ROLL RATE BASED STABILITY CONTROL - THE ROLL STABILITY CONTROL ™ SYSTEM

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ABSTRACT

This paper presents the Roll Stability Control ™ system developed at Ford Motor Company. It is an active safety system for passenger vehicles. It uses a roll rate sensor together with the information from the conventional electronic stability control hardware to detect a vehicle's roll condition associated with a potential rollover and executes proper brake control and engine torque reduction in response to the detected roll condition so as to mitigate a vehicular rollover.

INTRODUCTION

The traditional electronic stability control (ESC) systems aim to control the yaw and sideslip angle of a moving vehicle through individual wheel braking and engine torque reduction such that the desired path of a vehicle determined through the driver's inputs (e.g., steering input) can be maintained. That is, ESC systems help the vehicle to follow the driver's intent such that the driver maintains good control of the vehicle regardless of the variation of road conditions.

Beyond yaw and sideslip control, brake controls in ESC systems have been pursued to mitigate vehicular rollovers in recent years. For example, [1] describes an enhanced system over Driver Stability Control systems for commercial trucks. [2] proposes a stand-alone function called Anti-Rollover Braking (ARB) when an impending rollover of a vehicle is sensed. In [3], engineers from Bosch describe a rollover mitigation function over its ESP system. Continental Teves has developed an Active Rollover Prevention (ARP) system. [4] proposes a Rollover Control Function (RCF). Note that the aforementioned systems use only ESC hardware. In addition to ESC-based brake controls, other chassis control systems have been pursued to mitigate rollovers, see [5], [6], [7], [8] and [9] for more details.

In order to achieve smooth rollover control without sacrificing other vehicle dynamics performance attributes with respect to road and driving condition variations, precise detection or prediction of a potential rollover event is critical. Due to the lack of precise detection of potential rollover conditions and driving conditions such as road bank and vehicle loading, the aforementioned approaches need to conduct necessary trade-offs between control sensitivity and robustness.

In this paper, a system referred to as Roll Stability Control™ (RSC), is presented. Such a system is designed specifically to mitigate vehicular rollovers. The idea of RSC, first documented in [10], was developed at Ford Motor Company and has been implemented on various vehicles within Ford Motor Company since its debut on the 2003 Volvo XC90. The RSC system adds a roll rate sensor and necessary control algorithms to an existing ESC system. The roll rate sensor, together with the information from the ESC system, help to effectively identify the critical roll conditions which could lead to a potential vehicular rollover. Such critical roll conditions need to be discriminated from those due to road bank variations and to be characterized with respect to vehicle loading variations. RSC then applies pressure to the brake(s) on the wheel(s) of the outside of the turn. This reduces lateral force and helps keep the inside wheels firmly on the ground, thus reducing the likelihood of a rollover event.

Although a complete RSC system includes many algorithm modules such as sensor off-set compensation, sensor signal filtering and processing, sensor plausibility, active wheel lift detection, software enhancement of brake hydraulics, longitudinal velocity computation, etc., this paper focuses on vehicle roll dynamics and state estimation as well as the RSC control strategy. Interested readers may find more details on those topics from various patents granted to Ford Motor Company such as (but not limited to) [11],[12],[13],[14],[15] and [16].

This paper is organized as follows. The vehicle roll stability and state estimation are discussed in the next section. The sequential section provides a brief description of vehicle loading estimation. Wheel lift detection is discussed in the next section. The last two sections focus on various RSC control strategies and the conclusions.

VEHICLE ROLL DYNAMICS SENSING AND STATE ESTIMATION

Vehicular roll instability (rollover) is the condition where a vehicle has divergent roll motion along its roll axis.

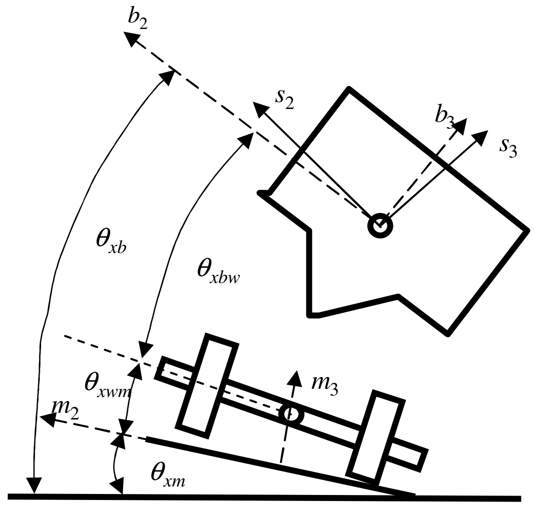

Figure 1. The roll angle definitions for a vehicle driven on a banked road.

Consider a vehicle driven on a general road surface. Figure 1 shows a rear view of the vehicle. Its roll instability can be identified and characterized by using the vertical travel of the wheel centers with respect to the smooth road surface. That is, it is said to be roll instable if it has sustainable two wheel lift from the road surface (both wheels are on the inside of a turn).

The roll instability can also be determined by using various roll information. In order to define the various roll angles, we define two coordinate systems: a body-fixed coordinate system b with axes b_1, b_2 and b_3 (called the body frame) and a road coordinate system m with axes m_1, m_2 and m_3 which is attached to the road surface but moves and yaws with the vehicle body (called a moving road frame). The roll angle of the vehicle body with respect to the sea level is denoted as θ_{xb}, the road bank angle with respect to the sea level is denoted as θ_{xm}, the roll angle between the wheel axle and the road surface is denoted as θ_{xwm} (which is called a wheel departure angle), and the roll angle between the body and the axle of the wheels is denoted as θ_{xbw} (which is called a chassis roll angle).

The critical roll angle defining a potential rollover event is the relative roll angle θ_{xbm} between the vehicle body and the moving road, which is defined as

$$\theta_{xbm} = \theta_{xbw} + \theta_{xwm} \quad (1)$$

or

$$\theta_{xbm} = \theta_{xb} - \theta_{xm} \quad (2)$$

If the magnitude of θ_{xbm} is greater than a threshold for a certain duration, the vehicle is likely to be roll-instable.

The relative roll angle θ_{xbm} may be determined through laser height sensors which measure the distances of the vehicle body at the sensor mounting locations from the road surface along the direction of the laser beams. However using them in mass production for rollover detection purpose is generally cost prohibitive with the current technology. Therefore using the other sensors equipped with the vehicle is desired.

			RSC→→→→→→→→→→→→→	
		ESC→→→→→→→→→→→→→→→→→→→→→		
	Traction Control→→→→→→→→→→→→→→→→→→→→			
ABS→→→→→→→→→→→→→→→→→→→→→→→→→→→→→→→				
1988	**1998**	**2001**	**2003**	**2005 -**
Key Sensor Inputs: Wheel Speeds	Key Sensor Inputs: Same as ABS	Key Sensor Inputs: Steering Wheel Angle, Yaw Rate, Lateral Acceleration, MC Pressure, Longitudinal Acceleration	Key Sensor Inputs: Roll Rate The RSC System was first available on the 2003 Volvo XC90	AdvanceTrac® with RSC is standard on Explorer, Mountaineer, Aviator and Navigator since the 2005 MY, E-350 Econoline Extended Passenger vans since the 2006 MY, and Expedition, Escape, Edge, Explorer SportTrac, Lincoln MKX, Mazda CX – 9, Land Rover LR2, Mazda Tribute and Mercury Mariner for 2007 MY

Figure 2. The evolution of the sensors used in vehicle stability control systems.

The sensor set used in the RSC system evolved from the initial sensors equipped on an anti-lock brake system (ABS), see Figure 2. It includes a centralized motion sensor cluster called the RSC sensor cluster, a steering wheel angle sensor, four wheel speed sensors, a master cylinder pressure sensor, etc. The RSC sensor cluster adds a roll rate senor to the ESC motion sensor cluster, i.e., it is composed of a roll rate sensor, a yaw rate sensor, a lateral accelerometer and a longitudinal accelerometer, which are packaged together along the three orthogonal directions.

Since the measuring directions of the RSC sensor cluster do not always coincide with the directions of the body frame b, it is necessary to define a sensor frame s. The angular differences between frame b and frame s are called the sensor misalignments, which are usually generated due to the mounting errors when the RSC sensor cluster is attached to the vehicle body. Although the sensor misalignments are relatively small, they may need to be corrected in order to avoid potential signal contamination. In addition to the sensor misalignments, oftentimes the misalignment between the vehicle body and the road surface due to unevenly distributed loading inside the vehicle may also need to be corrected. For example, a vehicle with heavy loading near the rear axle might cause the RSC sensor cluster to be tilted with a pitch angle relative to the road surface. These misalignments can be conditionally determined based on the sensor and the calculated signals and the driving conditions.

The kinematics of the RSC sensor cluster can be expressed as in the following equations after small angle approximations and neglecting the vehicle's vertical velocity [17]

$$\begin{aligned} \dot{\theta}_{xs} &\approx \omega_{xs} + \omega_{zs}\theta_{ys} \\ \dot{v}_{xs} &\approx a_{xs} + \omega_{zs}v_{ys} + g\theta_{ys} \quad (3) \\ \dot{v}_{ys} &\approx a_{ys} - \omega_{zs}v_{xs} - g\theta_{xs} \end{aligned}$$

where ω_{xs} and ω_{zs} are the angular rates along the longitudinal and vertical directions, a_{xs} and a_{ys} are the longitudinal and lateral accelerations of the origin of the sensor frame attached to the RSC sensor cluster, v_{xs} and v_{ys} are the longitudinal and lateral velocities of the origin of the sensor frame. θ_{xs} and θ_{ys} are the roll and pitch angles of the sensor frame with respect to the sea level. Notice that v_{xs} in (3) can be related to the vehicle reference velocity calculated based on the wheel speed sensor signals.

Based on (3), it is not hard to find the following:

(i) the roll rate sensor only provides global information of the sensor frame with respect to the sea level, $\dot{\theta}_{xb}$ as in Figure 1, which cannot be directly used as a control variable to drive the RSC system;

(ii) the global roll and pitch angles can be determined from the accelerometers if the lateral velocity v_{ys} is known, however in reality, it is unknown;

(iii) the lateral velocity or sideslip angle can be determined from the lateral acceleration sensor signal if the global roll angle can be determined;

(iv) the roll rate sensor will have non-zero output even if there is no roll attitude change when there is yaw rate on a pitched road.

Since there are uncertainties in the roll rate sensor signal and in the computation of the pitch angle, direct integration of the first equation in (3) is not practical due to the potential of integration drift. Therefore, in order to use the roll rate sensor information to determine critical roll angles and roll conditions used for RSC, various computations are required.

Chassis Roll Angle Estimation

Let's first consider computing the roll angle θ_{xbw} between the body-fixed frame and the axle of the wheels, which is called the chassis roll angle.

Let F_{yf} and F_{yr} be the resultant forces along the lateral direction of the RSC sensor cluster but applied to the vehicle body through the front and rear roll centers of the vehicle. Let the vertical distance from the vehicle body c.g. location to the front and rear roll centers be h_f and h_r. Let l_{s2cg} be the longitudinal distance between the origin of the RSC sensor cluster and the c.g. of vehicle body. Using Newton's law in the sensor frame s, we obtain the following equations of motion

$$M_s(a_{ys} + l_{s2cg}\dot{\omega}_{zs}) = F_{yf} + F_{yr}$$
$$I_z\dot{\omega}_{zs} = F_{yf}b_f - F_{yr}b_r \quad (4)$$
$$I_x\dot{\omega}_{xs} = F_{yf}h_f + F_{yr}h_r - K_{roll}\theta_{xbw} - D_{roll}\dot{\theta}_{xbw}$$

where I_x and I_z are the moments of inertia of the vehicle body with respect its longitudinal and vertical body axes; K_{roll} and D_{roll} are the equivalent roll stiffness and damping rate for the suspension system; b_f and b_r are the distance of the vehicle body c.g. to the front and rear axles with $b = b_f + b_r$.

Based on equations in (4) and using the Laplace transformation, the chassis roll angle can be computed as in the following

$$\theta_{xbw} = T_1(s)a_{ycgs} + T_2(s)\omega_{sx} + T_3(s)\omega_{zs} \quad (5)$$

where $T_1(s), T_2(s)$ and $T_3(s)$ are three transfer functions which can be obtained through the inertia parameters, their formulas can be found in [18], and

$$a_{ycgs} = a_{ys} + l_{s2cg}\dot{\omega}_{zs}$$

is the lateral acceleration of the vehicle body at its c.g. location but projected along the lateral direction of the frame s.

Notice that the above calculated chassis roll angle is based on a linear model with a fixed vehicle body roll axis, hence it will deviate from the true value if the vehicle has wheel lift and if the vehicle enters into the nonlinear suspension operation region. Such a computation can be sensitive to the vehicle's loading due to the variation of the center of gravity and roll moment of inertia. However if there is no wheel-lift, θ_{xbw} closely models the true relative roll angle between the vehicle body and the road if the vehicle parameters, such as the sprung mass and height of the c.g. are accurate. Hence a small magnitude of θ_{xbw} is a good indication of a roll-stable situation.

Global Roll Angle Estimation

The aforementioned chassis roll angle will be saturated when one side of the vehicle is about to lift from the ground due to the suspension saturation and it is independent of the wheel departure angle θ_{xwm}. Therefore θ_{xbw} can no longer characterize the relative roll between the vehicle body and the road during a potential rollover event.

In order to overcome this, a roll angle based on the roll rate sensor signal is pursued. Based on the analysis before, roll angle obtained through the roll rate sensor is a global roll angle and includes various components as shown in Figure 1.

Since θ_{xs} computed based on the roll rate sensor signal is the sum of the road bank, the wheel departure angle, and the chassis roll angle, it provides a means to confirm certain variables if the other variables are known. On the other hand, if the vehicle is driven on level ground without wheel lift, the global roll angle θ_{xs} matches the chassis roll angle θ_{xbw}. Such a global roll angle can also be used in determining the road camber status which could have a significant influence on the roll stability of the vehicle.

As mentioned before, there are various uncertainties when trying to capture the velocity of the global roll angle θ_{xb}. Denote the uncertainties due to sensor off-sets, drifts and misalignments in roll and yaw rate sensors as $\Delta\omega_{xs}$ and $\Delta\omega_{zs}$, the chassis pitch angle due to suspension motion as θ_{ybw} and a steady state characterization of the global pitch angle as θ_{ybss}, then the velocity of the global roll angle can be related to the estimated value from the sensor signal $\dot{\hat{\theta}}_{xb}$ as

$$\dot{\hat{\theta}}_{xb} = \dot{\theta}_{xb} - \Delta\dot{\theta}_{xb} \quad (6)$$

and the uncertainties $\Delta\dot{\theta}_{xb}$ can be expressed as

$$\Delta\dot{\theta}_{xb} = \Delta\omega_{xs} + \omega_{zs}\theta_{ybss} + \Delta\omega_{zs}\theta_{ybss} + \Delta\omega_z\theta_{ybw} \quad (7)$$

And $\dot{\hat{\theta}}_{xb}$ can be calculated from the known variables as in the following

$$\dot{\hat{\theta}}_{xb} = \omega_{xs} + \omega_{zs}\theta_{ybw} \quad (8)$$

where θ_{ybw} is the chassis pitch angle (see [6] for detail).

If the steady state capture of the vehicle body's global pitch angle θ_{ybss} can be estimated, such as in [19],

$\Delta\dot{\theta}_{xb}$ and $\dot{\hat{\theta}}_{xb}$ can be alternatively computed as in the following

$$\begin{aligned}\Delta\dot{\theta}_{xb} &= \Delta\omega_{xs} + \Delta\omega_{zs}\theta_{ybss} + \Delta\omega_{z}\theta_{ybw} \\ \dot{\hat{\theta}}_{xb} &= \omega_{xs} + \omega_{zs}(\theta_{ybw} + \theta_{ybss})\end{aligned} \quad (9)$$

Since the uncertainties in $\Delta\dot{\theta}_{xb}$ defined in (7) are usually dominated by low frequency content, an anti-drift integration filter $T_{adi}(s)$ is used to integrate $\dot{\hat{\theta}}_{xb}$ to obtain the dynamic content of the true global roll angle. Notice that, in critical roll instable situations, such a roll velocity $\dot{\hat{\theta}}_{xb}$ defined in (8) or (9) together with $T_{adi}(s)$ can be used to characterize the roll conditions that might lead to a potential rollover.

Since $T_{adi}(s)$ removes both the low frequency content of the uncertainty and the low frequency content of the true global roll angle, a steady-state recovery term is used. This leads to the following estimation of the global roll angle

$$\hat{\theta}_{xb} = T_{adi}(s)\dot{\hat{\theta}}_{xb} + T_{ss}(s)\theta_{xbss} \quad (10)$$

where θ_{xbss} is the steady state capture the roll angle. One computation of θ_{xbss} is

$$\theta_{xbw} + \theta_{xwm}$$

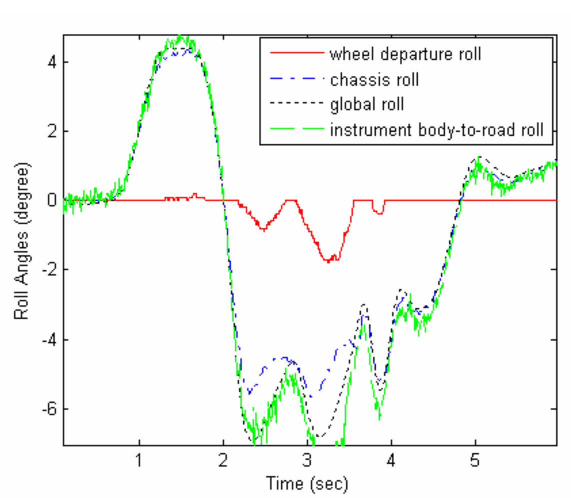

Figure 3. A comparison of the measured roll angle and the calculated roll angle when the vehicle is driven on a level ground.

Another computation of θ_{xbss} is the global roll angle from the 3rd equation of (3) by setting $\dot{v}_{ys} = 0$ or setting $\dot{v}_{ys}$ to the computation generated from the linear sideslip angle. Further details regarding the computation of θ_{xb} can be found in [20].

Figure 3 provides a comparison between the computed global roll angle estimation $\hat{\theta}_{xb}$ using (10) and the relative roll angle between the vehicle body and the moving road using laser height sensors, for a vehicle driven on level ground during a lane change maneuver. Since the road is level, the bank angle θ_{xm} of the moving road is zero. Hence the global roll angle should match the relative roll angle between the body and the road.

Wheel Departure Angle Estimation

The global roll angle together with the chassis roll angle discussed in the previous sections can quantify the axle angle, which is the sum of the road bank angle and the wheel departure angle, but can not determine the magnitude of each.

By utilizing the roll dynamics of the vehicle and wheel lift detection methods to be described later, the conditional determination of the wheel departure angle is obtained.

Let's denote the axle velocity as

$$\dot{\theta}_{xaxle} = \dot{\hat{\theta}}_{xb} - \dot{\theta}_{xbw} \quad (11)$$

then the velocity of the wheel departure angle is

$$\dot{\theta}_{xwm} = \dot{\theta}_{xaxle} - \dot{\theta}_{xm} \quad (12)$$

Integrating (12) gives

$$\theta_{xwm}(t) = \int_0^t \dot{\theta}_{xaxle}(\tau)d\tau - \theta_{xm}(t) \quad (13)$$

Since θ_{xwm} becomes non-zero when there is wheel lift, it is obvious that the integration should be conducted whenever wheel lift is initiated. Assume at time instant t_0, there is a detected wheel lift. Let the road bank angle at time t_0 be θ_{xm0}. Then, (13) implies

$$0 = \int_{0}^{t_0} \dot{\theta}_{xaxle}(\tau)d\tau - \theta_{xm0} \quad (14)$$

At time instant t such that $t_0 \le t \le t_f$ (t_f is the time instant when the lifted wheels come back in contact with the road surface), we subtract (14) from (13) and obtain the following

$$\theta_{xwm}(t) = \int_{t_0}^{t} \dot{\theta}_{xaxle}(\tau)d\tau - [\theta_{xm}(t) - \theta_{xm0}] \quad (15)$$

If the vehicle is driven on level ground or on a constant road bank, (15) leads to

$$\theta_{xwm}(t) = \hat{\theta}_{xwm}(t) = \int_{t_0}^{t} \dot{\theta}_{xaxle}(\tau)d\tau \quad (16)$$

Notice that $\hat{\theta}_{xwm}$ is a good approximation of θ_{xwm} if the change in road bank is small, i.e., if

$$\Delta\theta_{xm}(t) = \theta_{xm}(t) - \theta_{xm0} \quad (17)$$

is close to zero or negligible with respect to $\hat{\theta}_{xwm}$. This is true for the following conditions:

(i) the vehicle is driven on a level ground;

(ii) the vehicle is not driven on a transient road bank;

(iii) during the time when there is wheel lift the road bank doesn't change much in comparison with the road bank at the time when the wheel lifting starts;

(iv) during the time when there is wheel lift, the vehicle is driven very aggressively such that the roll velocity due to the road bank is much smaller than the roll velocity due to the wheel departure and the chassis roll.

Notice that the afore-mentioned cases cover a large portion of the scenarios where wheel lift could occur, especially since wheel lift is often short in duration (typically less than 1 second). During this time the magnitude of change of the road bank is typically very small. Therefore, the magnitude of change in road bank should be much less than the magnitude of $\hat{\theta}_{xwm}$. A detailed computation regarding wheel departure angle can be found in [21].

Figure 3 shows the computed chassis roll angle, global roll angle, wheel departure angle and the instrumented roll angle between the body and the moving road for a vehicle driven on level ground in a double lane change maneuver (with detuned control). It is not hard to see that the wheel departure angle fills the gap between the true relative roll between the body and the moving road, and the chassis roll angle.

Road Bank Angle Estimation

The relative roll angle between the vehicle body and the road surface can be computed based on (1) using the variables calculated in the previous sections and it can also be computed based on (2) using the road bank angle information. The advantage of using (2) is that it relies on the known characterization of the road bank based on the computed variables and its influence on the vehicle's roll tendency.

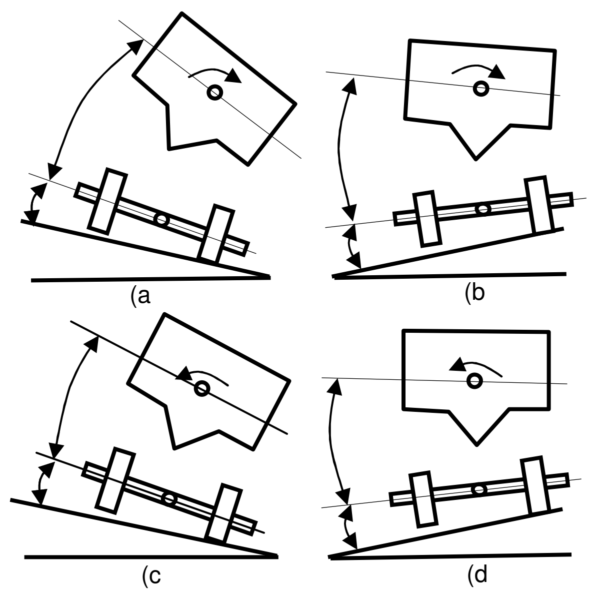

Figure 4. The 4 cases for a vehicle turning left on a banked road.

Figure 4 shows the 4 patterns of the interaction between the vehicle dynamics and the road bank when a vehicle turns to the left on banked roads. (a) and (c) are the off-camber turns and (b) and (d) are on-camber turns.

In the off-camber turns, (a) indicates the worst case scenario where the vehicle roll motion is amplified by the road bank, while in (c) the vehicle rolls in the opposite direction of the road bank, hence the vehicle has less tendency to rollover. In the on-camber turns (b) and (d), the vehicle roll motions are either reduced or increase in the direction which does not cause rollover at all.

Based on the computed wheel departure angle, chassis roll angle and the global roll angle, and the physical meaning of road bank, a road bank adjustment in order to generate favorable control variable θ_{xbm} for RSC using (2) can be conducted as in [22,23].

Rear Sideslip Angle Estimation

Based on the third equation in (3), the lateral velocity of the vehicle at the origin of the sensor frame can be calculated if the global roll angle is available. Further analysis shows that such a lateral velocity is the only unknown if using the RSC sensor cluster signals, which satisfied a second order differential equation without involving the other unknowns such as the global roll and pitch angles. Therefore, using the RSC sensor set the lateral velocity can be computed which is robust to road bank and slope and the driving conditions, see [24] for a detailed discussion.

The sideslip angle defined at the rear axle of the vehicle can be determined as in the following

$$\beta_{ra} = \frac{v_{ys} - \omega_{zs} l_{xs2ra}}{\max(v_x, \underline{v}_x)} \quad (18)$$

where $\underline{v}_x$ is the minimum lateral velocity threshold and l_{xs2ra} is the distance between the sensor location and the rear axle location.

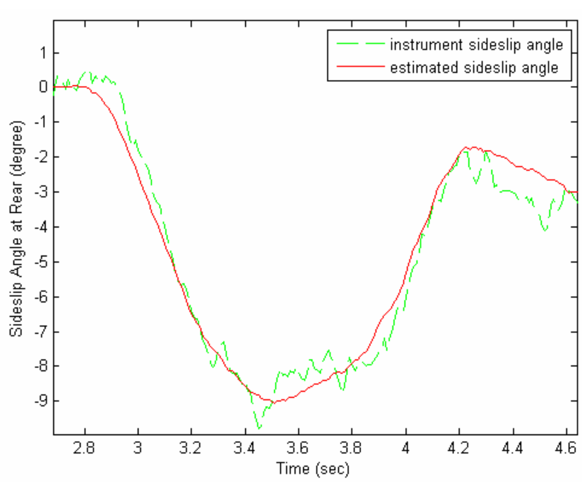

Figure 5. A comparison of the calculated sideslip angle and the measured sideslip angle.

Figure 5 shows a comparison between the measured sideslip angle and the calculated sideslip angle during a maneuver. The measured sideslip angle is calculated based on the velocity sensors equipped with the vehicle which measures the longitudinal and lateral velocities of the vehicle body at the velocity sensor mounting location.

VEHICLE LOADING DETECTION

One of the important control variables used in the RSC system is the relative roll angle θ_{xbm} between the vehicle body and the road surface, which directly measures the potential of a rollover event. Such an angle can be computed as in (1). Hence the accuracy of the chassis roll angle θ_{xbw} can influence the RSC control performance.

Since chassis roll angle is calculated through a linear roll model, the parameters used in the model are functions of characteristics such as the height of the c.g. and the sprung mass. One challenge with using these parameters in computing θ_{xbw} is that they vary with the vehicle loading conditions.

For example, a 150 pound roof load for a typical SUV with a curb weight of 5000 pounds may cause a 30% error in the chassis roll angle calculations if computed assuming no load. Note that a 150 pound load accounts for only a 3% mass variation over the vehicle curb weight. If the above parameters are fixed at certain nominal values in the RSC system, it is conceivable that optimal control performance may not be achieved under a different loading condition. For example, if the parameters in the chassis roll angle model are determined based on nominal vehicle loading condition assumptions, without considering variations due to loading, the chassis roll angle may be under estimated for vehicles with load that raises the c.g. On the other hand, if the parameters in the chassis roll angle model are determined based on a certain loading condition that raises the c..g., it may be over estimated for vehicles without load.

In order to improve the overall performance of the RSC system, it is desirable to estimate and update the vehicle parameters periodically or adaptively adjust them in real time based on the actual behavior of the vehicle.

The loading condition of the vehicle can be determined based on the fact that during level road driving the chassis roll angle must match the vehicle's global roll angle when the vehicle doe not have wheel lift.

By equating (5) and (10), the composite parameters used to determine the chassis roll angle can be learned through a real-time least-square parameter identification algorithm. Such information is used to adjust the feedback control gains so as to request more aggressive brake pressure when appropriate.

WHEEL LIFT DETECTION

In order to confirm when the vehicle wheels are firmly on the ground and when the vehicle has wheel lift, wheel-lift detection is conducted in RSC. Wheel-lift status is also used in estimating wheel departure angle by determining when to conduct the integration in (16). The wheel lift detection includes an active wheel lift detection (AWLD) logic and a passive wheel lift detection (PWLD) logic. The integrated wheel lift detection (IWLD) integrates AWLD and PWLD to provide the final wheel-lift status. The wheel lift status for each wheel is set to one of 5 levels which assume values of 2, 4, 8, 16 and 32 that indicate the wheel being absolutely grounded, possibly grounded, no indication, possibly lifted and absolutely lifted, respectively.

AWLD is used to determine if a wheel is lifted or grounded by checking the wheel rotation in response to a given brake pressure. More specifically, it sends a small brake pressure to an inside wheel, then checks the response of that lightly braked wheel. If the vehicle lateral acceleration sensor indicates a hard cornering of the vehicle on a high mu surface and the inside wheel experiences a longitudinal slip ratio larger than a threshold in response to a relatively small brake pressure, then this inside wheel is likely to be lifted from the ground. Due to the reactive nature of this strategy, a lifted conclusion based on AWLD suffers a potential time delay.

The intent of PWLD is to determine if a wheel is lifted or grounded by checking the vehicle dynamics and wheel speed behavior without actively requesting brake pressures. Namely, it passively monitors the wheel speeds together with the other key vehicle dynamics variables to determine if the speeds indicate a potential wheel lift condition.

In order to capitalize on the benefits of AWLD during steady-state driving conditions and the instantaneous nature of PWLD during dynamic maneuvers, an integration of AWLD and PWLD is required. Figure 6 illustrates such an integration. A detailed description of the above wheel lift detection methods can be found in [25].

Figure 7 shows the final wheel lift detection status for a wheel during a J-turn maneuver with a detuned control. The brake pressure due to the AWLD request and the wheel speed response are also included in the figure.

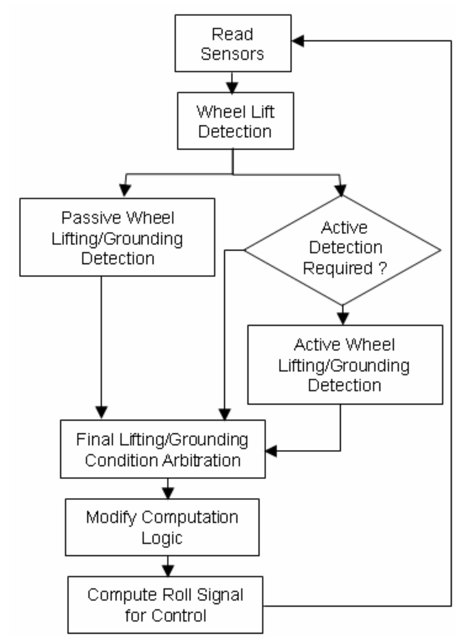

Figure 6. The integration between AWLD and PWLD.

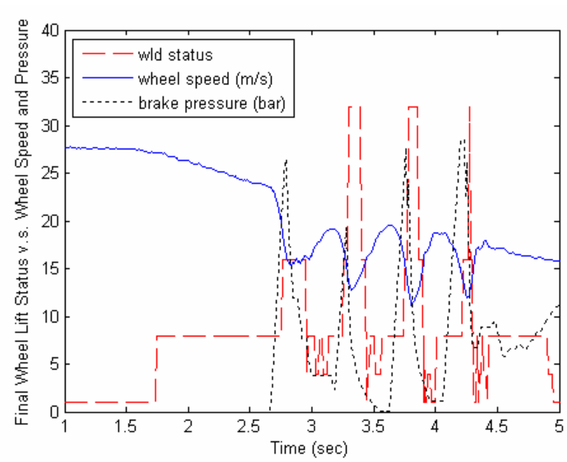

Figure 7. The wheel lift detection flag for an inside wheel during a J-turn maneuver (with detuned control).

RSC CONTROL STRATEGY

The RSC control strategies are designed to utilize all the available information to overcome the delays in the brake hydraulics and to provide effective brake torques to counteract the vehicle body roll motion which may lead to a rollover. It includes a Transition Control module which performs control for the transitional portion of a dynamic maneuver, and a Quasi-steady State Feedback Control which performs control for less dynamic maneuvers. The interaction between those two strategies provides an effective control for mitigating vehicular rollovers, see [26], [27], [28] and [29] for more details.

Transition Control

In order to execute the RSC function, a large brake pressure is requested on the front outside wheel during potentially roll-instable events. When the RSC function requests the maximum pressure build rate, significant delays in brake pressure buildup can occur due to limitations in the hydraulic capabilities. Therefore, if a brake pressure buildup is requested after the roll instability is underway, there may not be sufficient time to build an adequate control pressure to mitigate the roll-instable event. To deal with such a brake pressure build delay, the first control strategy used in the Transition Control module is a feedforward control that is used to pre-charge the hydraulic system. Such a feedforward control utilizes the prediction information based on the driver's steering and the other vehicle state information to provide a pressure build prior to the roll instability. Note that this pre-charge is designed to minimize pressure build delay, and therefore is a relatively small pressure to overcome the inertia in the brake controls pump and to reduce the caliper knockback.

The other control strategy used in the Transition Control module is a feedback control which is the coordination and combination of three feedback control commands based on three different control signals so as to achieve three different control objectives.

One of the feedback control signals used in RSC is θ_{xbw}. The brake pressure command from θ_{xbw} uses a PD feedback control where the control gains and deadbands are functions of various measured and computed signals. Notice that θ_{xbw} is adjusted to adapt to various vehicle loading conditions. Since for sufficiently aggressive transitional maneuvers, the roll momentum can result in a lifting of the center of gravity of the vehicle at the end of the transition. It is an objective of this θ_{xbw} based PD feedback control to introduce effective roll damping before the occurrence of wheel lift by rounding off the buildup of lateral force when needed as it approaches its peak level in the final phase of the transition.

Due to the limitation in hydraulic capabilities, a leading indicator of θ_{xbw} is needed to effectively utilize the roll feedback so as to sufficiently mitigate potential rollovers. Therefore another control signal used in the Transition Control module is the model-based linear sideslip angle, β_{falin}, at the front axle, which is the front tire lateral force divided by the front tire cornering stiffness

$$\beta_{falin} = \frac{F_{yf}}{C_f} \quad (19)$$

where F_{yf} is the front cornering force which can be obtained from (4) and C_f is the cornering stiffness for the front wheels.

The control based on β_{falin} significantly leads the θ_{xbw} control. However, β_{falin} also has the potential to be relatively erratic, potentially leading to a premature reduction in control effort. Therefore, a robust signal is needed to fill in the resulting control gap between β_{falin} and θ_{xbw} control. A yaw rate-based PD controller can accomplish this. Notice that such a yaw rate-based PD control also provides adequate yaw damping to minimize the occurrence of excessive yaw rate overshoot in limit maneuvers, which further reduces the occurrence of excessive sideslip angle and lateral forces that significantly exceed the steady state cornering capacity of the vehicle. Hence it can increase the roll stability margin of the vehicle especially during aggressive maneuvers. A goal in such a yaw rate-based PD control is to provide as much yaw damping as possible without inhibiting the responsiveness of the vehicle or becoming intrusive.

In such a control structure including three feedback controllers and a feedforward controller, the phasing in a fishhook maneuver would be such that a particular controller is dominant as the transitional maneuver progresses (see Figure 8), which supports smooth intervention and reduces the potential for exciting pitch dynamics in the vehicle.

Because the transition control is designed to lead the roll PID control intervention used in the Quasi-steady State Feedback Control module (to be discussed in the next subsection) in a given maneuver, the roll PID control can then be initiated at a significantly higher pressure level, requiring less magnitude of the feedback signal to achieve the critical pressure level required to stabilize the vehicle.

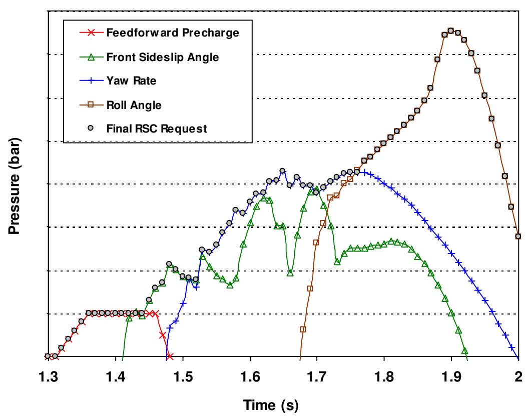

Figure 8. Pressure profile of the transition control during a fishhook maneuver.

In addition to the feedforward control such as the caliper pre-charge functionality, pressure build prediction and actuator delay compensation have also been introduced in the Transition Control module. Limitations in pressure build rates are compensated for by projecting forward when a pre-determined pressure level is likely to be requested, based on the chassis roll angle, roll rate, roll acceleration, and estimated caliper pressure. Pressure is built during the transition so that the desired peak pressure can be achieved when it is needed so as to reduce the effects of limited pressure build rates.

A detailed description regarding the Transition Control module can be found in [27].

Quasi-Steady State Feedback Control

During a quasi-steady state dynamic condition (usually in the non-linear dynamic region but with less dynamic content), a vehicle could experience slow buildup but extended wheel lift or sideslip angle. For example, during a J-turn maneuver for a vehicle with roof loading which raises its c.g., the vehicle could have one- or two-wheel lift before building up a large sideslip angle at the vehicle's rear axle. Note that the rate of change of the roll rate, yaw rate and the driver's steering wheel angle are all small. In this case the aforementioned transition control is no longer effective enough. While for the same maneuver if the vehicle has a lower c.g., the vehicle might slowly build up sideslip angle before one- or two-wheel lift occurs. A similar event could occur in a decreasing radius turn, such as those on some freeway on- or off-ramps.

These quasi-steady state conditions cannot be effectively captured by the computations used in ESC systems due to sensing limitation of the ESC sensor set. Under these driving conditions, the ability to detect and accurately estimate the slow build up of wheel departure angle and rear sideslip angle of the vehicle becomes critical for providing appropriately timed stabilizing torque. Using the RSC sensor cluster, the proper computation of the wheel departure angle θ_{xwm} and the rear sideslip angle β_{ra} referenced in the previous sections are possible. Hence the RSC system can provide the incremental ability to control the vehicle in the quasi-steady state region in addition to the highly dynamic rolling and yawing conditions.

Roll Angle Based Feedback Control

The relative roll angle θ_{xbm} between the vehicle body and the moving road is the main feedback control variable in this feedback controller structure.

For vehicles with a high c.g. and driven with rather steady state steering input, the wheel lift could build up at relatively low lateral accelerations (i.e., before a large rear sideslip angle is built up), thus leading to the buildup of the wheel departure angle. Since the Transition Control module described earlier does not address this scenario, the wheel departure angle based θ_{xbm} provides a unique characterization of such quasi-steady state conditions, hence an effective roll angle based feedback is possible. Therefore a PID feedback structure based on the relative roll angle between the body and the road (including wheel departure angle) θ_{xbm} is proposed.

The PID controller deadbands and gains are established at a level such that an appropriately progressive brake torque level is requested during periods of increasing wheel departure angle, while allowing for vehicle to do well in limit handling maneuvers without unnecessary brake interventions whenever the wheel departure angle is minor or non-existent.

Rear Sideslip Angle Based Feedback Control

For cases where a vehicle is operating with a low c.g. and is being driven in a near limit steady state maneuver, such as a J-turn, the vehicle may experience abrupt wheel lift if the vehicle's sideslip angle at the rear axle builds up to a certain threshold, i.e., the rear sideslip angle can slowly build up before a large wheel departure angle can build up.

In those cases, the roll-angle feedback control will be non-existent; yet buildup of rear side slip angle can occur at a slow rate. If such a condition is left undetected, the slowly growing rear sideslip angle can potentially lead to a sudden roll instability. Hence in this case, the calculated rear sideslip angle provides the ability to measure this slowly building sideslip angle.

A PD feedback controller structure using the calculated rear sideslip angle as the control variable is devised to control such diverging sideslip angle tendency.

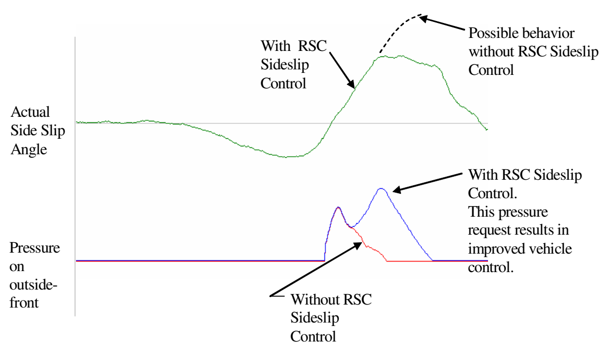

Figure 9. The RSC Sideslip Control brake pressure profile during a quasi-steady state maneuver.

Figure 9 shows, during a J-turn maneuver for a vehicle with nominal load, how RSC sideslip angle control requests brake pressure on the outside front wheel that extends beyond the ESC pressure request. Such control leads to reduced vehicle sideslip angle, which further reduces the tire lateral force helping to mitigate a potential rollover during such a quasi-steady state condition.

Control Integration inside RSC

The control strategies discussed in the previous subsections include the feedforward control within the Transition Control module which aims to prepare the brake hydraulics so as to eliminate delays in the brake pressure buildup, the feedback control within the Transition Control module which aims to mitigate rollover occurring during very dynamic conditions such as fishhooks and double lane changes, and the Quasi-steady State Feedback Control module which aims to mitigate rollovers occurring during non-dynamic conditions such as J-turn and decreasing radius turns.

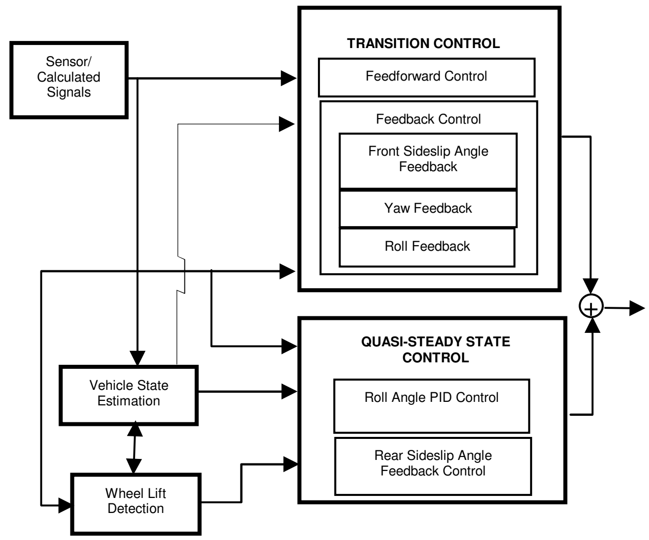

Figure 10. RSC Algorithm Integration.

In order to achieve a coordinated or combined control strategy, an integration among the afore-mentioned control strategies are conducted. Figure 10 provides a schematic overview of such integration.

RSC Interfacing with the Other Functions

The ESC system gives a driver the full ability to control the vehicle, but with intervention when needed to help the vehicle follow the driver's intent. One of the biggest differentiators between ESC and RSC is that the brake control in RSC is no longer solely in response to driver intent.

It is possible that the RSC system may cause the vehicle to reduce the lateral force at the outside tire patches, which could lead to the activation of the ESC system to request understeer control during a RSC activation, i.e., the RSC function is counteracted by the ESC understeer control. For this reason, it is important to integrate the RSC and ESC functions.

On the other hand, if during an RSC activation ESC oversteer control is also activated, the arbitrated brake pressure should pick the maximum between the ESC oversteer control pressure command and the RSC control pressure command together with a slip control function.

Notice that RSC function must also be integrated with the ABS function. While ABS aims to maintain a certain slip target to optimize stopping distance and steerability when in an ABS event, RSC will likely request an alternate slip target, so as to modulate lateral forces and subsequently reduce the resulting roll moment.

Since the active wheel lift detection is checking if a potentially lifted inside wheel will develop slip as a result from a small brake pressure build, the wheel can enter ABS event. Therefore, the active wheel lift detection used in RSC will also need to interact with the ABS function.

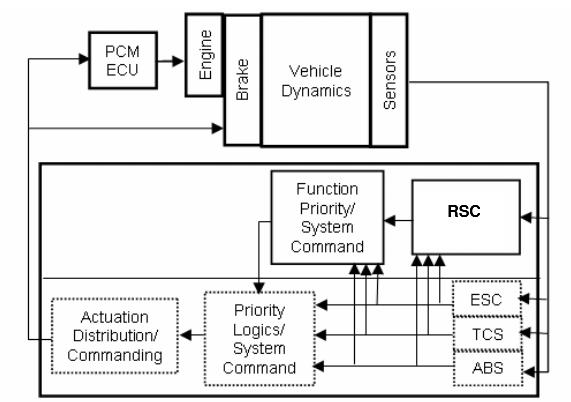

Figure 11. Function partition in a brake control Electronic Control Unit (ECU)

The RSC system resides in the brake ECU where the ABS, TCS and ESC functions reside, such that the integration between RSC and the existing brake control functions can be easily implemented. A block diagram for such an integration is shown in Figure 11, where the lower block depicts the brake ECU which is divided into two parts: the lower portion contains the existing functions and their priority and arbitration logic together with all the fail-safe and interface logic; the upper portion includes the RSC function and its priority and arbitration logic.

CONCLUSION

The Roll Stability Control ™ system discussed in this paper provides a system to mitigate vehicular rollovers, which works in harmony with and compliments the other functions existing in the current ESC systems. The addition of a roll rate sensor allows the RSC system to detect imminent rollover events regardless of variations of the vehicle loading condition and the road condition in both transition maneuvers and quasi-steady state maneuvers. The road bank determination conducted in the RSC system can also be used to improve ESC sideslip angle control during a slow sideslip buildup or on banked roads.

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