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ROLLOVER PREVENTION FOR SPORTS UTILITY VEHICLES WITH HUMAN-IN-THE-LOOP EVALUATIONS

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Abstract

An anti-rollover control algorithm based on the Time-To-Rollover (TTR) index is proposed in this paper. A simple model with steering and differential braking inputs was developed to calculate the TTR in real-time. The TruckSim dynamic simulation software is used to verify the final control performance, as well as simulating the dynamics in the driving simulator. Both the simple and complex (TruckSim) models were tuned to match the behavior of a 1997 Jeep Cherokee vehicle up to lateral acceleration of 0.6g. The performance of the proposed control system is compared with other threshold-based control algorithms. Finally, a human-in-the-loop experiment is conducted to study the performance of the proposed algorithm under more realistic driving conditions.

1. Introduction

In the U.S., more than 35,000 people were killed in road accidents in 1997 (NHTSA, 1998). About one fourth of these deaths were the result of non-collision crashes. The report also showed that rollover occurred in about 90% of the first harmful events of non-collision fatal crashes. Furthermore, the percentage of rollover occurrence in fatal crashes was significantly higher than other types of crash accidents. Compared to the other types of vehicles, sports utility vehicles (SUV) have the highest rollover rates in all crashes.

Recently, the National Highway Traffic Safety Administration (NHTSA) announced that it would provide additional information (Garrott et al., 1999) about the rollover stability of vehicles in its future safety rating. One major thrust behind this new initiative is the well-publicized rollover incidents of several SUVs and passenger cars (Suzuki Samurai, Isuzu Trooper, Mercedes A-class, and the Mercedes/Swatch SmartCar). Since vehicle safety is a crucial factor influencing consumers' purchasing decisions and government regulations, it seems fair to say that rollover stability is becoming an important element in the overall vehicle safety performance.

Rollover prevention can be achieved by employing rollover warning and anti-rollover systems. Most existing rollover warning systems (Woodrooffe, 1993, Rakheja et al., 1990, Preston-Thomas et al., 1990, Freedman et al., 1992, McGee et al., 1993, Strickland et al., 1997, Winkler et al., 1998) are based on signal threshold techniques. These systems turn on the warning actions when the vehicle roll angle or the lateral acceleration exceeds a pre-selected threshold value. They are usually conservative and do not predict impending rollover danger in the future, which is very important for drivers to correct the dangerous maneuver and avoid the rollover accident before the threshold value is exceeded. To prevent/reduce rollover, one of the most important enabling techniques is the development of accurate rollover threat indices. A rollover warning/control algorithm will work well only if the impending vehicle rollover threat can be accurately represented. The authors proposed a Time-To-Rollover (TTR) metric (Chen et al., 1999a, Chen et al., 1999b) for rollover prevention. In theory, TTR provides a better assessment of impending rollover threat, and thus could be the basis for rollover warning and anti-rollover control algorithms. In (Chen et al., 1999a), it was shown that the TTR for SUVs is too short for human drivers to respond. It is an indication that an active control may be necessary to assist the drivers for rollover prevention. A new anti-rollover control scheme based on the TTR metric is proposed in this paper.

Three types of actuation mechanisms were proposed in the literature for rollover reduction: four-wheel steering (4WS), active suspension, and differential braking. Furleigh et al. (1988) proposed multiple steered axles for articulated heavy trucks. The tractor has one front steering axle and two rear axles. Their algorithm assumes that the steering at the rear axles is proportional to the front steering angle. The proportional gain is defined as a function of vehicle forward speed. The lateral acceleration of the trailer can be reduced at high-speed obstacle maneuvers. Dunwoody (1993) proposed an active roll control system consisting of a hydraulic fifth wheel and active suspension to control the roll motion of the vehicle. It can raise the static rollover threshold by 20-30%. Lin et al. (1994, 1996a, and 1996b) proposed a roll control

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system based on active suspension and lateral acceleration feedback for single-unit and articulated trucks. Their control system was found to reduce the transient and steady state load transfer for a range of maneuvers and increase the rollover safety of the vehicle. Sampson et al. (1998) proposed a more systematic control design by using state feedback and the LQR technique. Palkovics et al. (1998) proposed a Roll-Over Prevention (ROP®) system for commercial vehicles. They utilized the measurements of the wheel speed and the lateral acceleration to estimate wheel lift-off. If a wheel life-off is detected by estimation, ROP® will activate full-braking of the vehicle. Wielenga (1999) proposed an Anti-Rollover Braking (ARB™) which applies differential braking instead of full braking. The rebound bumper contact is used to detect tire lift-off. Whenever the tire lift-off is detected or the lateral acceleration of the vehicle exceeds a threshold value, the differential braking will be activated.

Differential braking is selected as the actuation mechansim for the proposed TTR based anti-rollover control system because of three reasons: (1) 4WS and active suspension are more difficult and expensive to implement. (2) Differential braking is also considered as the most effective way to reduce the lateral acceleration of the vehicle. (3) Differential braking can reduce the forward speed that contributes to lateral acceleration of the vehicle, which cannot be reduced by 4WS and active suspension. The proposed anti-rollover controller will activate the differential braking if the TTR is less than a preset value, for example, 0.5 second. In this paper, the performance of the proposed controller will be compared to the algorithm based on the threshold value of roll angle or lateral acceleration.

Many active safety systems were designed without considering the existence of a driver. The driver's interaction with and perception of the system performance may pose a problem. Although in general a driver may gradually adapt to the active safety system and change his/her behavior, the performance of the human-in-the-loop system is not guaranteed and needs to be studied carefully. The performance of the proposed TTR based anti-rollover control algorithm (which has been designed through simulations) will be studied by using a driving simulator made available to us by the Oakland University (Smid, 1999).

The remainder of this paper is organized as follows: a brief review of the Time-To-Rollover index is presented in Section 2. In Section 3, the TTR based anti-rollover control algorithm is defined and presented. The test condition and the results of a human-in-the-loop evaluation study are presented in Section 4. Finally, conclusions are made in Section 5.

2. Time-To-Rollover Metrics

The tire lift-off is defined as the unacceptable rollover event in this research. To be more precise, when we use the term "Time-To-Rollover" we actually mean "Time-To-Tire-lift-off". This definition will not result in any major change in the overall algorithm development. A more aggressive or conservative roll event definition can be used and the overall design process to be described below will remain the same.

After a "rollover" (more precisely, tire-lift-off) has occurred, a true-TTR can be computed in an after-thought manner. In other words, whenever the vehicle roll angle exceeds the defined threshold value, we can roll back the clock and define a point 0.2 seconds before this tire-lift-off incident to have a "true-TTR" of 0.2 seconds. Ideally, if we can re-construct this TTR index in real-time, and in a predictive manner, the severity of rollover threat can be accurately represented and reported. Based on this index, various warning/control systems can be designed.

A model-based TTR is defined as following: assuming the input (steering angle) stays fixed at its current level in the foreseeable future, the time it takes for the vehicle sprung mass to reach its critical roll angle is defined as the (model predicted) TTR. Under normal driving conditions, the model predictive TTR is usually quite large. For implementation considerations, we can saturate the model predicted TTR at X seconds. In other words, we will integrate the speed-dependent yaw-roll decoupled model for up to X seconds (see Figure 1). If it is found that the vehicle does not rollover, the model-predicted TTR is said to be X seconds.

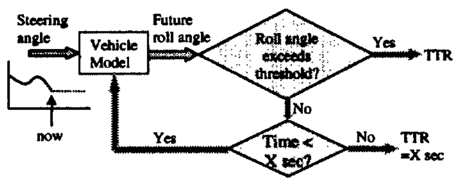

Figure 1. Flow chart for the TTR calculation.

In order to calculate TTR accurately, the simple model obtained in (Chen et al., 1999a) is modified to include the effect of differential braking. The modified yaw-roll model is shown in Figure 2, based on which the model predicted TTR will be calculated throughout this paper.

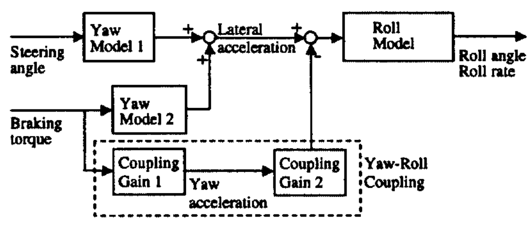

Figure 2. Structure of the modified simple model including the effect of differential braking.

If the braking torque is positive, the computed braking torque is applied to the right front tire. When the braking torque is negative, the computed braking torque is applied to the left front wheel. The discrete-time transfer functions of yaw model 1, 2, and roll model were identified by using standard system identification techniques on vehicle test data. The test data corresponds to a 1997 Jeep Cherokee vehicle and was made available to us by VRTC. The structures of those transfer functions are shown in Equations (1), (2), and (3).

$$T_{yaw1}(z)=\frac{b_0z^2+b_1z+b_2}{z^2+a_1z+a_2} \tag{1}$$

$$T_{yaw2}(z)=\frac{d_0z^4+d_1z^3+d_2z^2+d_3z^1+d_4}{z^4+c_1z^3+c_2z^2+c_3z^1+c_4} \tag{2}$$

$$T_{roll}(z)=\frac{f_0z^3+f_1z^2+f_2z+f_3}{z^4+e_1z^3+e_2z^2+e_3z+e_4} \tag{3}$$

where a_i's, b_i's, c_i's, d_i's, e_i's, and f_i's represent the coefficients of the numerators and denominators of the transfer functions. The structures shown in Eqs.(1)-(3) are motivated by vehicle dynamics and may be gain-scheduled against vehicle speed if needed.

Since TTR predicts the vehicle motion, it can be used to preview the future and thus preventative action is possible. In the following section, an anti-rollover control algorithm is designed and its performance will be compared against control systems based on sensor signals (e.g., lateral acceleration or roll angle).

3. Anti-Rollover Control

3.1 Simulation Model

TruckSim (MSC, 1999) is the simulation model used in this research. TruckSim was a simulation program developed by the Engineering Research Division of the University of Michigan Transportation Research Institute. TruckSim has been commercialized and now can be licensed from the Mechanical Simulation Corporation (MSC). A Jeep Cherokee model is built by using the parameters published by Vehicle Research and Test Center (VRTC) (Salaani et al., 1999). The TruckSim template we used is for a single unit vehicle with two axles. It includes a 14 DOF model with 37 state variables. The dynamic responses of the Cherokee model in TruckSim are verified by the vehicle test data of the 1997 Jeep Cherokee, which was also obtained from VRTC of NHTSA.

The maneuvers used for model verification are constant speed J turn and pulse steer under three steering levels (both right or left) directions at 25 and 50 mph, respectively. The lateral acceleration and roll angle responses of TruckSim under selected test conditions are shown in Figures 3 and 4. The black solid lines represent 10 test runs under the specific maneuver type and the gray solid line represents one run selected from the 10 runs. The steering of the selected run is then input to TruckSim. The gray dot line represents the response of the TruckSim model. As can be seen from Figures 3 and 4, the lateral acceleration and roll angle responses of the TruckSim simulation are quite close to the test data.

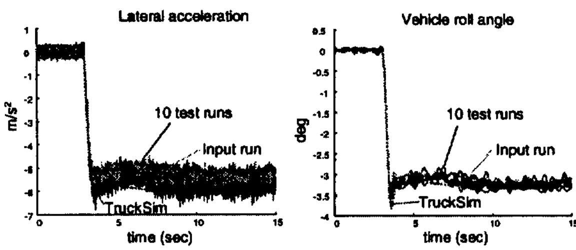

Figure 3. Cherokee model response (25 mph right turn, 0.6g).

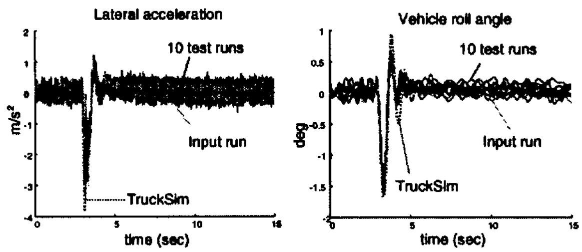

Figure 4. Cherokee model response (50 mph pulse steer right turn).

The root mean square (RMS) values of the lateral acceleration and roll angle response errors of TruckSim are listed in Table 1. Since the responses of TruckSim have been verified against test data with lateral acceleration level of up to 0.6g, TruckSim will be viewed as a full nonlinear complex model accurately representing the Cherokee vehicle

Table 1. RMS values of the lateral acceleration and roll angle response errors of TruckSim compared to the test data of 1997 Jeep Cherokee.

	RMS value
Lateral Acceleration	0.372 m/s2 (0.0379 g)
Roll Angle	0.2130 deg

3.2 TTR based Anti-rollover control

As mentioned in Section 1, differential braking is chosen as the actuation mechanism for anti-rollover control in this research. The effects of differential braking to anti-rollover control are shown in Figures 5 and 6. Figure 5 (Palkovics et al., 1998) shows the effect in the roll plane. By controlling the tire slip ratio within the ABS operation region, we can reduce lateral tire forces. Figure 6 shows the effect in the yaw plane. The lateral expression is also shown in Figure 6. By activating differential braking, we can reduce the lateral tire forces and increase the longitudinal tire forces. In consequence, both $\dot{v}$ and u are reduced. The yaw moment resulting from the longitudinal tire force can also reduce r. Therefore, we can reduce the lateral acceleration of the vehicle. Since the roll motion is excited mostly by the lateral acceleration, the roll angle is also reduced.

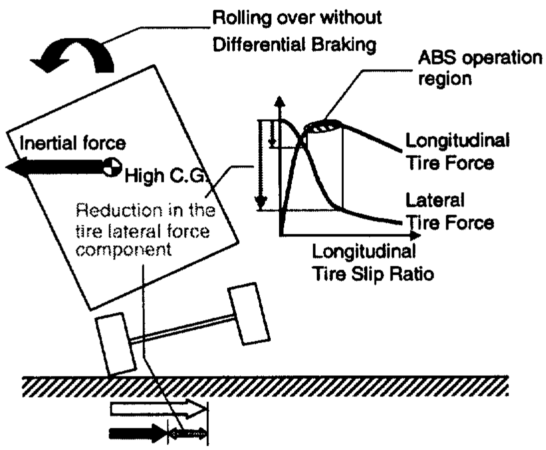

Figure 5. Effects of differential braking (I).

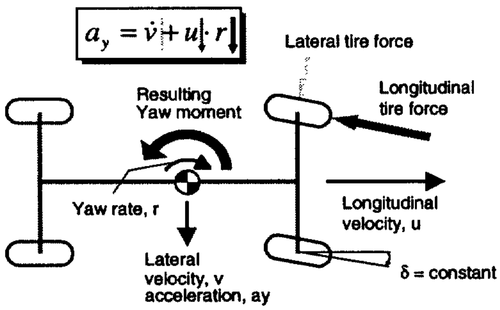

Figure 6. Effects of differential braking (II).

Existing anti-rollover systems (e.g., the ARB system by Wielenga, 1999) usually trigger the control action based on tire lift-off or lateral acceleration threshold. We propose to activate the control action by using the TTR metric because of its predictive nature. The proposed control scheme is shown in Figure 7.

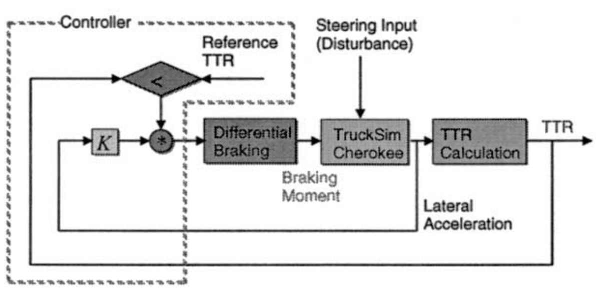

Figure 7. TTR based anti-rollover control

As mentioned in Section 3.1, a Cherokee vehicle model will be used for the verifications of the proposed anti-rollover control algorithm. The driver's steering input is viewed as the only disturbance to the vehicle (i.e., road superelevation and wind gust disturbances are ignored). The reference TTR is set to be equal to the maximum TTR value, which is 0.5 seconds. The roll angle threshold used for calculating the model predicted TTR is set to be 3 degree in this paper. Lateral acceleration of the vehicle is used as the feedback signal. The simple model listed in Section 2 with the brake dynamics is used to design the proportional control gain by using the root-locus technique. The brake dynamics are approximated by a first order system with the time constant of 0.2 seconds (refer to Equation (4)). The block diagram of the overall system used to design the proportional gain is shown in Figure 8.

$$\text{Brake dynamics} = \frac{1}{0.2s+1} \tag{4}$$

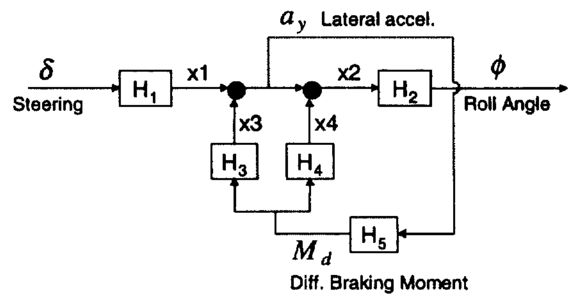

Figure 8. Block diagram of the overall system used to design the proportional gain.

In Figure 8, $H_1 = T_{yaw1}(z)$, $H_2 = T_{roll}(z)$, $H_3 = T_{yaw2}(z)$, $H_4 = -$ yaw-roll coupling, and H_5 = the product of the proportional control gain K and the brake dynamics in discrete-time with a sampling time 0.01 seconds. The transfer function from the steering to the roll angle is shown in Equation (5). The system order of H_{ay} is 11.

$$H_{ay} = \frac{H_1 H_2 (1 + H_4 H_5)}{1 - H_3 H_5} \tag{5}$$

From the root-locus analysis shown in Figure 9, we found K = 130100 is a good design.

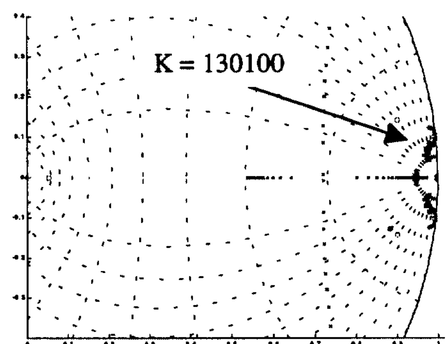

Figure 9. Root-locus analysis of H_{ay} (K = 130100)

The feedback loop can also be designed by feeding back the roll angle. The block diagram of the roll-angle feedback system is shown in Figure 10.

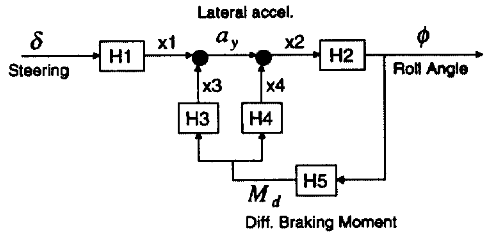

Figure 10. Block diagram of the overall system used to design the proportional gain.

The closed-loop transfer function from steering to roll angle is

$$H_\phi = \frac{H_1 H_2}{1 - H_2 H_3 H_5 - H_2 H_4 H_5} \quad (6)$$

By plugging-in the sub-blocks, it was found that feeding back the roll angle results a higher order closed-loop system than feeding back the lateral acceleration signal. The order of H_ϕ was found to be 19, which is much higher than H_{ay}. Since it is generally much more difficult to design a controller to stabilize higher order systems, we chose to feedback the lateral acceleration signal for designing the proportional control gain.

3.3 Simulation Results

In this subsection, the performance of the TTR based anti-rollover control will be compared to the threshold-based control system with either lateral acceleration or roll angle feedback. Step steering and fishhook steering (see Figure 11) are used to verify the system performance.

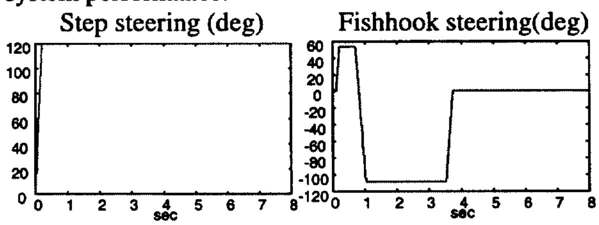

Figure 11. Steering patterns used for the controller verifications.

In this simulation study, the reference TTR is set to be 0.5 sec, the roll angle threshold is set to be 3 degrees, and the saturation limit of the brake is set to be 1000 N-m. For the control algorithms based on the threshold signals, lateral acceleration threshold, Ay, is set to be 0.4g and roll angle threshold, Roll, is set to be 3 degrees. In other words, all three controls use proportional feedback and the control action will be turned on only after the specified threshold value is reached. The main vehicle outputs we examined include maximum lateral acceleration, maximum roll angle, and minimum tire normal force. These three variables are good indication of the severity of vehicle roll. The simulation results for step steering and fishhook steering are shown in Tables 2 and 3, respectively.

Table 2. Step steering evaluation of control performances with differential braking.

Triggered by	None	TTR	Ay	Roll
Fz_min (N)	329.1	641.7	247.21	0
Roll_max (deg)	4.7839	4.7283	5.2932	5.6753
Ay_max (g)	0.85609	0.66003	0.69753	0.77933

Table 3. Fishhook steering evaluation of control performances with differential braking.

Triggered by	None	TTR	Ay	Roll
Fz_min (N)	428.53	617.38	249.56	0
Roll_max (deg)	5.1012	4.8298	5.4325	5.8304
Ay_max (g)	0.8376	0.58455	0.63256	0.74147

As can be seen from Tables 2 and 3, the control action triggered by TTR has the best performance. It can increase the minimum tire vertical load, reduce the maximum roll angle, and reduce the maximum lateral acceleration. For control triggered by Ay or Roll, the results are worse than the vehicle without control in terms of both Fz_min and Roll_max. The control triggered by Roll has tire lift-off for both the step and fishhook maneuvers. Vehicle responses of the TTR based anti-rollover control under the fishhook maneuver are shown in Figures 12 and 13. It seems the proposed rollover control system works very well.

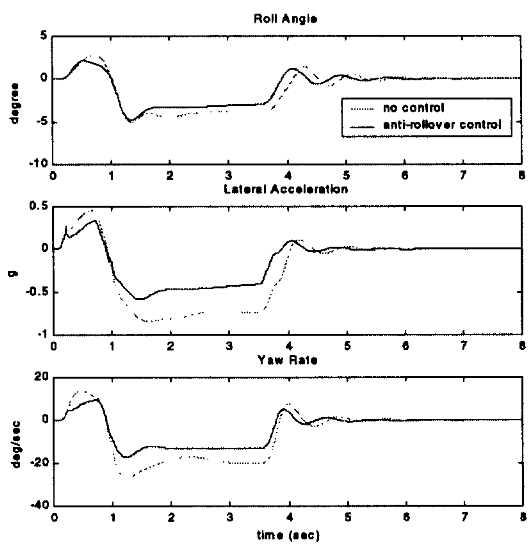

Figure 12. Responses of roll angle, lateral acceleration, and yaw rate of the TTR-based anti-rollover control under fishhook steering.

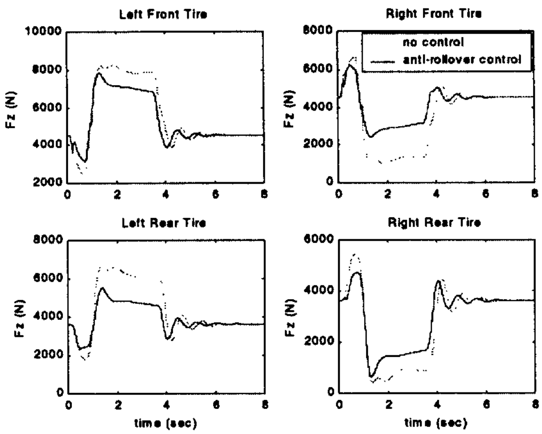

Figure 13. Responses of tire vertical loads of the TTR-based anti-rollover control under fishhook steering.

4. Human-In-The-Loop Evaluation

The performance of the proposed TTR-based rollover control algorithm is further examined by using a human-in-the-loop (HIL) test setup in this section.

4.1 Driving simulator

The driving simulator consists of two parts: the Virtual Vehicle System Simulator (VVSS) of Oakland University is used as the graphical interface and TruckSim of MSC is used to simulate the vehicle dynamics. The experimental setup is shown in Figure 14.

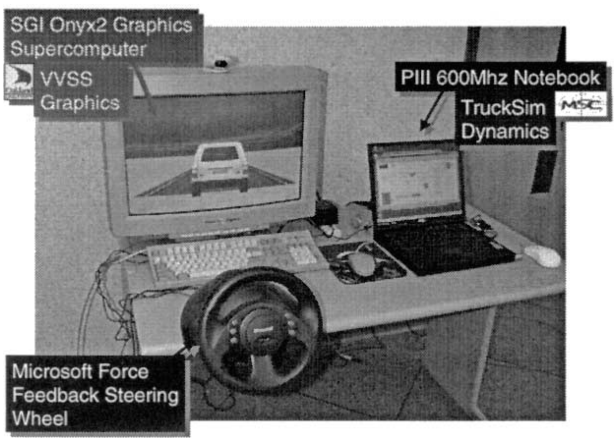

Figure 14. Experimental setup of the driving simulator.

VVSS is a simulation platform with an integrated network environment. VVSS was run on a SGI Onyx2 graphics supercomputer at the University of Michigan's Virtual Reality Lab. TruckSim and the TTR-based anti-rollover control were run on a Pentium III 600 MHz notebook computer. The communication between Onyx2 and the notebook computer was through a 100BT Ethernet network. A Microsoft force feedback steering wheel was connected to the notebook computer as the steering interface. The notebook computer took the test driver's steering input via the game port and sent it to TruckSim. The human driver can interact with the driving simulator though a visualization monitor and the steering wheel. In order to simplify the experiment, the driver is not allowed to brake or accelerate the vehicle.

Humans are known to use multiple feedback cues, including vehicle lateral speed and yaw rate to steer the vehicle. As can be seen from Figure 15 (Koibuchi et al., 1996), the differential braking applied on the front-outer wheel while turning will make the vehicle more understeer. As a result, humans tend to steer more to compensate this additional understeering effect. The saturation of the brake was set to be 250 N-m to prevent excessive interactions.

18 people participated in the evaluation test. Each driver was asked to drive two vehicles: with and without control. Each vehicle is driven through 4 test tracks, which will be described in section 4.2. The vehicle configurations were not revealed to the drivers until the end of the driving test.

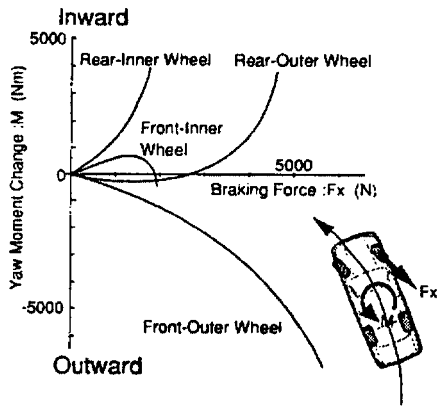

Figure 15. Yaw moment change by braking force for each wheel.

4.2 Test Tracks

The test tracks are designed based on the Man-Off-The-Street (MOTS) course (Rice et al., 1976). The original MOTS course is shown in Figure 16.

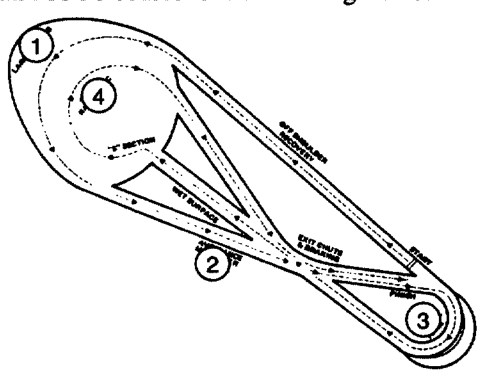

Figure 16. The Man-Off-The-Street (MOTS) course.

Four parts of the MOTS course were selected shown in Figure 17. The 1st test track is a large radius arc, the 2nd test track is an avoidance maneuver, the 3rd test track is modified to be a small radius arc instead of the original gravel turn, and the 4th test track is modified to become similar to the original combination of the "S" section and the small radius arc. The entrance speeds of each test track were designed to be relatively challenging-which is determined subjectively by the authors. They are 85 km/h for the 1st and 2nd test tracks, 75 km/h for the 3rd test track, and 65 km/h for the 4th test track. The screen shots of the test tracks are shown in Figure 18.

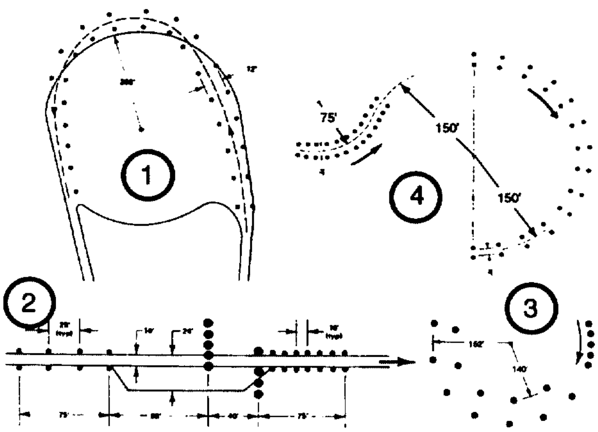

Figure 17. Test tracks used in the human-in-the-loop evaluation.

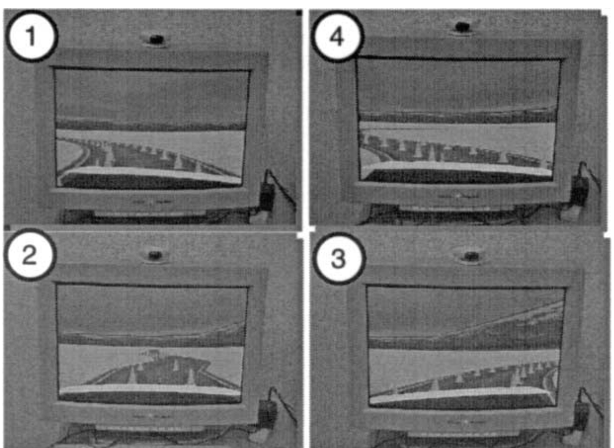

Figure 18. Screen shots of the test tracks.

4.3 Evaluation Results

The driving results are analyzed and the results are summarized in Figure 19. A test run with the TTR-based control was said to be a success if it improves over the no-control vehicle (larger minimum tire vertical load Fz_min, a smaller maximum roll angle Roll_max, or a smaller maximum lateral acceleration Ay_max.). The word "success rate" is calculated from the number of successes divided by 18 tests on the same test track. As can be seen from Figure 19, the proposed control algorithm doesn't perform well with human-in-the-loop. Most of the "success rate" is well below 50%, with the exception of lateral acceleration.

Due to the fact that the human drivers close the loop between the vehicle and the environment, the steering patterns could be quite different for vehicles with or without control, even on the same test track. It seems to be a fairer comparison if the same steering inputs (from the no-control cases) are used to verify the performances. As can be seen from the "steering evaluation" of Figure 19, the control still doesn't perform well except for the avoidance maneuver (test track#2), which is the most severe maneuver. Another interesting result is that the proposed control can reduce Ay_max with a success rate of 100%. This is perhaps related to the fact that the proposed control algorithm uses lateral acceleration as the feedback signal. However, a good performance in Ay_max does not translate into better performance in terms of reducing Roll_max or increasing Fz_min.

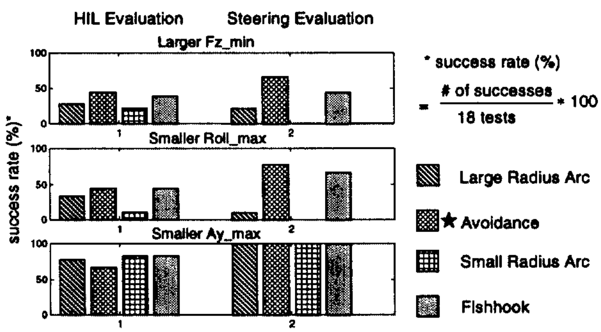

Figure 19. Human-in-the-loop evaluation results.

5. Conclusions

A Jeep Cherokee vehicle model was built in as the target for the proposed anti-rollover control. A Time-To-Rollover (TTR) based anti-rollover control is designed and verified under standard evaluation maneuvers such as step steering (J-turn) and fishhook maneuvers as well as using a root-locus analysis. In simulations, the proposed rollover control algorithm works quite well, compared with existing threshold-based control algorithms, even when a very simple (proportional control) is used in our system. A human-in-the-loop driving test is then performed by using a driving simulator. The proposed TTR based anti-rollover control was found to perform unsatisfactorily. On average, its performance was worse than the vehicle without control. This preliminary result seems to suggest that a more complicated or better-tuned control is necessary.

Acknowledgement

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