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VEHICLE DYNAMICS CONTROL WITH ROLLOVER PREVENTION FOR ARTICULATED HEAVY TRUCKS

Daniel D. Eisele Huei Peng
University of Michigan, Ann Arbor, MI

Abstract

Rollover and jack-knifing of articulated heavy trucks are serious threats for motorists. Active safety technologies have been demonstrated to have potential to reduce or prevent the occurrence of these types of accidents. The Vehicle Dynamics Control (VDC) system utilizes differential braking to affect vehicle response and has been shown to be quite effective in controlling vehicle yaw response. In this paper, a VDC system that improves yaw, lateral, and roll stability is presented. The objectives of this VDC design are to prevent or reduce the likelihood of rollover and jack-knifing and to make the vehicle more closely follow the driver's intended path. A linear root locus study is performed to tune controller gains in a systematic fashion. Nonlinear dynamics simulations of a generic articulated heavy truck are run with the TruckSim and Matlab/Simulink software to study the performance of the proposed VDC algorithm. Human-in-the-loop driver models are used to obtain realistic steering inputs on predetermined test track. The simulation results of maneuvers utilizing these driver models, as well as maneuvers utilizing prescribed steering inputs, are presented. VDC is shown to stabilize the vehicle, rollover and jack-knifing are prevented and the vehicle more accurately follows the driver's intended path.

1. Introduction

Rollover and jack-knifing of heavy articulated trucks are serious hazards to safety for all motorists. An out of control heavy truck has the potential to cause injury or death to its operator as well as other motorists and can inflict much damage to property because of its size.

An indication of the potential danger that articulated heavy trucks create can be found in accident statistics. Rollover was involved in 595 fatal accidents and 10000 injury crashes involving large trucks in 1996 (NHTSA, 1997). 69 percent of these fatal accidents and 60 percent of these injury crashes involved combination trucks. Additionally, jack-knifing occurred in 266 fatal accidents and 2000 injury crashes in 1996.

The Vehicle Dynamics Control system (VDC) actively brakes individual wheels to directly influence vehicle yaw dynamics. Thus VDC can prevent jack-knife and, if so designed, can even reduce the chance of rollover. Application of VDC to prevent rollover as well as improve yaw dynamic is the main topic of this paper.

VDC traditionally has been applied to control the lateral and yaw vehicle dynamics (Zanten et al, 1995). The objectives of most current VDC systems are to prevent spin out and improve the ability of the vehicle to follow the driver's intended path. Typical VDC systems use differential braking of individual wheels to generate a stabilizing yaw moment. VDC systems are typically designed so that application of differential braking and the resulting yaw moment ensure that at least two criteria are met. The first is to keep the vehicle side slip angle β from growing large. This is to prevent spin out. The second is to manipulate the yaw moment so that the vehicle yaw rate matches the estimated desired yaw rate commanded by the driver. This will increase the chances that the vehicle will follow the driver's intended path.

VDC's ability to improve stability and yaw rate tracking has been shown in simulations (Zanten et al, 1995) and in testing (Laffler et al, 1998). Through differential braking, VDC is able to prevent spinouts that occur during obstacle avoidance, panic braking, steady state cornering, and split-μ road surface conditions. VDC implementations are becoming increasingly widespread for passenger cars, however manufacturers have not been as quick to adopt VDC for heavy trucks. BOSCH has developed a VDC system specifically for articulated heavy vehicles (Hecker et al, 1997). This system applies differential braking to the tractor wheels to prevent jack-knifing.

While yaw instability leading to jack-knifing is a serious problem that can be avoided with VDC, rollover is of great concern as well. One approach to directly reduce roll angle and the chance of rollover uses active suspension (Sampson et al, 1998). While effective, active roll control requires expensive actuators beyond those currently employed on vehicles. An attractive option would require only the existing actuators currently in use on heavy trucks. One such actuator system that is already employed is the braking system.

Braking can provide an effective means of reducing roll due in part to the coupling between roll, lateral, and yaw dynamics. This coupling allows the use of yaw moment generation through differential braking to influence roll. A second effect is provided by the nonlinear nature of pneumatic tires. Since roll is directly caused by the lateral acceleration of the center of mass, a reduction in lateral acceleration can also reduce the roll angle. When tires are made to generate a longitudinal force through braking, their reserves of grip for lateral force generation are correspondingly reduced. Thus, actively braking the axles can reduce lateral acceleration and in turn reduce the roll angle. A third beneficial effect of braking is that vehicle speed is reduced. Rollover often occurs when the vehicle is traveling at a rate too high for a given curve. Braking slows the vehicle increasing the chance that it will negotiate the curve without rolling over.

These effects have been utilized in a software addition to the Electronic Brake System (EBS) (Palkovics et al, 1998). EBS is a new system that provides brake control for heavy trucks. Palkovics's system uses the wheel speed sensors from the EBS to determine if wheel liftoff has occurred. If it is determined that it has, the system brakes the opposing wheels to reduce its lateral force generation capability and vehicle speed.

Another approach (Wielenga, 1999) uses a lateral acceleration sensor in combination with suspension stroke sensors to determine when rollover is a threat. This system activates the front brakes to lower vehicle speed and induce understeer. While this understeer decreases the likelihood of rollover, it may conflict with the driver's desire to remain on the road.

Both Palkovics' and Wielenga's approaches consider only rollover prevention. If VDC is applied to control the yaw dynamics of the heavy articulated truck, it is possible to incorporate antirollover prevention measures as well. The main focus of this paper is to ascertain the effectiveness of VDC to antirollover as well as yaw/side slip control when applied to a heavy articulated vehicle.

In the following sections of this paper are presented as follows: Section 2. summarizes the modeling of the vehicle and driver, Section 3. describes the VDC algorithm used, Section 4. presents the simulation results used to test the VDC system and Section 5. states the conclusions obtained from this work.

2. Modeling

2.1 TruckSim Vehicle Model

TruckSim is a simulation package that can model articulated and single unit trucks of various sizes and with varying number of axles. It uses a graphical front end that allows the user to adjust model and simulation parameters and to analyze simulation results with plots and animations. It also includes an interface with Simulink allowing easy control design. TruckSim is available commercially from Mechanical Systems Corporation.

The choice of vehicle configuration (shown in Figure 1) chosen for this paper is a generic 3 axle/2 axle tractor-semitrailer combination with fifth wheel. The TruckSim model for this vehicle has 91 state variables and 29 multibody degrees of freedom. The tractor sprung mass has six degrees of freedom (lateral, longitudinal, vertical, yaw, pitch, and roll). The semitrailer is constrained by the fifth wheel but can roll, yaw, and pitch about this constraint. The fifth wheel is modeled as a ball-joint with stiffness in roll. In addition to the tractor and semitrailer sprung masses, the unsprung suspension masses are modeled with vertical and roll degrees of freedom. Individual wheel speeds are also taken into account.

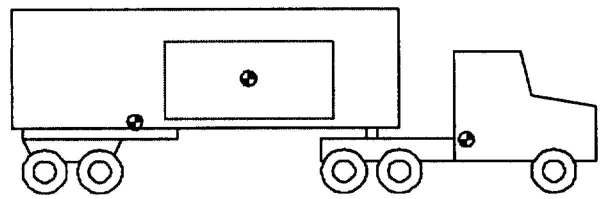

Figure 1. 3 Axle Tractor/2 Axle Trailer with Trailer Load

Important submodels of the TruckSim model are the nonlinear tire model and the nonlinear spring model. The tire model is critical to accurately predicting loss of grip in the longitudinal and lateral directions (Yasui et al, 1996). Correct modeling of longitudinal grip is necessary to help define the saturation limit of the differential braking applied by the VDC. Correct modeling of lateral grip is important for analyzing yaw instabilities such as jackknifing.

2.2 STI Crossover Principle Driver Model

This model (Rosenthal et al, 1988) is based on the principle that a human will adjust his/her inputs such that the frequency response of the loop transfer function of the vehicle with driver will cross 0 dB at a particular frequency. The human will try to ensure this occurs regardless of the plant he/she is controlling. The model calculates the driver's steering wheel input based on motion cues and dynamics associated with human reactions.

The STI model calculates lane position error and road curvature error based on the speed dependent road kinematics. These signals are fed to the driver along with yaw rate to determine steering wheel angle. The dynamics due to the driver's reaction times and neuromuscular system are modeled as a second order system with delay. As stated above, gains for the driver controller are chosen such that the loop transfer function will cross 0 dB at the chosen crossover frequency ω_c. These gains are speed dependent and are updated in real-time during the simulation.

This driver model provides good performance in the constant radius ramp-entering maneuver. The driver is able to enter the ramp and quickly find the steady state steering required to remain in the desired lane position through the curve.

2.3 Lane Preview Driver Model

To achieve good performance during the double lane change maneuver, a second driver model is used. In this model, the driver is a PI controller. The outputs of the control are delayed and modified by neuromuscular dynamics as in the STI model. The control attempts to minimize the lane position error. It is fed an error signal based on a previewed lane position and previewed lane command. The future lane position is estimated using the Euler approximation

$$y(t+t_p) \approx y(t)+t_p\dot{y}(t)+(t_p^2/2)\ddot{y}(t) \qquad (1)$$

where t_p is the preview time and y is the lane position. Preview allows this driver model to react in advance of the double lane change command to compensate for the lag due to the vehicle dynamics. This ensures a better match between the lane command and the lane position during a sudden maneuver.

3. Control Algorithms

3.1 Electronic Braking System

The Electronic Braking System (EBS) is designed to modulate brake pressure to limit the tire slip ratio λ so that braking forces are maximized and steerability is retained (see figure 2). A rule-based EBS is used in this research. Brake pressures are determined from the driver's input, the VDC differential pressures, and tire longitudinal slip ratio. Figure 3 shows the block diagram of the braking system. To simplify the design of the EBS, values for λ were taken directly from TruckSim outputs rather than estimated. The rules for the EBS are:

if $|\lambda|<\lambda_{min}$ then $\Delta P_{ebs}=-C_{EBS1}(\lambda_{min}-|\lambda|)|P_{drv}+P_{vdc}|$
else if $|\lambda|>\lambda_{max}$ then $\Delta P_{ebs}=C_{EBS2}(|\lambda|-\lambda_{max})|P_{drv}+P_{vdc}|$
else if $|\lambda|>\lambda_{peak}$ then $\Delta P_{ebs}=C_{EBS3}\Delta P_{ebs}$
$P_{ebs}(k)=P_{ebs}(k-1)+\Delta P_{ebs}$

where C_{EBS1}, C_{EBS2}, and C_{EBS3} are control gains. The first two rules increase and decrease the brake pressure respectively to keep λ in between λ_{min} and λ_{max}. The third rule gradually releases the brake pressure when λ is near the value corresponding to tire peak longitudinal force. Despite the fact that ΔP_{ebs} can be positive or negative, the EBS will never increase brake pressure above $|P_{drv}+P_{vdc}|$.

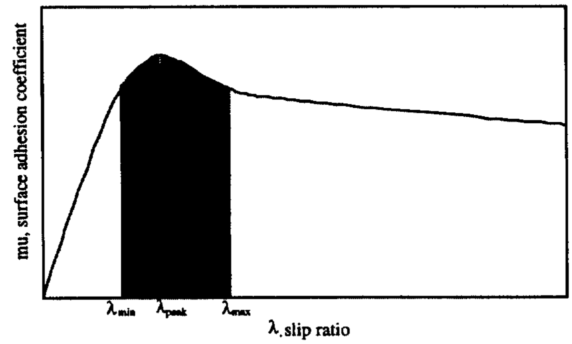

Figure 2. Longitudinal Braking Force Curve

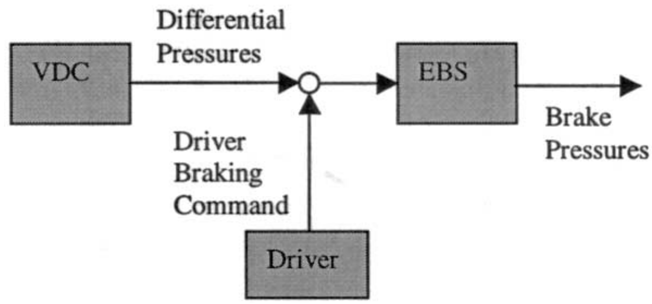

Figure 3. The Braking System Block Diagram

3.2 Vehicle Dynamics Control System

The VDC detailed in this paper performs three tasks: rollover prevention, side slip reduction, and yaw rate following. The VDC determines differential braking pressures for all three objectives and outputs a differential braking pressure to the EBS for each wheel. The differential pressures are determined through proportional feedback control. The braking scheme employed by this VDC design is shown in figure 4. The arrows denote VDC differential braking

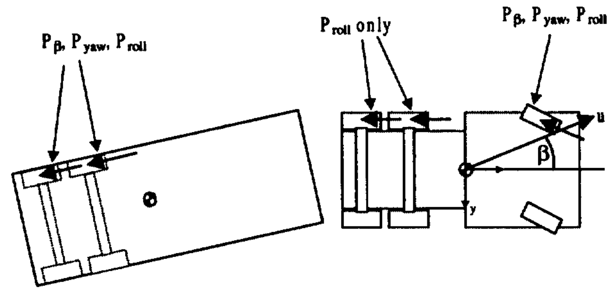

Figure 4. The VDC Braking Scheme

while u and β are the vehicle speed and side slip respectively. Figure 5 depicts the inputs and outputs of the VDC.

The VDC performs rollover prevention by influencing the lateral acceleration and the roll angle. If the following sum is greater than a threshold value, the VDC determines a differential braking pressure:

$$\left|C_1 * \phi + C_2 * \dot{\phi} + C_3 * A_y\right| > Threshold_{roll} \qquad (2)$$

where ϕ, $\dot{\phi}$, and A_y are the roll angle, roll rate, and lateral acceleration, respectively. C_1, C_2, and C_3 are weighting terms for each of these quantities. If this inequality holds, the pressure term is determined by the following equation:

$$\Delta P_{roll} = \left| K_1 * \phi + K_2 * \dot{\phi} + K_3 * A_y \right| \quad (3)$$

where K_1, K_2, and K_3 are control gains. The differential braking is applied to the side of the vehicle which will provide a stabilizing yaw moment.

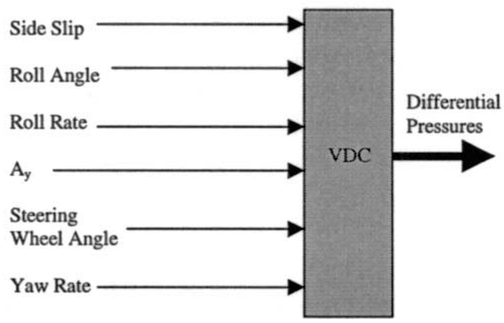

Figure 5. Input/Output Structure of the VDC

The VDC also uses differential braking to make the vehicle yaw rate match the driver's desired yaw rate. Specifically, this control is designed to reduce understeer that may be caused by the stabilizing yaw moment of the roll angle reduction control. This is intended to prevent the vehicle from understeering off the road. This yaw rate following portion of the VDC is not activated until the following three conditions are met:

$$\dot{\psi} * \dot{\psi}_d > 0 \quad (4)$$

$$|\dot{\psi}_d| > |\dot{\psi}| \quad (5)$$

$$|\dot{\psi} - \dot{\psi}_d| > Threshold_{yaw} \quad (6)$$

where $\dot{\psi}$ is the tractor yaw rate and $\dot{\psi}_d$ is the estimated desired yaw rate of the driver. (4) ensures that the VDC is only turned on when yaw rate and the desired yaw rate are of the same sign. This condition is imposed to avoid conflicts with the side slip reduction control. If $\dot{\psi}$ and $\dot{\psi}_d$ are of the same sign it is less likely the vehicle is oversteering. (5) checks that the vehicle is indeed understeering. This will now be the case if the desired yaw rate has greater magnitude than the actual yaw rate. (6) prevents the VDC from activating during normal driving situations when the differences between the actual and desired yaw rates are small.

The desired yaw rate as estimated by the VDC is a function of the steering wheel angle and the vehicle longitudinal velocity. It is calculated with the following equations:

$$\dot{\psi}_d = \delta_s * H(u) \quad (7)$$

$$H(u) = m * u + b \quad (8)$$

where δ_s is the steering wheel angle and u is longitudinal velocity. m and b are constants determined from linear polynomial fit for the ratio of steady state yaw rate to steering wheel angle. The desired yaw rate is then assumed to be proportional to the steering wheel angle as shown in (7). The differential pressure is determined by the following equation:

$$\Delta P_{yaw} = K_{yaw} * |\dot{\psi} - \dot{\psi}_d| \quad (9)$$

where K_{yaw} is the control gain.

This pressure is applied to the inner rear tractor wheels to generate a destabilizing yaw moment, which will increase the yaw rate to reduce understeer.

The side slip reduction control is performed in a similar manner as the yaw rate following and rollover prevention controls. This control utilizes the simultaneous yaw rate/vehicle side slip control design (Zanten et al, 1995). This portion of the VDC is activated when the following two criterion are met:

$$\beta * \dot{\beta} > 0 \quad (10)$$

$$\beta > Threshold_\beta \quad (11)$$

where β and $\dot{\beta}$ are the side slip angle and side slip rate respectively. (10) indicates that β is moving toward regions of instability. (11) ensures that the side slip angles generated in normal driving maneuvers do not activate this portion of the VDC. The differential pressure is calculated according to:

$$\Delta P_{sideslip} = K_\beta * |\beta + \dot{\beta}| + K_{yaw} * |\dot{\psi} - \dot{\psi}_d| \quad (12)$$

where K_β is a control gain.

Note that the yaw rate following pressure term is included in (12). When the conditions are sufficient to activate side slip reduction, the yaw-rate following portion of the VDC is turned off to avoid conflicts. The side slip pressure is applied to the side of the vehicle that will induce understeer, lower lateral acceleration, and reduce the side slip angle. This differential pressure is only applied to the front tractor axle and the trailer axles and not the rear tractor axles since jack-knifing can be caused easily when the rear tractor axles loose grip.

It should be noted that VDC never releases brake pressure since the differential pressure terms are greater than or equal to 0. While vehicle stability in some maneuvers may be improved by reducing brake pressure, it is felt that the VDC would be safer overall if brake pressure was applied rather than reduced.

3.3 Root Locus Analysis

To simplify the controller design process, a linear model of the vehicle was created. This allowed root locus analysis to determine the controller gains listed in equations (3), (9) and (12).

The linear model is based on the linear single unit yaw-roll vehicle model developed by Segel (1956) and

Sampson (1998). This model provides for yaw, roll, and side slip degrees of freedom for the tractor. The trailer has roll and yaw degrees of freedom, but is constrained by the hitch. The tractor and trailer unsprung masses also have roll degrees of freedom. The linear model is matched to TruckSim by linearizing nonlinear vehicle parameters at static equilibrium.

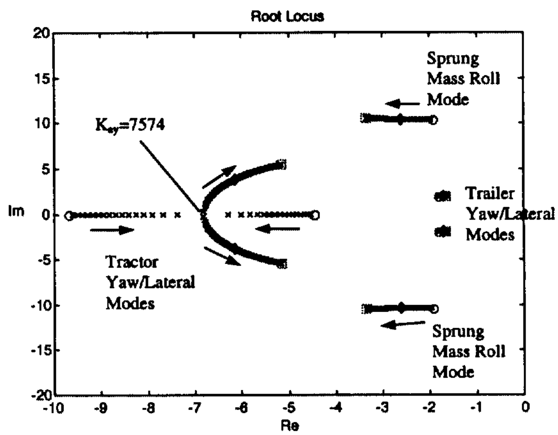

Figure 6. Root Locus Plot for K_{ay}

The root locus analysis is performed by varying the control gains one at a time and plotting the eigenvalues of the system. This process is performed for multiple iterations until the best compromise for all control gains is achieved. Figure 6 shows a root locus plot for varying values of K_{ay}. The arrows indicate the movement of the eigenvalues as K_{ay} increases.

4. Simulation Results

Table 1. Simulation Results

Maneuver	Driver Interaction	VDC On/Off	Maximum Roll Angle	Maximum Lane Error
Step Steer	Open-loop	Off	Rollover	NA
		On	3.5 deg.	NA
Fish Hook	Open-loop	Off	Rollover	NA
		On	3.6 deg.	NA
Double-lane Change	Closed-loop	Off	2 deg.	58 ft.
		On	1 deg.	4 ft.
Ramp-Entering	Closed-loop	Off	Rollover	2.1 ft.
		On	7 deg.	2 ft.

4.1 Open-Loop Input Simulation Results

To study the performance of VDC in representative worst case scenarios without driver interaction, two maneuvers using open-loop steering inputs are shown. The first maneuver is step steering. This maneuver approximates a sudden sharp turn. The second maneuver is the fishhook. The vehicle is steered sharply in one direction then steering is quickly reversed to a greater magnitude in the opposite direction. This maneuver attempts to attain a higher roll angle than step steering for a given steering magnitude.

4.2 Step Steering

Figure 7 shows the vehicle response without VDC to a step steering input of 120 degrees (2.094 rad). Initial vehicle speed is 60 mph. The road surface coefficient of friction, μ, is 0.85 corresponding to dry road conditions. Cruise control is not activated so vehicle speed is allowed to drop during the maneuver. Rollover is defined as the point at which roll angle exceeds 45 degrees. It can be seen that the vehicle quickly exceeds this threshold.

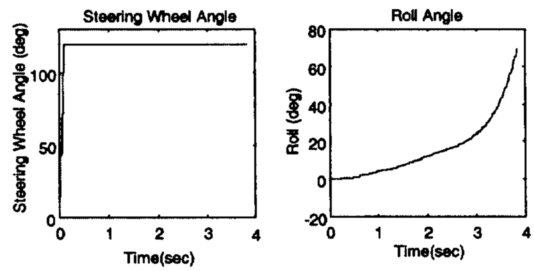

Figure 7. The Response of the Vehicle without VDC to Step Steering

Figure 8 shows the vehicle response when the VDC is activated. The VDC is successful in stabilizing the vehicle. Roll angle does not exceed 4 degrees despite the severity and sudden application of the maneuver.

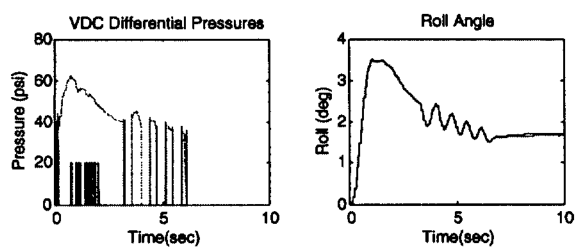

Figure 8. The Response of the Vehicle with VDC to Step Steering

4.3 Fishhook

The fishhook steering input was modified from that used by NHTSA for light vehicle untripped rollover (Garrott et al, 1999). The first steering peak is –270 degrees. Then the steering is reversed to 320 degrees. Vehicle speed is 30 mph. The coefficient of friction is 0.85. No braking is applied but vehicle speed is allowed to drop during the maneuver.

Figure 9 shows the vehicle response with the VDC deactivated. The vehicle rolls over in 4 seconds. Figure 10 shows the vehicle response with the VDC activated. As with the step steering maneuver, the vehicle is stabilized and roll angle is limited to less than 4 degrees.

The VDC system performs well with these open-loop inputs. VDC is able to effectively limit the roll angle in both the step steering and fishhook maneuvers. Rollover is prevented in both cases despite the sudden application of severe steering.

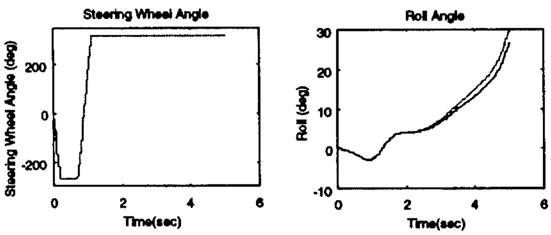

Figure 9. The Response of the Vehicle without VDC to Fishhook Steering

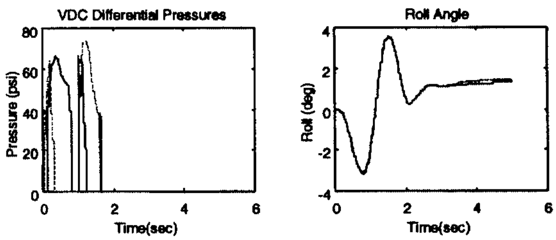

Figure 10. The Responses of the Vehicle with VDC to Fishhook Steering

4.4 Human-in-the-Loop Simulation Results

In this section, the results of simulations using closed-loop driver models are described. The first simulation uses the STI driver model to examine a ramp-entering maneuver. The second simulation uses a lane position with preview driver model to examine a double lane change. While the performance of VDC has been shown to decrease the chance of rollover for maneuvers where steering and braking are open loop with respect to vehicle behavior, it is important to study the interaction between the human and VDC. These simulations show the ability of VDC to achieve the objectives of rollover prevention, side slip reduction and yaw rate control.

4.5 Ramp-Entering

This maneuver is executed using the STI crossover principle driver model. The simulated driver will attempt to follow a constant 300 ft. radius curve. The vehicle's initial velocity is 40 mph. Road friction μ, is 0.85 simulating a dry road. This situation demonstrates VDC's ability to prevent rollover when the driver enters the curve at an unsafe speed. Figure 11 compares the vehicle responses with and without VDC during this maneuver.

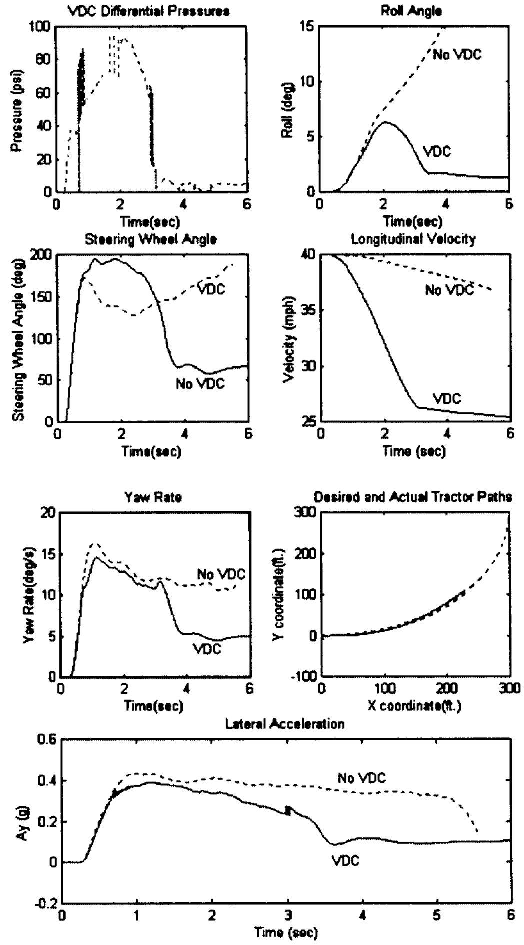

Figure 11. The Responses of the Vehicle During Ramp-Entering

When VDC is switched off, the vehicle is unable to negotiate the curve. The truck rolls over in 4.5 seconds. The lateral acceleration necessary to negotiate the curve at 40 mph is within the traction limit of the vehicle. It is however above the roll threshold. Note that the driver is able to follow the intended path up to rollover. It is desirable that VDC prevent rollover without detrimentally affect the ability of the driver to follow the same curve.

With VDC activated, the vehicle is able to negotiate the curve. Roll angle peaks at 6 degrees and drops to less than 2 degrees for the rest of the maneuver. Note also that the vehicle still follows the intended path and displays good tracking of the desired yaw rate.

4.6 Double Lane Change

The double lane change displays VDC's ability to ensure the vehicle follows the path intended by the driver during an obstacle avoidance situation. The lane position with preview driver model is used to generate steering inputs. To test VDC's ability to maintain vehicle stability while simultaneously following the desired path, a low friction road surface is simulated. For this maneuver μ is 0.2. A braking input from the driver of 15 psi for 4-seconds is used to generate yaw instability during the lane change. The lane change width is 12 ft. Figure 12 shows the responses of the vehicle during this maneuver.

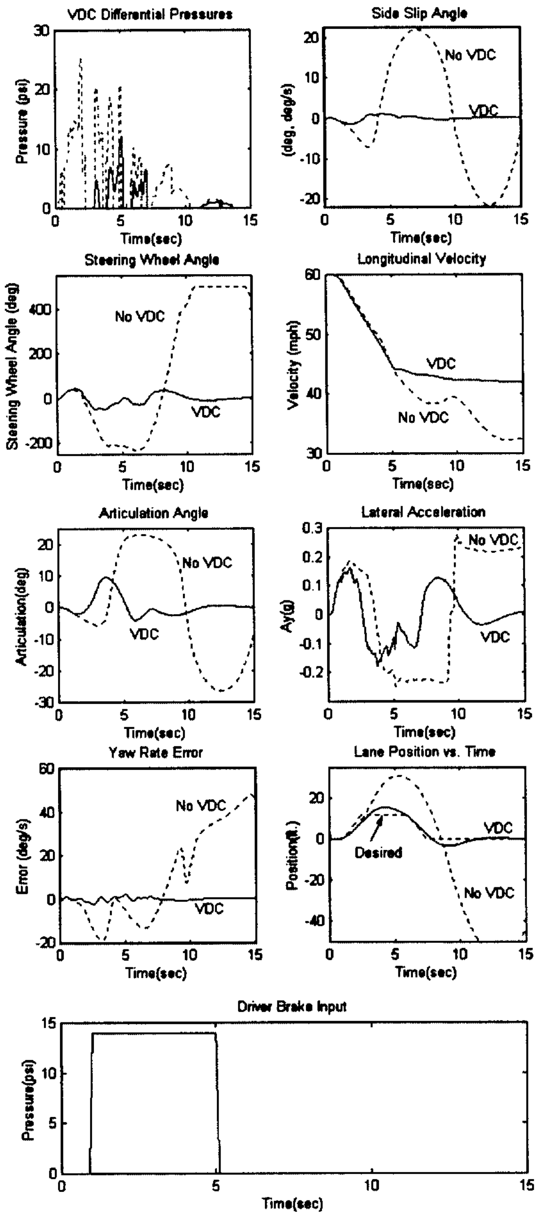

Figure 12. The Responses of the Vehicle during Closed Loop Double Lane Change

Without VDC, the driver model is unable to keep the vehicle close to the desired lane position. The truck overshoots both the initial lane change and the switch back to the original lane. The trailer swings wide during both direction changes and the vehicle nearly jackknifes. The side slip grows to greater than 20 degrees in both directions. The yaw rate does not match the driver's desired yaw rate indicating loss of vehicle control.

When VDC is activated, vehicle control is maintained. The vehicle now tracks the lane position command very well. The side slip angle is kept below 2 degrees. The tractor yaw rate very closely matches the desired yaw rate. Trailer articulation does not exceed 10 degrees and the vehicle successfully completes the maneuver in the proper lane.

5. Conclusions

The VDC system shows the potential to greatly improve the safety of the articulated heavy truck. VDC is successful in avoiding rollover and jack-knife in the maneuvers presented. The control objectives of limiting roll angle, limiting side slip angle, and tracking the desired yaw rate are all achieved by this VDC design.

In representative maneuvers utilizing open-loop steering inputs VDC achieves the goal of reducing roll angle. For both step steering and the fishhook, rollover occurs without VDC and is prevented when VDC is activated.

Simulations utilizing closed-loop driver models also show the benefits of VDC. The ramp-entering simulation demonstrates the ability of the VDC to prevent rollover. While the baseline vehicle rolls over during this maneuver, the VDC equipped vehicle does not. Additionally, VDC does not cause conflicts between the driver's desire to follow the curve and antirollover objective. Its action limits roll but does not cause excessive understeer. The closed-loop double lane change also displays the advantages of VDC. While the baseline vehicle cannot stay on the road and nearly jack-knifes, the VDC equipped vehicle is able to follow the driver's intended path. Yaw rate is well controlled and side slip is limited to safe values.

Design of the VDC system for an articulated heavy truck does present challenges. Due to the vehicle's high polar moment of inertia, generating a yaw moment of the magnitude required to prevent jack-knifing is difficult. Application of brakes can even cause jack-knife if applied incorrectly. Thus the braking scheme chosen for the VDC system must be well thought out to be successful. Despite this difficulty, VDC may be an essential addition to the safety systems of the next generation of heavy trucks.

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