, and vehicle response filters.

Note that the PID controller only represents a driver's foot on the throttle or brakes, and is not part of the rollover mitigation control strategy. It receives an error signal that is the difference between the desired and current vehicle velocity. The desired velocity is defined by the maneuver profiles and can be time varying depending on the selected maneuver. The parameters of this PID controller are tuned to provide a signal similar to a realistic driver input trying to maintain constant speed. The output of the controller is limited to values between -1 and 1 which represent the driver applying full throttle, full brakes, or anything in between.

The throttle control signal is a value between -2 and 0 and is limited for two main reasons. First, the signal cannot be greater than 0 because increasing velocity will never result in greater rollover stability. Secondly, the signal is allowed a minimum value of -2 so it can override a driver input of 1 in the event that the driver provides full throttle whenever full brakes is needed to maintain rollover stability. This maximum value of -2 is available if the end-user wants the

controller to be able to override the driver's command. If it is not desired to allow the controller to override the driver then a greater value can be established as the limit. The throttle control signal is added to the limited value from the PID controller. This summation results in the unlimited adjusted throttle value. This output is then limited to values between -1 and 1 which represent full throttle or full brakes.

The last part of the throttle dynamics are the vehicle response filters, which can be seen better in Figure 6.5. The vehicle's acceleration and braking responses are represented by separate lag filters. These filters are separate due to the natural differences in these responses, as it is noted that vehicle acceleration due to engine power and torque differs from the deceleration of the vehicle as a result of applying the brakes. A switch determines which filter to process based on the signal that is input. Signals less than 0 go through the brake filter, while all signals greater than 0 go through the acceleration filter. Next, the signal goes into the vehicle damping filter, which applies the effects of the vehicle's inertia. Finally, the signal is limited to be between 0 and 65 MPH.

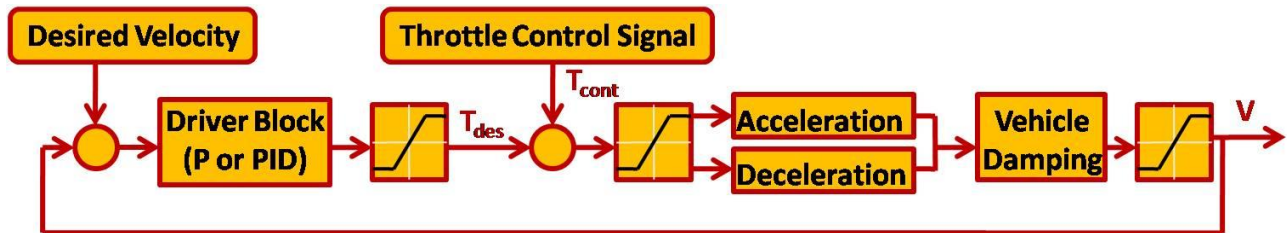


Figure 6.4. Throttle Comparator Used to Determine Vehicle Velocity

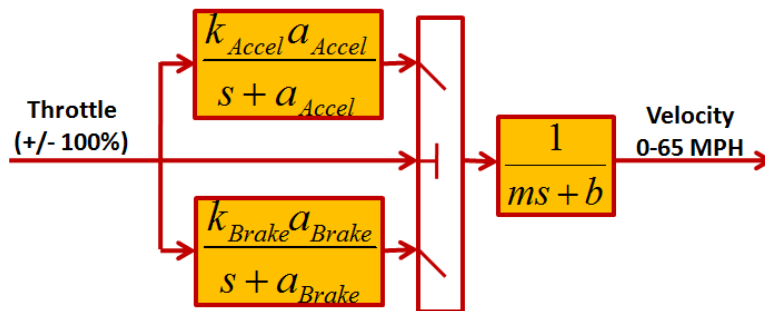


Figure 6.5. Acceleration/Deceleration Filters

Chapter 7

Testing with Simulated Vehicle Model

Two different tests were performed to check the control system. First, the constant radius test was iterated over a range of values and was used to determine the conservativeness of the stability boundary. Next, the NHTSA Fishhook 1B maneuver was utilized to show the effectiveness of the entire mitigation scheme. The constant radius is a quazi-static test and the Fishhook 1B maneuver is a dynamic test and these two maneuvers were chosen to show the ability of the system in both types of situations. The results of these tests can be seen in the following sections.

7.1 Conservativeness of Stability Boundary

There are several reasons that it is important for the stability boundary to be conservative. First, the boundary was developed using data from a static test and it was assumed then that adding a level of conservativeness to the boundary would be suitable to allow the system to properly detect rollover in dynamic situations. Secondly, conservatism helps the system to activate sooner. This is important with many mitigation techniques as the techniques may perform better with more time to react to the potential rollover.

To study the conservativeness of the stability boundary, the quasi-static constant radius test was utilized. This test was conducted over a range of ground plane angles and steering angles to determine the maximum velocity that could be obtained at each combination. The results of each run were then plotted against the stability boundary to determine if the boundary was always less than these tests values. The results can be seen in Figure 7.1. As can be seen, the stability

boundary reflects a maximum velocity that is lower than the obtained velocities from the test at every point tested. This is due to the conservative nature of the stability boundary.

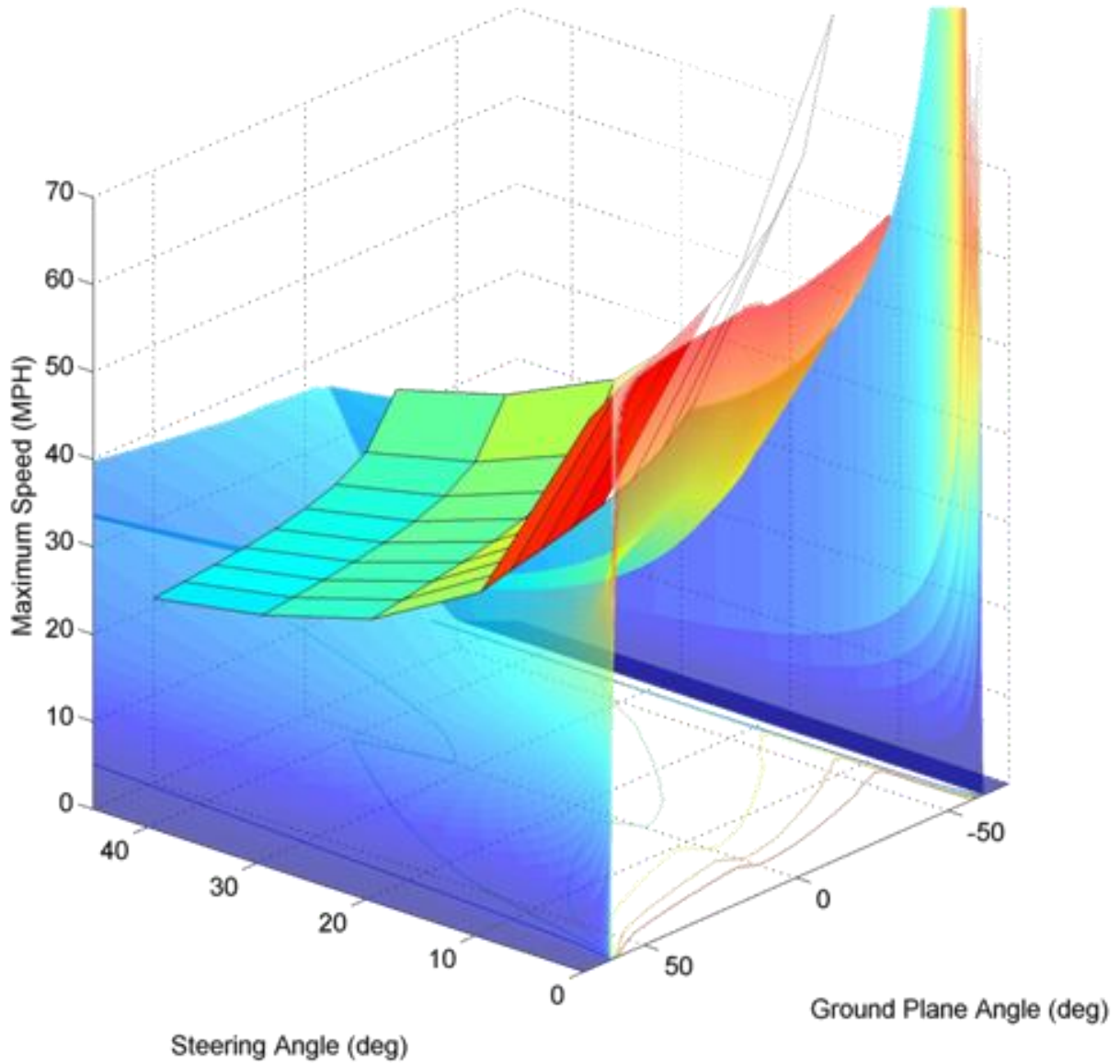


Figure 7.1. Results of Constant Radius Test Compared to Stability Boundary

7.2 Overall Effectiveness Evaluation of Detection and Mitigation Strategy

To test the overall effectiveness of the system, the Fishhook 1b maneuver was performed with and without the controller. The results can be seen in Figure 7.2. As can be seen, the vehicle failed to perform the maneuver without the help of the controller.

The top left plot shows the vehicle velocity. Both simulations start the same, however when the simulation reaches 30 seconds and the vehicle makes its first turn, the controlled vehicle slows down while the uncontrolled vehicle maintains speed.

The top right plot shows that both simulations utilized the same steering input, although the uncontrolled simulation did not finish the maneuver because the vehicle rolled over.

The bottom left plot shows the output of the stability boundary interpreter. This value represents the difference between the current vehicle speed and the maximum safe speed given the ground plane angle and steering angle at the given time. The controlled simulation avoids crossing 0, while uncontrolled simulation fails shortly after crossing that boundary.

Finally, the last plot shows the control signal. Obviously the uncontrolled simulation has no signal, while the controlled simulation is shown delivering a maximum value at strategic times to keep the vehicle under control.

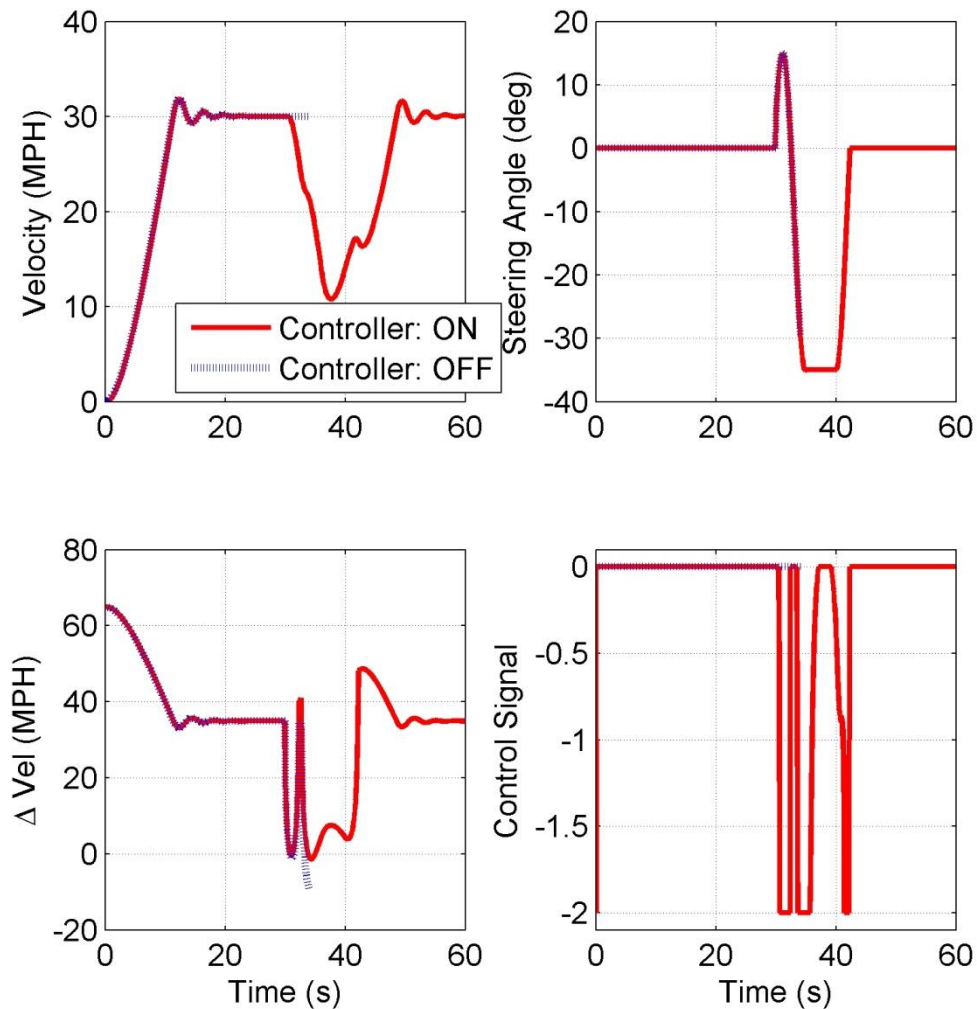


Figure 7.2. Results of Fish Hook Maneuver With and Without the Control Scheme

Chapter 8

Conclusions

The problem of vehicle rollover mitigation has been thoroughly studied and a cost effective solution has been proposed. This study began with an investigation of existing methodologies, algorithms, and hardware. A combination of existing and novel concepts was used to create this system. The process and discoveries are reviewed in this chapter.

The study began by developing two different vehicle models of different levels of complexity. The first was a simple model, which used rigid suspension and tires. It was found that this model is capable of producing results matching that of actual test data provided from field testing (field test data not included in this report). This model was also able to provide a stability boundary similar to that of a model with higher complexity. However, this model was not appropriate for the development of a control strategy because it does not include dynamic motions of the vehicle suspension.

The second vehicle model assumed no slip and had rigid tires but included suspension motion and 3D dynamics. It was found that this model was sufficient for developing a rollover mitigation strategy and was used to do so. Although this model did not incorporate any tire dynamics, it is believed that the system that has been developed is general enough to be implemented and adjusted on a more complex model and still be effective. The nature of the proposed mitigation strategy inherently allows for a level of conservatism which can easily compensate for not incorporating tire dynamics.

It was hypothesized that a boundary exists in the space of vehicle velocity, steering angle and ground plane angle that separates space of stable vehicle states from the space of unstable

vehicle states, where instability is associated with vehicle rollover. The vehicle models were used to test and develop this hypothesis. It was determined that vehicle rollover stability can be characterized by the vehicle state relative to a stability boundary. It was also found that an appropriate factor of safety can be applied to make the boundary conservative.

Next, a system was devised to determine the likelihood of impending rollover based on vehicle states relative to the stability boundary. It was found that impending vehicle rollover can be accurately detected. By utilizing a set of low-cost sensors and easily implementable algorithms, vehicle states can be mapped against a set of known limits regarding rollover. These sensors may already be available on vehicles, or can be implemented using existing hardware.

Lastly, a throttle control algorithm was implemented on the vehicle model and used to prevent rollover. This algorithm utilized modified end-stop control to determine the required control signal based on information provided by the stability boundary. By adjusting the control characteristics of the modified end-stop controller, the control signal and performance of the mitigation strategy could be changed. This throttle control algorithm was used as part of the throttle control mitigation technique was shown to be an effective means of preventing vehicle rollover.

8.1 Future Work

There are many avenues to follow to continue the advancement of this technology. First, a more complex vehicle model including 3-D dynamics, tire dynamics, and environmental effects could be developed and the control system implemented. It is believed that the system can be adapted to be effective with such a model and studying these adaptations can be found beneficial to the advancement of this technology.

Also, implementing this mitigation strategy within a vehicle driving simulator would allow for Human-in-the-Loop testing (HIL). This would allow developers to see how the system responds to very realistic human inputs and would also allow developers to see first-hand how this system directly affects ride quality and handling. It is believed that this testing would be beneficial to the advancement of this technology.

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Appendix A

SimMechanics Model

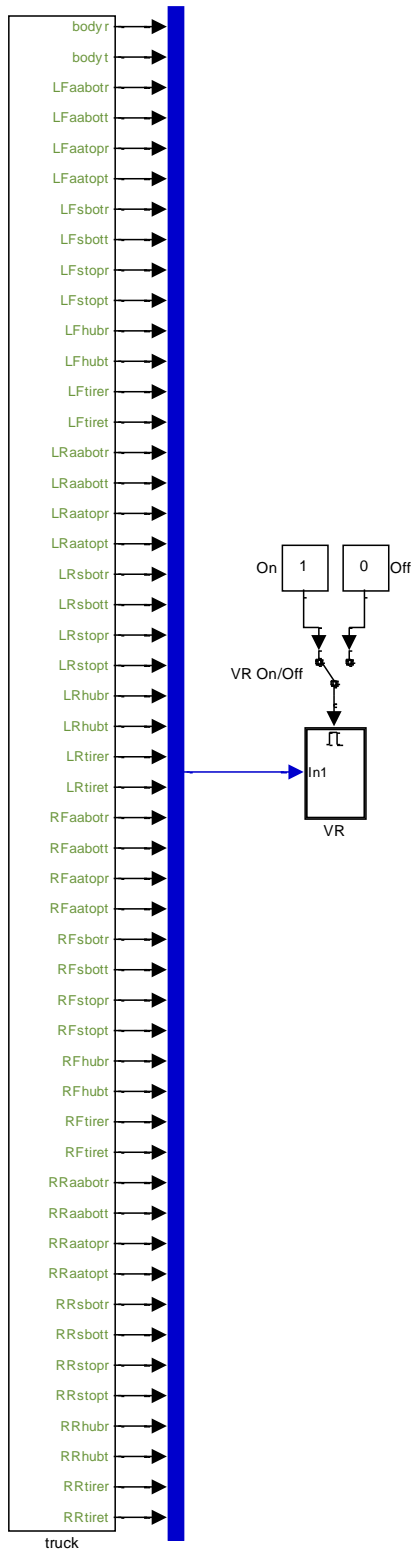


Figure A.1. Vehicle Outputs to Virtual Reality Block

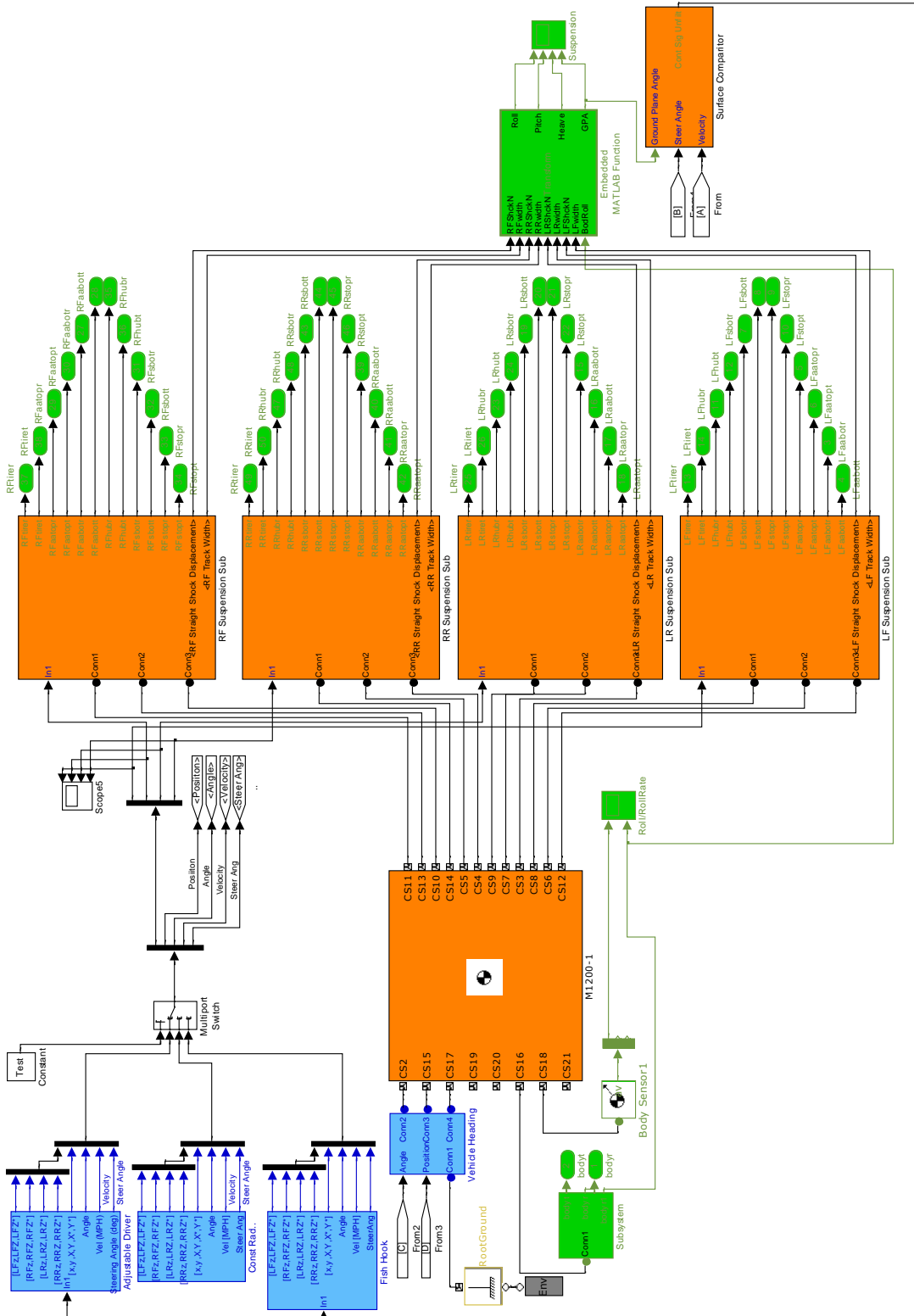


Figure A.2. Entire Vehicle Model Showing Relationships of All Sub-Systems

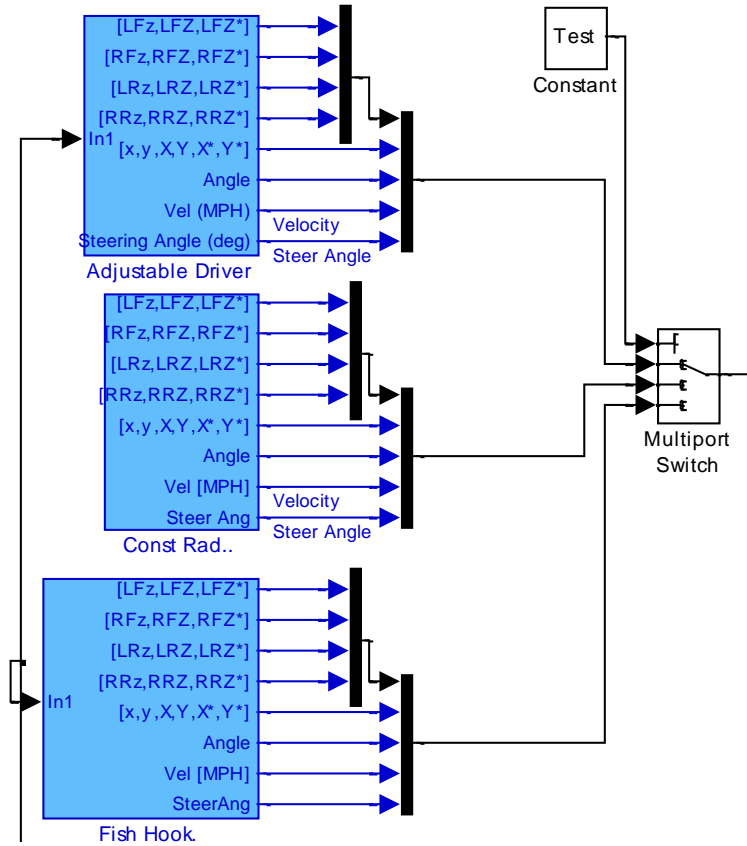


Figure A.3. Vehicle Maneuver Profile Blocks with Switch.

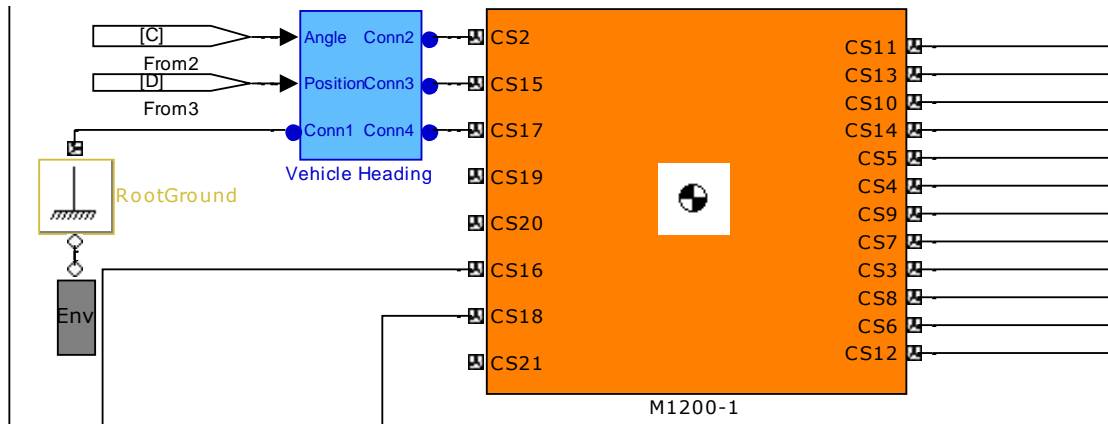


Figure A.4. Main Vehicle Body Featuring Coordinate Position and Heading Angle on Left and Suspension Mount Points on Right

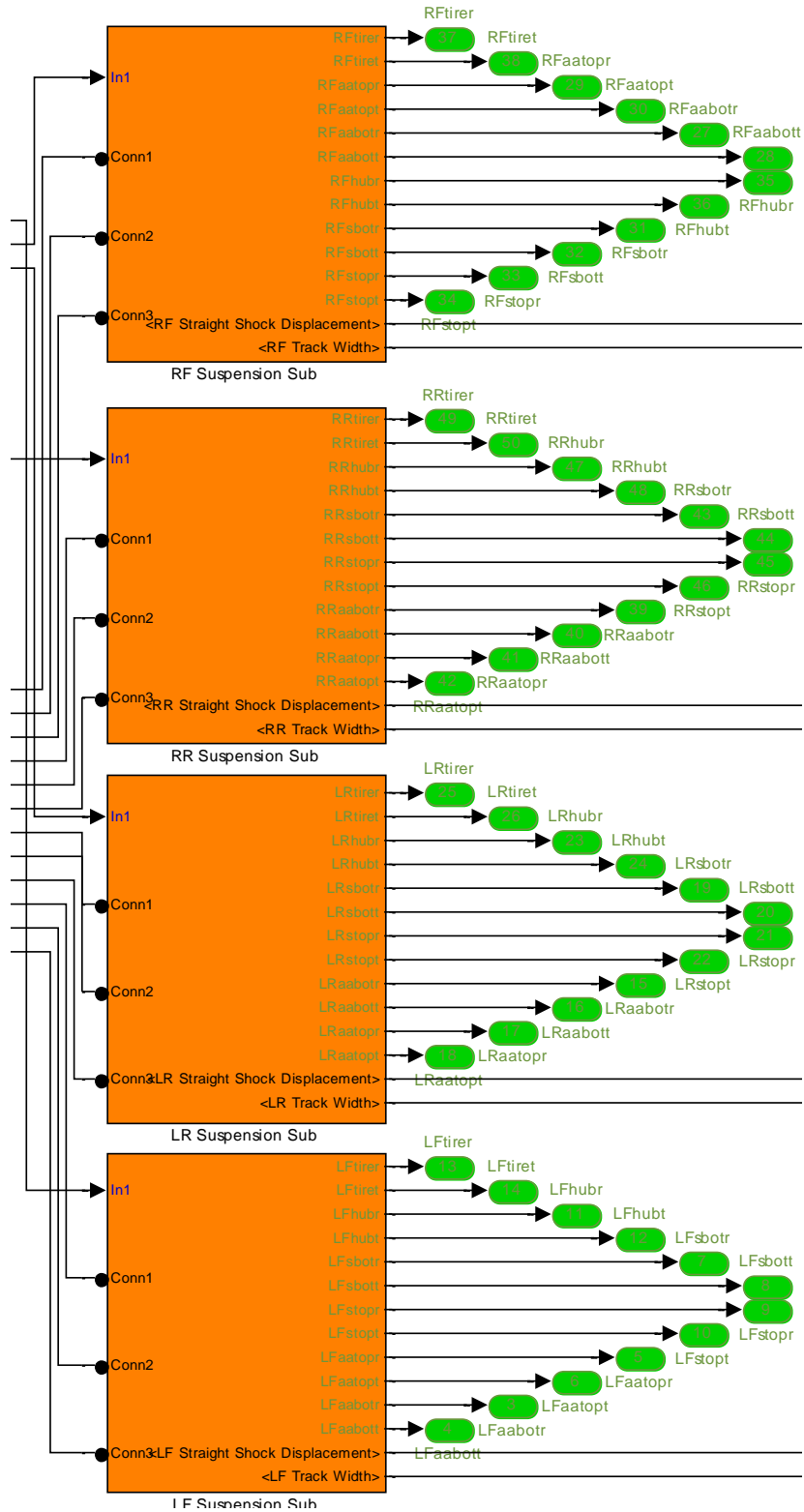


Figure A.5. Suspension Sub-Systems, Each Featuring 3 Connections to the Main Vehicle Body and Individual Tire Displacements on Left and Virtual Reality and Vertical and Horizontal Tire Displacements on Right

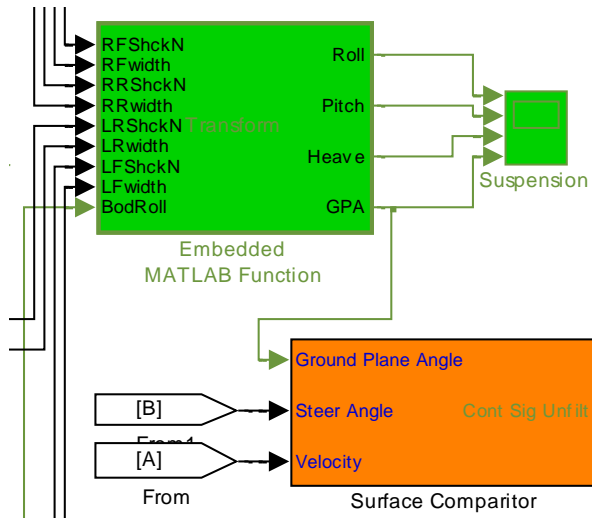


Figure A.6. Tire Deflection Transform Including Ground Plane Angle Solver and Stability Boundary Comparitor

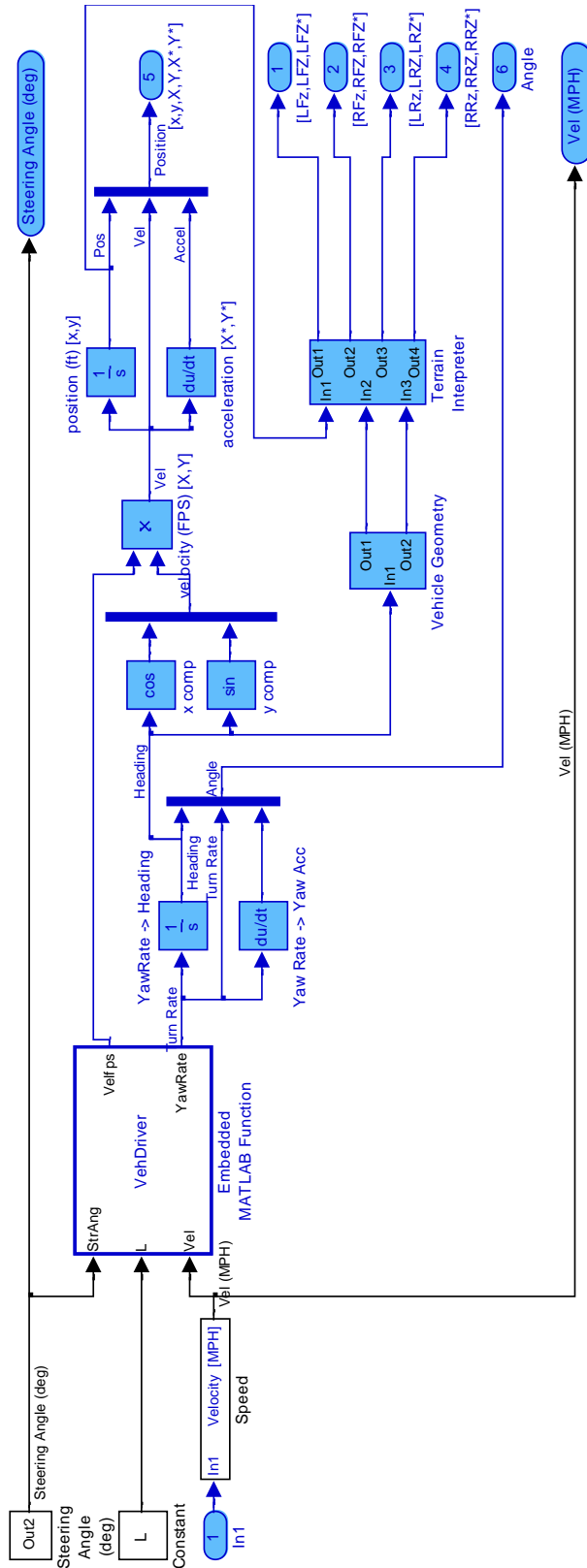


Figure A.7. Random Input Maneuver Profile

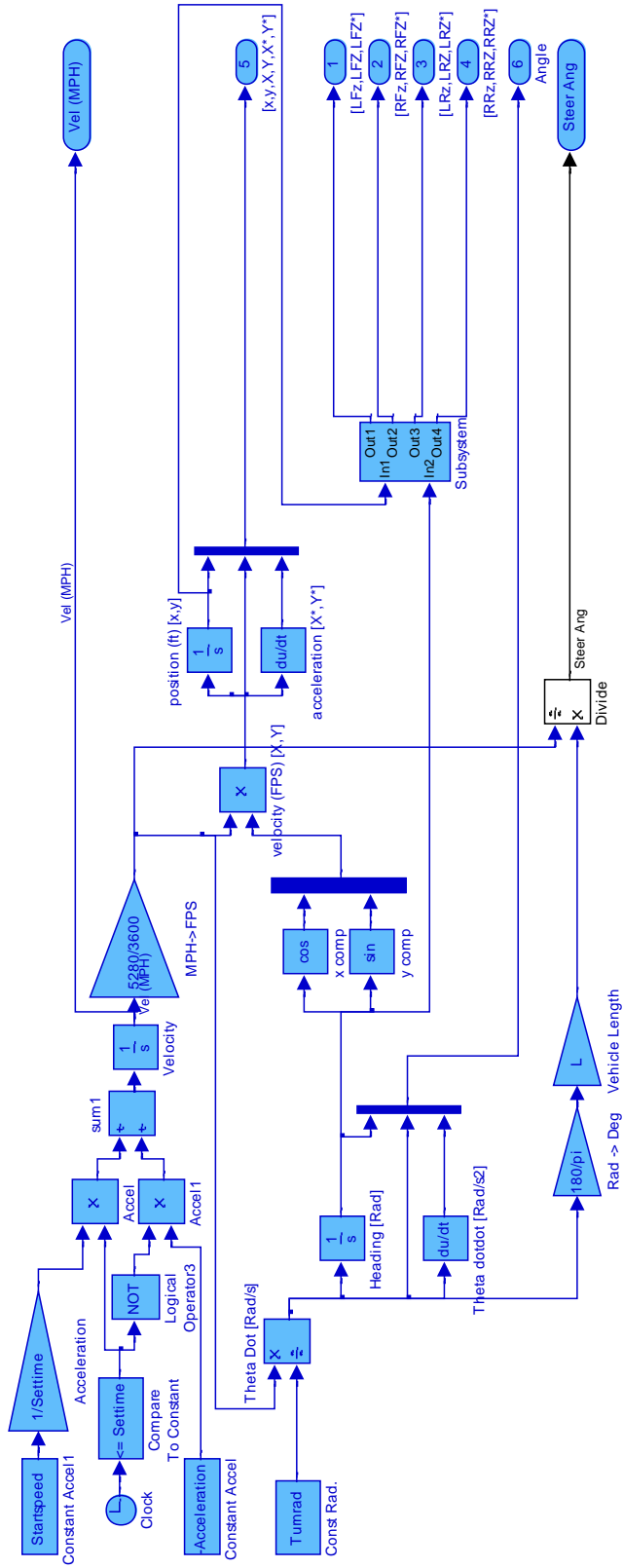


Figure A.8. Constant Radius Maneuver Profile

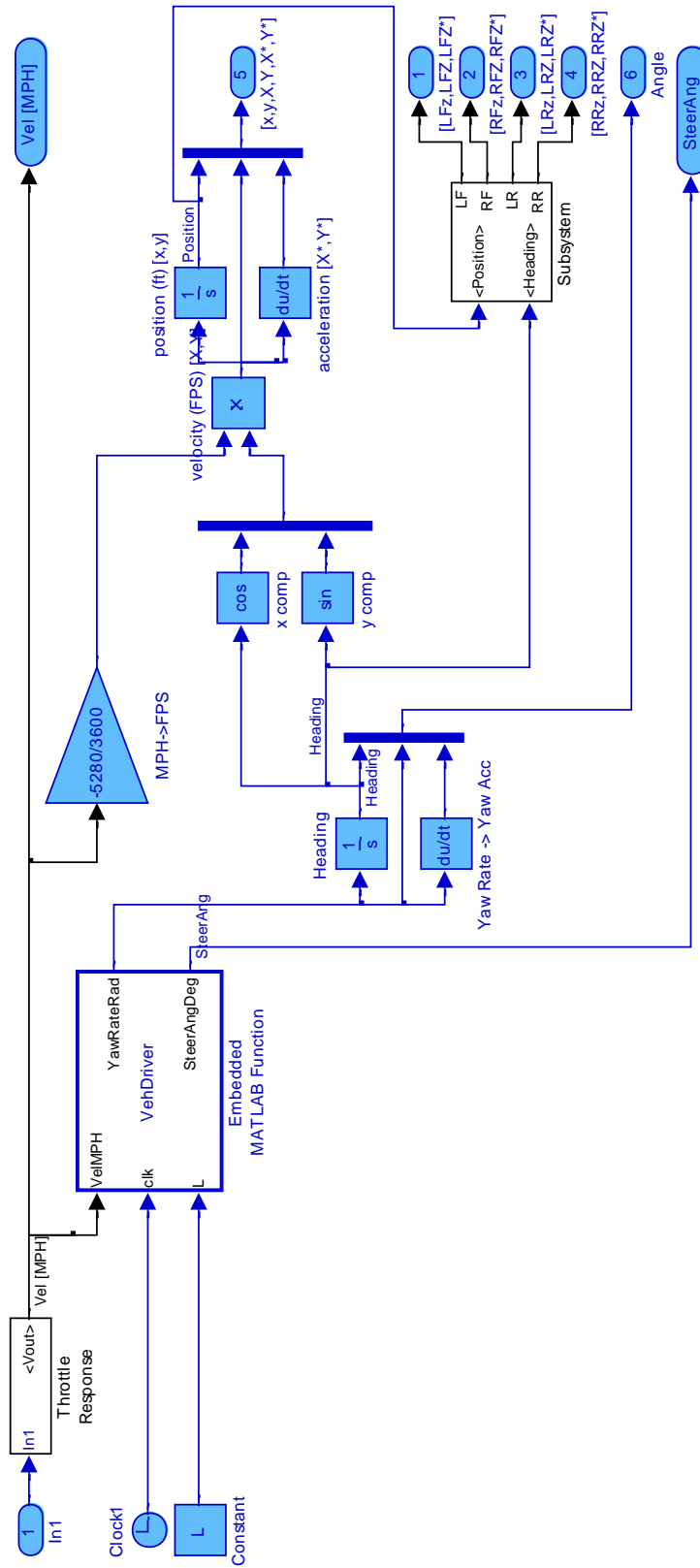


Figure A.9. Fish Hook Maneuver Profile

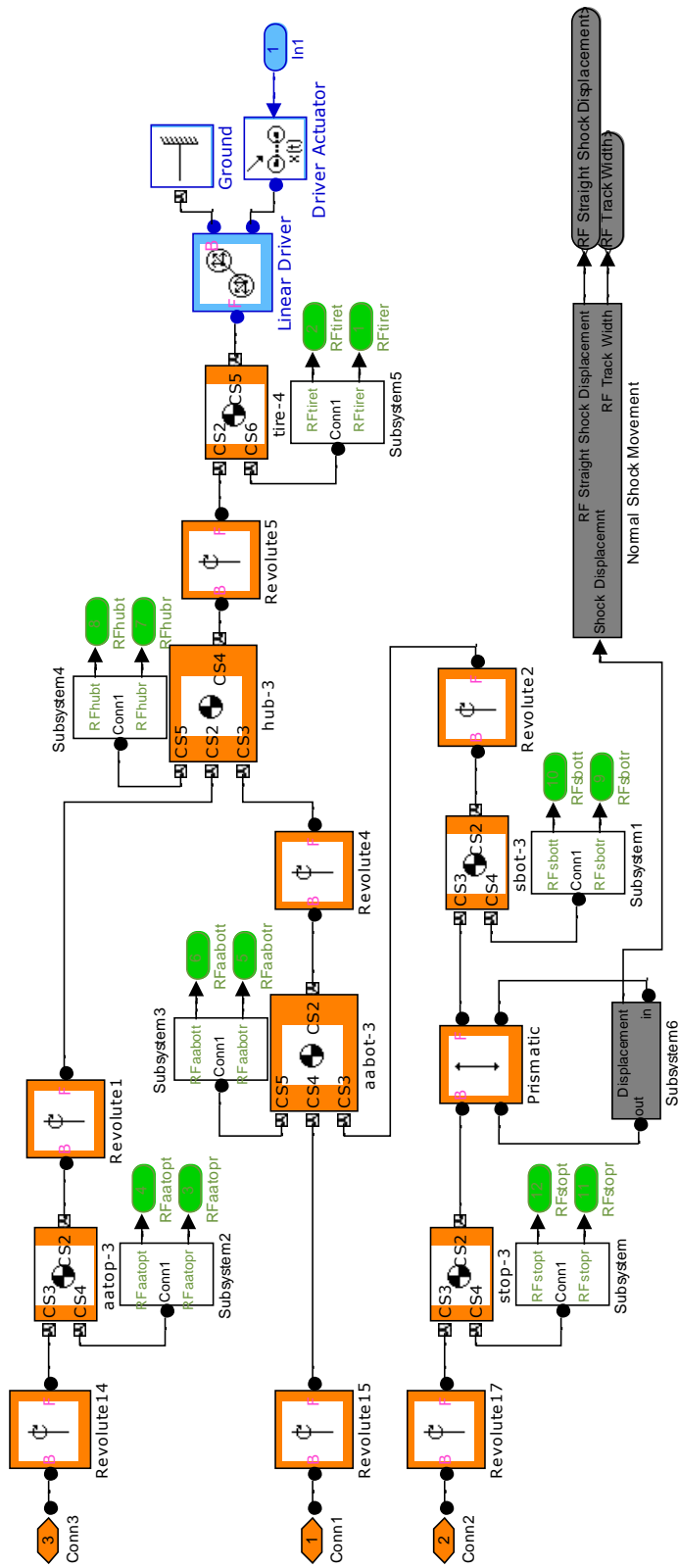


Figure A.10. Suspension Sub-Assembly

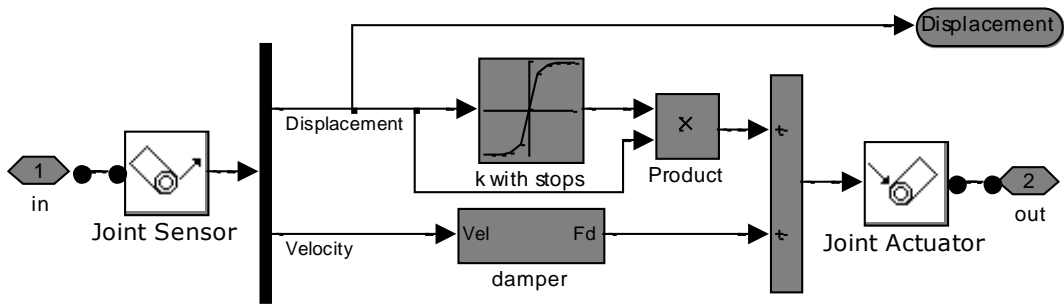


Figure A.11. Spring and Damper Sub-Assembly

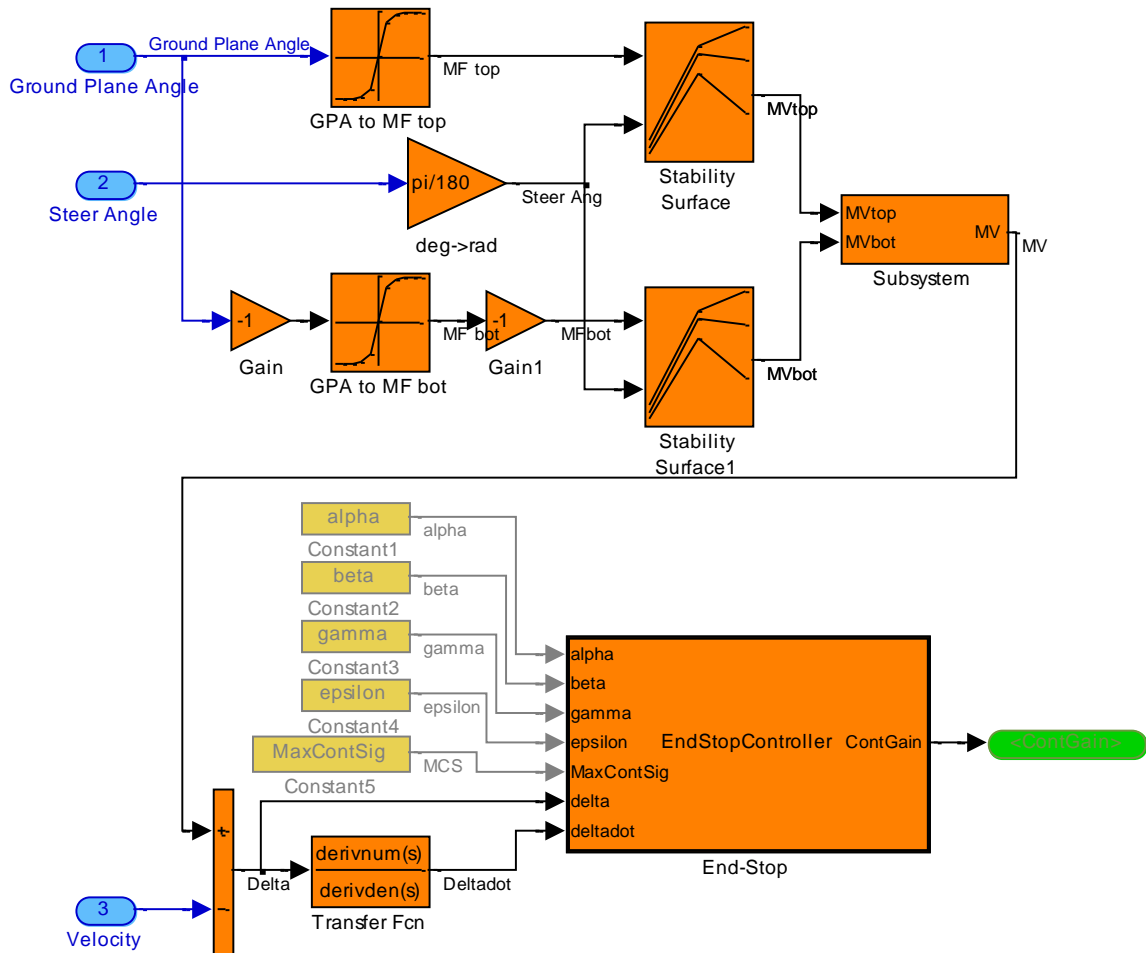


Figure A.12. Stability Boundary Comparitor

Appendix B

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Figure B.1. Fair Use Analysis of A Comprehensive Experimental Examination of Test Maneuvers That May Induce On-Road, Untripped, Light Vehicle Rollover - Phase IV of NHTSA's Light Vehicle Rollover Research Program [2]

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Figure B.2. Fair Use Analysis of Detection of Vehicle Rollover [4]

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Figure B.3. Fair Use Analysis of A Method for Reducing On-Road Rollovers -- Anti-Rollover Braking [21]

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Figure B.4. Fair Use Analysis of Fundamentals of Vehicle Dynamics [28]