BRAKE CONTROL TO PREVENT THE ROLLOVER OF HEAVY VEHICLES BASED ON A LINEAR PARAMETER VARYING MODEL

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Keywords: vehicles, rollover avoidance, linear parameter varying control, stability, robustness.

Abstract

This paper is concerned with the use of active brake systems in order to decrease the rollover risk of heavy vehicles. Since the forward velocity of the vehicle changes in time, the combined yaw-roll dynamic model has a nonlinear structure. Selecting the forward velocity and the so-called normalized lateral load transfer at the rear side as scheduling parameters, a Linear Parameter Varying (LPV) model is constructed. In the control design based on the LPV model, the changes in the forward velocity, the performance requirements and the model uncertainties are taken into consideration. The control design is demonstrated in different situations.

1 Introduction

The aim of the rollover prevention is to provide the vehicle with the ability to resist overturning moments generated during cornering. Roll stability is determined by the height of the center of mass, the track width and the kinematic properties of the suspensions. The problem with heavy vehicles is a relatively high mass center and narrow track width. When the vehicle is changing lanes or trying to avoid obstacles, the vehicle body rolls out of the corner and the center of mass shifts outboard of the centerline, and a destabilizing moment is created.

In the literature there are many papers with different approaches on the active control of the heavy vehicles to decrease the rollover risk. Three main schemes concerned with the possible active intervention into the vehicle dynamics have been proposed: active anti roll bars, active steering and active brake. The control design is usually based on linear time invariant models and linear approaches. The forward velocity is handled as a constant parameter in the yaw-roll model, however, velocity is an important parameter as far as roll stability is concerned.

In this paper, a combined yaw-roll model including the roll dynamics of unsprung masses is formalized. This model is nonlinear with respect to the velocity of the vehicle. Thus, in our model velocity is handled as an LPV scheduling parameter. The controller based on this LPV model is adjusted continuously by measuring the vehicle velocity in real-time. The normalized lateral load transfer at the rear side is also applied as another scheduling parameter in order to focus on performance specifications. The control design itself is based on the Ljapunov quadratic stability criterion with respect to the uncertainty, which is caused by the difference between the linearized model and the actual nonlinear model, which contains the nonlinear tire force, the nonlinear suspension and actuator dynamics.

The structure of the paper is as follows. Section 2 discusses the active control problem to enhance roll stability of heavy vehicles. In Section 3 the combined yaw-roll model in which the forward velocity changes in time is constructed. Section 4 presents the control design based on the LPV method. Section 5 demonstrates the results of the control design.

2 Problem setup: Active control to enhance roll stability

The Figure 1 illustrates a combined yaw-roll dynamics of the vehicle is modelled by a three-body system, in which m_s is the sprung mass, $m_{u,f}$ is the unsprung mass at the front including the front wheels and axle, and $m_{u,r}$ is the unsprung mass at the rear with the rear wheels and axle.

A significant amount of research has been carried out to improve the active roll control of heavy vehicles. One of the methods proposed in the literature employs active anti roll bars by using a pair of hydraulic actuators in order to improve the roll stability of heavy vehicles. Lateral acceleration makes vehicles with conventional passive suspensions tilt out of corners. The center of the sprung mass shifts outboard of the vehicle centerline and this creates a destabilizing moment that reduces roll stability. The lateral load response is reduced by an active roll control

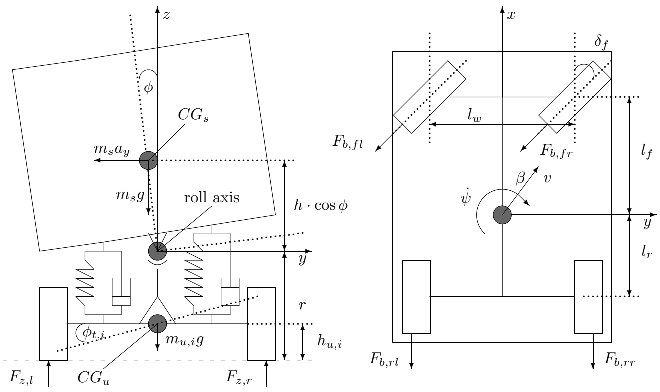

Figure 1: Rollover vehicle model

system in such a way that the control torque leans the vehicle into corners, see [7, 8, 12]. Since this mechanism is not able to reduce the lateral tire force, the objective of active anti roll bars is to generate a stabilizing moment to balance the overturning moment caused by lateral acceleration.

In the second scheme, an enhanced roll stability control system is presented focusing on rollover prevention by active steering. An actuator sets a small auxiliary front wheel steering angle in addition to the steering angle commanded by the driver. The aim is to decrease the rollover risk due to the transient roll overshoot of the vehicle when changing lanes or avoiding obstacles. The advantage of the active steering control is that it affects the lateral acceleration directly. However the active steering control has an effect on not only the roll dynamics of the vehicle it also modifies the desired path of the vehicle, so it affects the yaw motion. One extension of this method is the gain scheduling method, which takes into account the change of the vehicle velocity and the height of the center of gravity, see [1, 9].

In the third method, the electronic brake mechanism was proposed to increase rollover stability. In this method a small brake force is applied to each of the wheels and the slip response is monitored. In this way it is possible to established whether a given wheel is lightly loaded or the lift-off is close. When a dangerous situation is detected, unilateral brake forces are activated to reduce the lateral tire forces acting on the outside wheel, see [4, 5, 11]. The brake system reduces directly the lateral tire force which is the responsible force of rollover.

It should be noted that the physical limit of active anti roll bars is the relative roll angle between the body and the axle. In case of active steering as well as brake control the only physical limit is the actuator saturation. The active anti roll bars does not influence the yaw motion of the vehicle while the steering control and the brake control do. The aim of the rollover prevention system that uses the brake system of the vehicle is to decrease the lateral tire force acting on the wheels caused by lateral acceleration. So the lateral acceleration is an appropriate choice as a performance signal that should be minimized. However, if the lateral acceleration is small, i.e. the acceleration does not reach the critical value which causes the rollover of the vehicle, the controller also minimizes the signal. Hence, during a low speed maneuver (e.g. low-speed cornering) the control reduces the lateral acceleration to zero, and it would cause the vehicle to come to a halt. Thus, the goal is to design a controller that is only activated when the vehicle comes close to the rollover situation, i.e. the lateral acceleration reaches the critical value. In a normal driving situation the controller should not be activated and the behavior of the controlled vehicle dynamics should be exactly the same as the uncontrolled vehicle dynamics.

3 Nonlinear model of yaw-roll dynamics

The conditions of yaw-roll model used in control design are considered. It is assumed that the roll axis is parallel to the road plane in the longitudinal direction of the vehicle at a height r above the road. The location of the roll axis depends on the kinematic properties of the front and rear suspensions. The axles of the vehicle are considered to be a single rigid body with flexible tyres that can roll around the center of the roll. The tyre characteristics in the model are assumed to be linear. The effect caused by pitching dynamics in the longitudinal plane can be ignored in the handling behavior of the vehicle. The effects of aerodynamic inputs (wind disturbance) and road disturbances are also ignored. The roll motion of the sprung mass is damped by suspensions and stabilizers with the effective roll damping coefficients $b_{s,i}$ and roll stiffness $k_{s,i}$.

In the vehicle modelling the the lateral dynamics, the yaw moment, the roll moment of the sprung and the unsprung masses are taken into consideration. The symbols of the yaw-roll model are found in Table 1. The motion differential equations are the following.

$$mv(\dot{\beta} + \dot{\psi}) - m_s h\ddot{\phi} = Y_\beta \beta + Y_{\dot{\psi}} \dot{\psi} + Y_{\delta_f} \delta_f \tag{1}$$

$$-I_{xz}\ddot{\phi} + I_{zz}\ddot{\psi} = N_\beta \beta + N_{\dot{\psi}} \dot{\psi} + N_{\delta_f} \delta_f + \frac{l_w}{2} \Delta F_b \tag{2}$$

$$\begin{aligned} \Big(I_{xx} + m_s h^2\Big)\ddot{\phi} - I_{xz}\ddot{\psi} &= m_s g h \phi + m_s v h (\dot{\beta} + \dot{\psi}) \\ &\quad - k_f(\phi - \phi_{t,f}) - b_f(\dot{\phi} - \dot{\phi}_{t,f}) \\ &\quad - k_r(\phi - \phi_{t,r}) - b_r(\dot{\phi} - \dot{\phi}_{t,r}) \end{aligned} \tag{3}$$

$$\begin{aligned} -r\Big(Y_{\beta,f}\beta + Y_{\dot{\psi},f}\dot{\psi} + Y_{\delta_f}\delta_f\Big) &= m_{u,f} v (r - h_{u,f})(\dot{\beta} + \dot{\psi}) \\ &\quad + m_{u,f} g h_{u,f} \phi_{t,f} - k_{t,f}\phi_{t,f} \\ &\quad + k_f(\phi - \phi_{t,f}) + b_f(\dot{\phi} - \dot{\phi}_{t,f}) \end{aligned} \tag{4}$$

$$\begin{aligned} -r\Big(Y_{\beta,r}\beta + Y_{\dot{\psi},r}\dot{\psi}\Big) &= m_{u,r} v (r - h_{u,r})(\dot{\beta} + \dot{\psi}) \\ &\quad - m_{u,r} g h_{u,r} \phi_{t,r} - k_{t,r}\phi_{t,r} \\ &\quad + k_r(\phi - \phi_{t,r}) + b_r(\dot{\phi} - \dot{\phi}_{t,r}) \end{aligned} \tag{5}$$

Here, the tyre coefficients are given by: $Y_\beta = -(C_f + C_r)\mu$, $N_\beta = (C_r l_r - C_f l_f)\mu$, $Y_{\dot{\psi}} = (C_r l_r - C_f l_f)\frac{\mu}{v}$, $N_{\dot{\psi}} = -(C_f l_f^2 + C_r l_r^2)\frac{\mu}{v}$, $Y_{\delta_f} = C_f\mu$, $N_{\delta_f} = C_f l_f \mu$. These equations can be expressed in a state space representation. Let the state vector be the following:

$$x = \begin{bmatrix} \beta & \dot{\psi} & \phi & \dot{\phi} & \phi_{t,f} & \phi_{t,r} \end{bmatrix}^T \tag{6}$$

. The system states are the side slip angle of the sprung mass β, the yaw rate $\dot{\psi}$, the roll angle ϕ, the roll rate $\dot{\phi}$, the roll angle of the unsprung mass at the front axle $\phi_{t,f}$ and at the rear axle $\phi_{t,r}$ respectively. Then the state equation arises in the following form:

$$\dot{x} = A(v)x + B_1\delta_f + B_2 u \tag{7}$$

Table 1: Symbols of the yaw-roll model

Symbols	Description
h	height of CG of sprung mass from roll axis
$h_{u,i}$	height of CG of unsprung mass from ground
r	height of roll axis from ground
a_y	lateral acceleration
β	side-slip angle at center of mass
ψ	heading angle
$\dot{\psi}$	yaw rate
ϕ	sprung mass roll angle
$\phi_{t,i}$	unsprung mass roll angle
δ_f	steering angle
u_i	control torque
C_i	tire cornering stiffness
F_{zi}	total axle load
R_i	normalized load transfer
k_i	suspension roll stiffness
b_i	suspension roll damping
$k_{t,i}$	tire roll stiffness
I_{xx}	roll moment of inertia of sprung mass
I_{xz}	yaw-roll product of inertial of sprung mass
I_{zz}	yaw moment of inertia of sprung mass
l_i	length of the axle from the CG
l_w	vehicle width
μ	road adhesion coefficient

The δ_f is the front wheel steering angle. The control input is the difference of brake forces between the left and the right hand side of the vehicle.

$$u = \Delta F_b \tag{8}$$

The control input provided by the brake system generates a yaw moment, which affects the lateral tire forces directly. In our case it is assumed that the brake force difference ΔF_b provided by the controller is applied to the rear axle. This means that only one wheel is decelerated at the rear axle. This declaration is caused by an appropriate yaw moment. In our case the difference between the brake forces can be given $\Delta F_b = F_{b,rl} - F_{b,rr}$. This assumption does not restrict the implementation of the controller because it is possible that the control action be distributed on the front and the rear wheels at one of the two sides. The reason for distributing the control force to front and rear wheels is to minimize the wear of the tires. In this case a logic is required which calculates the brake forces for the wheels.

In the equation (7) the $A(v)$ matrix depends on the forward velocity of the vehicle nonlinearly. In the linear yaw-roll model the velocity is considered a constant parameter. However, forward velocity is an important stability parameter so that it is considered to be a variable of the motion. Hence the throttle is constant during a lateral maneuver and the forward velocity depends on only the brake forces. The differential equation for forward velocity is $m\dot{v} = -\Delta F_b$.

4 Control design based on the LPV method

This section describes the design of a brake control system for a single unit vehicle. The LPV modelling techniques allow us to take into consideration the nonlinear effect in the state space description. The LPV model is valid in the whole operating region of interest. The class of finite dimensional linear systems, whose state space entries depend continuously on a time varying parameter vector, $\rho(t)$, is called LPV. The trajectory of the vector-valued signal, $\rho(t)$ is assumed to be unknown, although its value can be accessed or measured in real time and is constrained a priori to lie in a specified bounded set. The idea behind using LPV systems is to take advantage of the knowledge of the dynamics of the system, see [3, 13].

The formal definition of an LPV system is given below:

$$\dot{x} = A(\rho)x + B(\rho)u \tag{9}$$

One characteristic of the LPV system is that it must be linear in the pair formed by the state vector, x, and the control input vector, u. The matrices A and B are generally nonlinear functions of the scheduling vector ρ. Quasi-LPV systems arise whenever any of the scheduling variables, ρ, are also a state of the system. In our case the state space representation dependence on velocity is nonlinear (see equation (7)). If the forward velocity v is chosen as a scheduling parameter, the differential equations of the yaw-roll motion are linear in the state variables.

Consider the closed-loop system in Figure 2, which includes the feedback structure of the model $G(\rho)$ and controller $K(\rho)$, and elements associated with the uncertainty models and performance objectives. In the diagram, u is the control input, which is generated by the brake system, y is the measured outputs, which contains the perturbed lateral acceleration of sprung mass a_y and the derivative of roll angle ϕ, n is the measurement noise. In the figure, δ_f is the steering angle as a disturbance signal, which is set by the driver. The z represents the performance outputs, i.e. the lateral acceleration a_y and the control input ΔF_b.

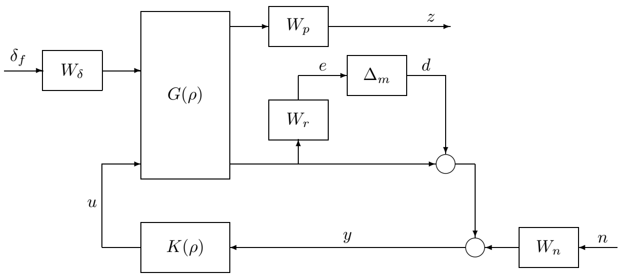

Figure 2: The closed-loop interconnection structure

The LPV synthesis used in this research requires a gridded parameter space. For the interconnection structure, $\mathcal{H}_\infty$ controllers are synthesized for several values of velocity in a range $v = [40\text{kmh}, 100\text{kmh}]$. The spacing of the grid points is selected on the basis of how well the $\mathcal{H}_\infty$ point designs perform for plants around the design point. 7 grid points are selected for the scheduling parameter design space. Weighting functions for both the performance and robustness specifications are defined in all of the grid points. With respect to the robustness requirement, the same frequency weighting functions are applied in the whole parameter space and the effect of the scheduling parameter is neglected. It is a reasonable engineering assumption, since the uncertainty, i.e. unmodelled dynamics and parametric uncertainties, does not depend on the forward velocity.

The nominal model usually approximates well the low-middle frequency range behavior of the plant. In the high-frequency range the model is uncertain, thus parametric uncertainty is needed to represent the unmodelled dynamics. The uncertainties of the model is represented by W_r and Δ_m. Design models used for roll stability control typically exhibit high fidelity at lower frequencies ($\omega < 10$ Hz), but they deteriorate rapidly at higher frequencies due to poorly modelled or neglected effects. Thus, W_r is selected as $W_r = 2.25\frac{S+20}{S+450}$.

The input scaling weight W_δ normalizes the steering angle to the maximum expected command. It is selected as $5\pi/180$, which corresponds to 5 degrees of steering angle command. W_n is selected as a diagonal matrix, which accounts for sensor noise models in the control design. The noise weight for the lateral acceleration is chosen 0.01 m/s^2 and for the derivative of roll angle ϕ 0.01 deg/sec.

The weighting function W_p represents the performance outputs, and it contains the W_{p_a} and W_{p_u} components. The purpose of these weighting functions is to keep the lateral acceleration and control input small over the desired frequency range. The weighting functions chosen for performance outputs can be considered as penalty functions. That is, weights should be large in a frequency range where small signals are desired and small where larger performance outputs can be tolerated. The weighting function W_{p_a} is selected as:

$$W_{p_a} = \phi_a \frac{(s/2000+1)}{(s/12+1)} \tag{10}$$

Here, it is assumed that in the low frequency domain the steering angle at the lateral accelerations of the body should be rejected by a factor of ϕ_a. The $W_{p_u} = 1 \cdot 10^3$ is selected for control design. Consequently, the maximal gain of the brake force difference can be 10^3 in the frequency domain. The ϕ_a is gain, which reflect the relative importance of the lateral acceleration in the LPV controller design. A large gain ϕ_a corresponds to a design that avoids the roll over situation. If the ϕ_a value is small the vehicle is in a normal driving situation, in which the minimization of lateral acceleration is not needed.

The lateral acceleration of the sprung mass should be small when the vehicle is close to the rollover situation. The rollover situation can be detected if the lateral load transfers for both axles are calculated. The lateral load transfer can be given:

$$\Delta F_{z,i} = 2\frac{k_{t,i}\phi_{t,i}}{l_w} \tag{11}$$

where the subscript i denotes the front and rear axles. The lateral load transfer can be normalized in such a way that the load transfer is divided by the total axle load. The normalized load transfer R_i value corresponds to the largest possible load transfer.

$$R_i = \frac{\Delta F_{z,i}}{F_{z,i}} \tag{12}$$

where $F_{z,i}$ (i=f,r) the total load to the front and rear axles. If the R_i takes on the value ± 1, the inner wheels in the bend lift off. The goal is to design a controller that is only activated when the vehicle comes close the rollover situation, i.e. the lateral acceleration reaches the critical value. In a normal driving situation the controller should not be activated. Consequently, when the acceleration is not critical the weighting function should be small and when the acceleration reaches the critical value the weight should be large to avoid the rollover.

In order to take into consideration such nonlinear function of the controller with respect to the operating domain a parameter dependent weighting function must be used. The weight should be scheduled by the normalized load transfer at the rear side R_r, which can be deduced from the rollover situation. The roll-over of a vehicle is affected by the suspension stiffness to load ratio, which is greater at the rear axle than at the front one. Thus, in a case of an obstacle avoidance in an emergency, the rear wheels are first to lift off. Using the normalized lateral load transfer the rollover of the vehicle can be predicted with high probability.

The ϕ_a is chosen to be parameter-dependent, i.e., the function of R_r. The parameter dependent gain ϕ_a captures the relative importance of the acceleration response. When R is small, i.e., when the vehicle is not in an emergency, $\phi_a(R)$ is small, indicating that the LPV controller should not focus on minimizing acceleration. On the other hand, when R approaches the critical value, $\phi_a(R)$ is large, indicating that the controller should focus on preventing the rollover. In this paper the parameter dependence of the gain is characterized by the constants R_1 and R_2. The parameter dependent gain $\phi_a(R)$ in equation (10) is as follows:

$$\phi_a(R) = \begin{cases} 0 & \text{if} \quad |R| < R_1 \\ \frac{2}{R_2 - R_1}(|R| - R_1) & \text{if} \quad R_1 \leq |R| \leq R_2 \\ 2 & \text{otherwise} \end{cases} \tag{13}$$

R_1 defines the critical status when the vehicle is close to the rollover situation, i.e. all wheels are on the ground but the lateral tire force of the inner wheels tends to zero. The closer R_1 is to 1, the later the controller will be activated. R_2 parameter shows how fast the controller should focus on minimizing the lateral acceleration. The smaller the difference between R_1 and R_2 is, the more quickly the performance weight punishes the lateral acceleration. In the control design the constants are selected as $R_1 = 0.85$ and $R_2 = 0.95$.

In the LPV model of the yaw-roll motion two parameters are selected, the forward velocity v and the the normalized lateral load transfer at the rear side R_r. The value v is measured directly, while the parameter R_r can be calculated by using the measured roll angle of the unsprung mass $\phi_{t,r}$.

5 Simulation results

In this section, illustrative examples are shown for the roll control mechanism, which is based on the LPV control. In the first example, cornering responses of a single unit vehicle model travelling at 70 kph can be seen. Figure 3 shows the responses using passive (dashed) and active (solid) breaking mechanisms. In case of the passive brake mechanism there is no additional unilateral brake force to prevent the vehicle from rolling over. In the passive case the brake force is only provided by the driver and it is distributed equivalently to all of the wheels and they do not generate yaw moment around the center of mass. The steering angle applied in the simulation is a step signal. In order to avoid the unrealistic change in the steering angle, a ramp signal is applied, where the signal reaches the maximum value (3.5 degrees) in 0.5 s and filtered at 4 rad/s to represent the finite bandwidth of the driver. In the passive case the forward velocity is constant during

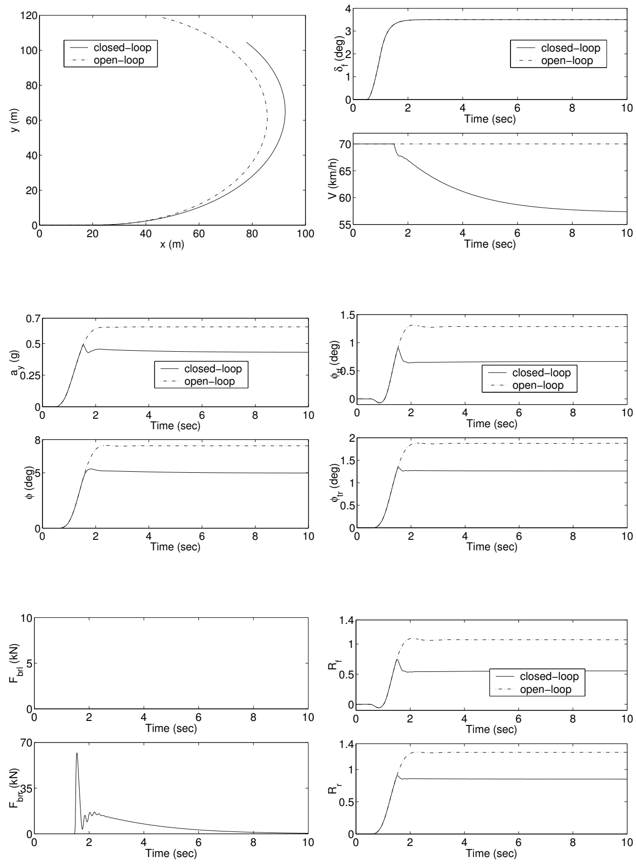

Figure 3: Time responses of cornering maneuver

the cornering maneuver because it is assumed the driver is not pushing on the brake pedal. In case of the active brake control the forward velocity is not constant because the brake force provided by the controller decelerates the vehicle. As the lateral acceleration increases, the normalized load transfer lifts up the rear axle more quickly then the front one since the ratio of the effective roll stiffness to the axle load is greater at the driven axle. In the passive case the normalized load transfers are over the value ± 1, which means that the lateral force on one of the curve inner side wheels becomes zero.

The yaw-roll model is only valid if the normalized load transfers are below the values ± 1 for both axles. Also note that in the passive case the relative displacement of the suspension $(\phi - \phi_{t,i})$ exceeds the acceptable limit, which is about $6 - 8$ degrees. This means that the suspension working space saturates and it no longer allows suspension travel. Using active brake control the relative roll angle is reduced significantly and it will be within the acceptable limit. The roll angle of the unsprung masses is slightly different due to the different suspension parameters and the stiffness to load ratio. The brake force is approximately 60 kN at the rear axle on the right-hand side.

In the second example, a double lane change maneuver is performed. This maneuver is often used to avoid an obstacle in an emergency. Figure 4 shows the time responses to the double lane change steering input. The maneuver has a 2 m path deviation over 100 m. The size of the path deviation is chosen to test a real obstacle avoidance emergency. The vehicle velocity is 80 kph. The steering angle input is generated in such a way that the vehicle with a passive brake control comes close to rollover situation during the maneuver and its normalized load transfers are over ± 1. The lateral acceleration in both the passive and the active case is identical if the normalized load transfers do not reach the critical value (the critical value is determined by R_1) but after the critical value is exceeded the control algorithm is activated and the active brake system reduces the lateral acceleration. At the time when the brake control is activated the rear-left wheel is braked to avoid the rollover of the vehicle (see Figure 4). Approximately $100kN$ control force is required for the rear-left wheel during this maneuver.

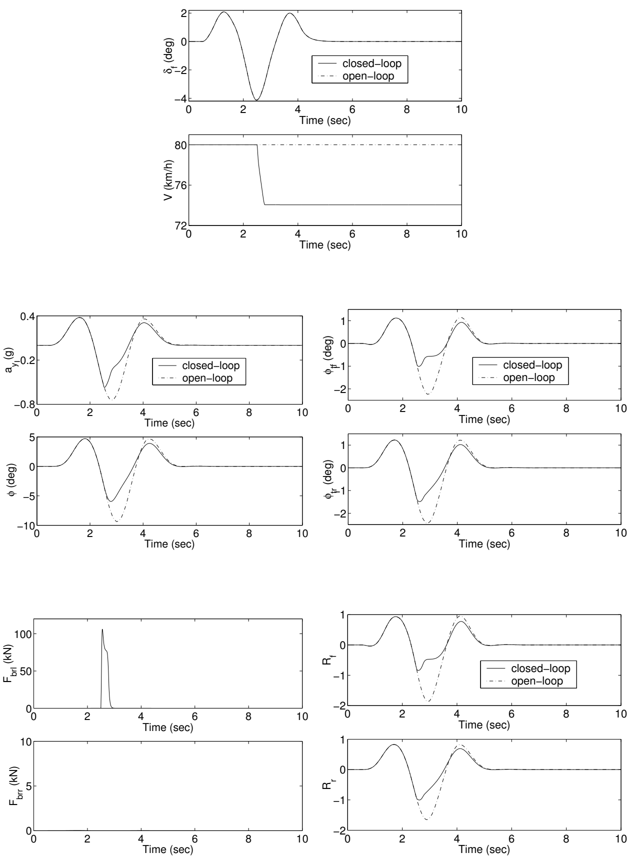

Figure 4: Time responses to double lane change steering input

Acknowledgement: This work has been supported by the Hungarian National Science Foundation (OTKA) under the grant $T-043111$, which is gratefully acknowledged.

6 Conclusion

The aim of this paper is to improve the roll stability of heavy vehicles by applying an active brake system based on the LPV modelling and control design. In the LPV modelling the forward velocity and the normalized lateral load transfer at the rear side are chosen as scheduling parameters. The control design based on the LPV method results in a compensator which takes the change in the velocity and the model uncertainties into consideration. The control design is illustrated in simulation examples.

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