

The Design of a Combined Control Structure to Prevent the Rollover of Heavy Vehicles

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In this paper a combined control structure to decrease the rollover risk of heavy vehicles is developed. In this structure active anti-roll bars are combined with an active brake control. Selecting the forward velocity and the lateral load transfer at the rear as scheduling parameters, a linear parameter varying model is constructed. In the control design the changes in the forward velocity, the performance specifications and the model uncertainties are taken into consideration. The control mechanism is demonstrated in various maneuver situations.

Keywords: Nonlinear Control Systems; Linear Parameter Varying Control; Robust Control; Vehicle Dynamics; Automotive Control; Prevention of Rollover

1. Introduction and Motivation

The aim of rollover prevention is to provide the vehicle with an ability to resist overturning moments generated during cornering. The problem with heavy vehicles in terms of the roll stability 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 several papers on the active control of the heavy vehicles with different approaches 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.

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 active anti-roll bars, which generate a stabilizing moment to balance the overturning moment in such a way that the control torque leans the vehicle into the corners, see [6,11,14,15]. In another case, the combined roll moment of the front and rear suspensions is designed to reduce body roll and distribute the roll moment, see [1,9].

In the second method, an enhanced roll stability control system focusing on rollover prevention by active steering is presented. 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

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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, but also modifies the desired path of the vehicle, so it affects the yaw motion. In this control a proportional feedback of both the roll rate and the roll acceleration are used, see [3]. One extension of this method is the gain scheduling method, which takes into account the change in vehicle velocity and the height of the center of gravity, see [2]. This control concept is also extended by a nonlinear steering control loop to prevent rollover, see [12].

In the third method, the electronic brake mechanism is 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 establish whether a given wheel is lightly loaded and the lift-off is imminent. When a dangerous situation is detected, unilateral brake forces are activated to reduce the lateral tire forces acting on the outside wheel, see [7,8,13]. The brake system reduces directly the lateral tire force, which is responsible for the rollover. Additional reasons for using the brake system of a vehicle are the use of the most appropriate actuator and the low cost of the solution.

It should be noted that the disadvantage of the active anti-roll bars is that the maximum stabilizing moment is limited physically by the relative roll angle between the body and the axle. The active anti-roll bars do not influence directly the yaw motion of the vehicle while the steering control and the brake control do. In the case of active steering as well as active brake control the only physical limit is the actuator saturation. These compensators, however, have effects on not only the roll dynamics of the vehicle but they also modify the desired path of the vehicle, so they affect the yaw motion. Thus, the different control structures should be combined in one control mechanism. In Odenthal *et al.* [12] the linear steering control is extended by nonlinear emergency steering and braking control. In terms of the autonomic vehicle control the combination of the brake and the throttle were proposed, see [10].

In our project a combined control mechanism, in which both the active anti-roll bars and the active brake control are applied, has been developed. In this paper, first a combined yaw-roll model including the roll dynamics of unsprung masses is formalized. This model is nonlinear with respect to the forward velocity of the vehicle, and it contains nonlinear components, for example, tires, dampers and actuators. The control design is based on a linear parameter varying (LPV) model, which is adjusted continuously by the forward velocity of the vehicle 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 Lyapunov quadratic stability criterion with respect to uncertainties. They are caused by the difference between the linearized model and the real nonlinear model, which contains the nonlinear tires, dampers and actuators. Thus, in the control design the changes in the forward velocity, the performance specifications and the model uncertainties are taken into consideration.

The structure of the paper is as follows. In Section 2, the LPV structure of the combined yaw–roll model in which the forward velocity changes in time is constructed. In Section 3, the combined control design is discussed. In the LPV model for control design the normalized lateral load transfer at the rear side is selected. The method of the LPV control design is also presented. In Section 4, the combined control mechanism is demonstrated in various vehicle maneuvers: in a double lane change and in a cornering situation. Finally, Section 5 contains some concluding remarks.

2. The LPV Model of the Vehicle Dynamics

2.1. The Nonlinear Model of the Combined Yaw – roll Dynamics

Figure 1 illustrates the combined yaw – roll dynamics of the vehicle, which is modelled by a three-body system. Here 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.

The preliminary conditions of yaw–roll model used in control design are the following. 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 $k_{s,i}, i \in \{f, r\}$. The driving

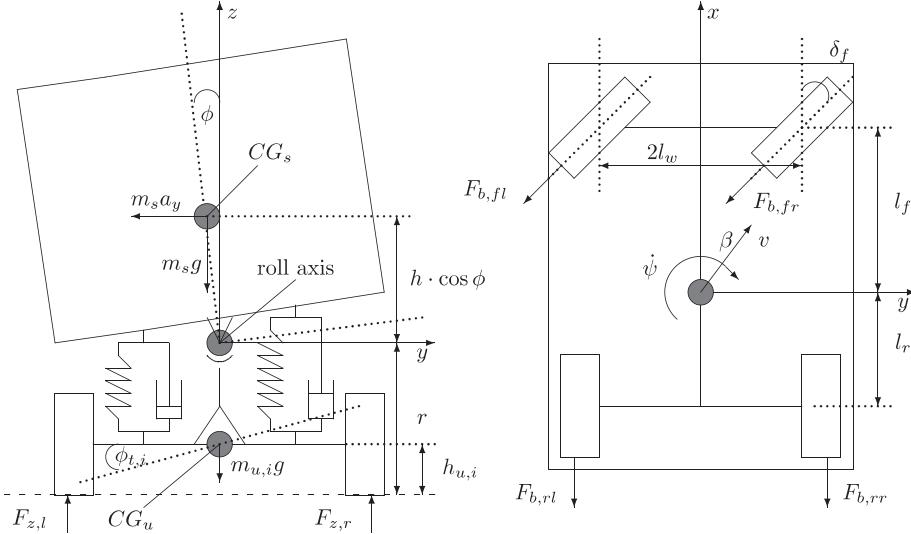


Fig. 1. Rollover vehicle model.

thrust is assumed to remain constant and it is distributed between the driven wheels, so it does not generate yaw moment around the center of mass.

In vehicle modeling the motion differential equations of the yaw – roll dynamics of the single unit vehicle are formalized, that is, the lateral dynamics, the yaw moment, the roll moment of the sprung mass, the roll moment of the front and the rear unsprung masses. The first equation is a force equation for the lateral dynamics, the others are torque balance equations for the yaw moments and roll moments. The detailed derivation of the equations of the yaw – roll dynamics of the single unit vehicle can be found in [15,14]. The symbols of the yaw – roll model are found in Table 1.

$$mv(\dot{\beta} + \dot{\psi}) - m_s h \ddot{\phi} = F_{y,f} + F_{y,r}, \quad (1)$$

$$-I_{xz}\ddot{\phi} + I_{zz}\ddot{\psi} = F_{y,f}l_f - F_{y,r}l_r + l_w \Delta F_b, \quad (2)$$

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

$$\begin{aligned} -rF_{y,f} &= m_{u,f}v(r - h_{u,r})(\dot{\beta} + \dot{\psi}) + m_{u,f}gh_{u,f}\phi_{t,f} \\ &\quad - k_{t,f}\phi_{t,f} + k_f(\phi - \phi_{t,f}) + b_f(\dot{\phi} - \dot{\phi}_{t,f}) + u_f, \end{aligned} \quad (4)$$

$$\begin{aligned} -rF_{y,r} &= m_{u,r}v(r - h_{u,r})(\dot{\beta} + \dot{\psi}) - m_{u,r}gh_{u,r}\phi_{t,r} \\ &\quad - k_{t,r}\phi_{t,r} + k_r(\phi - \phi_{t,r}) \\ &\quad + b_r(\dot{\phi} - \dot{\phi}_{t,r}) + u_r. \end{aligned} \quad (5)$$

Table 1 Symbols of the yaw–roll model.

Symbols	Description
m_s	Sprung mass
$m_{u,i}$	Unsprung mass
m	The total vehicle mass
v	Forward velocity
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
ΔF_b	Difference between the braking forces
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
$v k_{t,i}$	Tire roll stiffness
I_{xx}	Roll moment of inertia of sprung mass
I_{xz}	Yaw – roll product of inertia of sprung mass
I_{zz}	Yaw moment of inertia of sprung mass
l_i	Length of the axle from the CG
l_w	Half of the vehicle width
μ	Road adhesion coefficient

The lateral tire forces in the direction of the wheel ground contact velocity are approximated proportionally to the tire slide slip angle α .

$$F_{y,f} = \mu C_f \alpha_f, \quad (6a)$$

$$F_{y,r} = \mu C_r \alpha_r. \quad (6b)$$

The C_i is the tire side slip constant and α_i is the tire slide slip angle associated to front and rear axles. The chassis and the wheels have identical velocities at the wheel ground contact points. The velocity equations for the front and rear wheels in the lateral and in the longitudinal directions are as follows:

$$v_{w,f} \sin(\delta_f - \alpha_f) = l_f \cdot \dot{\psi} + v \sin \beta, \quad (7a)$$

$$v_{w,f} \cos(\delta_f - \alpha_f) = v \cos \beta, \quad (7b)$$

$$v_{w,r} \sin \alpha_r = l_r \cdot \dot{\psi} - v \sin \beta, \quad (7c)$$

$$v_{w,r} \cos \alpha_r = v \cos \beta. \quad (7d)$$

At stable driving conditions, the tire side slip angle α_i is normally not larger than 5° and the above equation can be simplified by substituting $\sin x \approx x$ and $\cos x \approx 1$. The classic equations for the tire side slip angles are then given as

$$\alpha_f = -\beta + \delta_f - \frac{l_f \cdot \dot{\psi}}{v}, \quad (8a)$$

$$\alpha_r = -\beta + \frac{l_r \cdot \dot{\psi}}{v}. \quad (8b)$$

These equations can be expressed in a state space representation. Let the state vector be the following:

$$x = [\beta \quad \dot{\psi} \quad \phi \quad \dot{\phi} \quad \phi_{t,f} \quad \phi_{t,r}]^T. \quad (9)$$

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 $\phi_{t,r}$.

Then the state equation arises in the following form:

$$\dot{x} = A(v)x + B_1 \delta_f + B_2 u. \quad (10)$$

The δ_f is the front wheel steering angle. The control inputs are roll moments between the sprung and unsprung mass generated by active anti-roll bars and the difference in brake forces between the left- and right-hand side of the vehicle.

$$u = [u_f \quad u_r \quad \Delta F_b]. \quad (11)$$

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

difference in brake forces ΔF_b provided by the compensator is applied on the rear axle. This means that only one wheel is decelerated at the rear axle. This deceleration generates an appropriate yaw moment.

This assumption does not restrict the implementation of the compensator because it is possible that the control action be distributed at the front and the rear wheels at either of the two sides. The reason for sharing the control force between the front and rear wheels is to minimize the wear of the tires. In this case a sharing logic is required which calculates the brake forces for wheels. Thus, the difference between the left and right brake forces is as follows:

$$\Delta F_b = (F_{b,rl} + d_2 F_{b,fl}) - (F_{b,rr} + d_1 F_{b,fr}), \quad (12)$$

where

$$d_1 = \frac{\sqrt{l_f^2 + l_w^2}}{l_w} \sin(\gamma + \delta_f),$$

$$d_2 = \frac{\sqrt{l_f^2 + l_w^2}}{l_w} \sin(\gamma - \delta_f), \quad \gamma = \arctan\left(\frac{l_w}{l_f}\right).$$

The first term in Eq. (12) is the sum of brake forces on the left-hand side and the second one is the brake forces on the right-hand side. It can be assumed that the steering angle is small at stable driving conditions, so the d_1 and d_2 are approximately equal to 1. Thus, the brake force difference provided by the control can be shared equivalently between the rear and front wheels at the suitable side.

That is,

$$F_{b,rl} = \frac{\Delta F_b}{2} \quad \text{and} \quad d_2 F_{b,fl} = \frac{\Delta F_b}{2} \quad (13)$$

or

$$F_{b,rr} = \frac{\Delta F_b}{2} \quad \text{and} \quad d_2 F_{b,fr} = \frac{\Delta F_b}{2}. \quad (14)$$

In Eq. (10) 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. Forward velocity is approximately equivalent to the velocity in the longitudinal direction while the slide slip angle β is small. It can be assumed that the side slip angle is small in stable driving conditions. Hence the driving throttle is constant during a lateral maneuver and the forward velocity depends on only the brake forces. The differential equation for forward velocity is

$$m\dot{v} = -F_{b,fl} - F_{b,fr} - F_{b,rl} - F_{b,rr}. \quad (15)$$

2.2. LPV Model for the Combined Yaw – Roll Dynamics

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 not to be known in advance, although its value is accessible (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 casual knowledge of the dynamics of the system, see [5,17]. The formal definition of an LPV system is given below:

Definition 1. For a compact subset $\mathcal{P} \subset \mathcal{R}^S$, the parameter variation set $\mathcal{F}_{\mathcal{P}}$ denotes the set of all piecewise continuous functions mapping \mathcal{R} (time) into \mathcal{P} with a finite number of discontinuities in any interval. The compact set $\mathcal{P} \subset \mathcal{R}^S$, along with continuous functions $A : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}, B : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n_u}$ represents an nth order LPV system $G_{\mathcal{F}_{\mathcal{P}}}$ whose dynamics evolve as

$$\dot{x} = A(\rho)x + B(\rho)u, \quad (16)$$

where $\rho \in \mathcal{F}_{\mathcal{P}}$.

One characteristics 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 . In our case the state space representation dependence on the velocity is nonlinear (see Eq. (10)). Choosing the forward velocity v as a scheduling parameter, the differential equations of the yaw – roll motion are linear in the state variables.

3. The Combined Control Design to Prevent the Rollover of Heavy Vehicles

3.1. LPV Control for the Combined Yaw – Roll Dynamics

The objective of the roll control system is to use both the roll moment from active anti-roll bars and a controlled brake system to maximize the roll stability of the vehicle. The rollover of the vehicle starts when, in a bend, the tire contact force on the inner wheels becomes zero. The rollover is caused by the high lateral inertial force generated by lateral acceleration. If

the position of center of gravity (CG) is high or the forward velocity of the vehicle is larger than allowed at a given steering angle the resulting lateral acceleration is also large and might initiate a rollover. The roll stability of the vehicle can be improved by two means. One of them is to generate a stabilizing moment to balance the overturning moment caused by lateral acceleration. The other one is the brake force that is able to reduce the lateral tire force which is responsible for rollover. The rollover situation can be detected if the lateral load transfers for both axles are calculated. The lateral load transfer can be given:

$$\Delta F_{z,ai} = 2 \frac{k_{t,i}\phi_{t,i}}{l_w}, \quad (17)$$

where $k_{t,i}$ the stiffness of tires at the front and rear axles, $\phi_{t,i}$ is the roll angle of the unsprung mass and l_w is the vehicle's width. The lateral load transfer can be normalized in such a way that the load transfer is divided by the total axle load.

$$R_i = \frac{\Delta F_{z,i}}{F_{z,i}}, \quad (18)$$

where the $F_{z,i}$ is the total axle load. The normalized load transfer R_i value corresponds to the largest possible load transfer. If the R_i takes on the value ± 1 then the inner wheels in the bend lift-off. By varying the control torques between the sprung and unsprung masses the active roll control system can manipulate the axle load transfers and the body roll angles. Moreover using the brake system of the vehicle a yaw moment can be generated by unilateral brake forces, which can reduce the lateral acceleration directly.

Note that there are other solutions to check the rollover coefficient, which is the basis of the stability problem. Odenthal *et al.* [12] has proposed a method for calculating the rollover coefficient. However, it neglects the roll angle of the unsprung masses, and it assumes that all the wheels have road contact. Palkovics *et al.* [13] has proposed a heuristic method in which a small effect was generated by the brake or throttle system.

The roll stability achieved by limiting the lateral load transfers to below the levels required for wheel lift-off. Specifically, the load transfers can be minimized to increase the inward lean of the vehicle. The center of mass shifts laterally from the nominal center line of the vehicle to provide a stabilizing effect. While attempting to minimize the load transfer, it is also necessary to constrain the roll angles between the sprung and unsprung masses ($\phi - \phi_i$) so that they are within the limits of the travel of suspension. The suspension might reach its physical limit and the lateral

displacement moment is not enough to balance the primary overturning moment caused by acceleration. In this case the lateral tire forces have to be reduced directly by the brake system to prevent the rollover of the vehicle.

The goal is to design the control that uses the active anti-roll bars all the time to prevent the rollover. The controlled brake system is only activated when the vehicle comes close to the rollover situation. In a normal driving situation the brake part of the control should not be activated. However, if the normalized load transfer reaches a critical value the brake system has to minimize the lateral acceleration to prevent the rollover. The critical value of the normalized load transfer is determined when the load transfer of one of the curve-inner wheels tend to zero.

In order to describe the control objective, let the yaw – roll dynamics have a partitioned representation in the following way:

$$\begin{bmatrix} \dot{x}(t) \\ z(t) \\ y(t) \end{bmatrix} = \begin{bmatrix} A(\rho) & B_1(\rho) & B_2(\rho) \\ C_1(\rho) & D_{11}(\rho) & D_{12}(\rho) \\ C_2(\rho) & D_{21}(\rho) & D_{22}(\rho) \end{bmatrix} \begin{bmatrix} x(t) \\ w(t) \\ u(t) \end{bmatrix}, \quad (19)$$

where $\rho \in \mathcal{F}_P$ is the scheduling vector, $y \in \mathcal{R}^{n_y}$ is the measured output, $u \in \mathcal{R}^{n_u}$ is the control input, $w \in \mathcal{R}^{n_w}$ is the exogenous disturbance, and $z \in \mathcal{R}^{n_z}$ is the performance output.

The induced \mathcal{L}_2 norm of a LPV system $G_{\mathcal{F}_P}$, with zero initial conditions, is defined as:

$$\|G_{\mathcal{F}_P}\| = \sup_{\rho \in \mathcal{F}_P} \sup_{\|w\|_2 \neq 0, w \in \mathcal{L}} \frac{\|z\|_2}{\|w\|_2}, \quad (20)$$

where w is the steering angle δ_f as a disturbance signal, which is set by the driver and z consists of the lateral acceleration a_y and the lateral load transfers associated with the front and rear axles $\Delta F_{z,i}$. In other words, the goal is to design an LPV control which minimizes the lateral acceleration and the lateral load transfers during a maneuver generated by the steering angle δ_f . The measured outputs are the lateral acceleration of the sprung mass a_y and the derivative of the roll angle $\dot{\phi}$. The control inputs are the roll moments generated by the active anti-roll bars u_i and the brake force ΔF_b .

If $G_{\mathcal{F}_P}$ is quadratic stable then this quantity is finite. The quadratic stability can be extended to the parameter dependent stability, which is the generalization of the quadratic stability concept.

Definition 2. Given a compact set $\mathcal{P} \subset \mathcal{R}^S$, and a function $A : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$, the function A is parametrically dependent stable over \mathcal{P} if there exist a

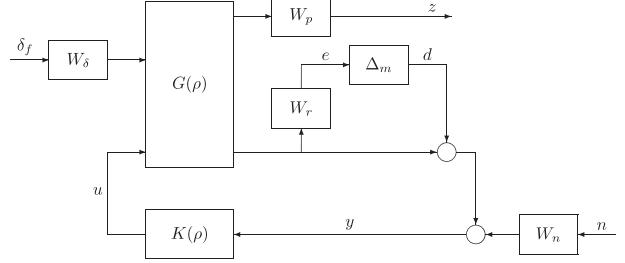


Fig 2. The closed-loop interconnection structure.

continuously differentiable function $X : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$, $X(\rho) = X^T(\rho) > 0$ such that

$$A^T T(\rho) X(\rho) + X(\rho) A(\rho) + \sum_{i=1}^s \left(\nu_i \frac{\partial X}{\partial \rho_i} \right) < 0 \quad (21)$$

for all $\rho \in \mathcal{P}$ and $|\dot{\rho}_i| \leq \nu_i$, $i = 1, 2, \dots, s$.

Applying the parameter-dependent stability concept, it is assumed that the derivative of parameters can also be measured in real time. This concept is less conservative than the quadratic stability because Eq. (21) is solved by finding a parameter-dependent $X(\rho)$ instead of a single X [16,17].

3.2. LPV control design

In this section the LPV control design is presented for a roll stability system. Consider the closed-loop system in Fig. 2, which includes the feedback structure of the model $G(\rho)$ and the compensator $K(\rho)$, 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, n is the measurement noise. In Fig. 2, δ_f is the steering angle as a disturbance signal, which is set by the driver and the z represents the performance outputs.

The nominal model is usually a good approximation of 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 are represented by W_r and Δ_m . Design models used for roll stability control typically exhibit high fidelity at lower frequencies ($\omega < 2$ rad/s), but they degrade rapidly at higher frequencies due to poorly modelled or neglected effects. Thus, W_r is selected as $W_r = 2.25(S + 20)/(S + 450)$.

The input scaling weight W_δ normalizes the steering angle to the maximum expected command. It is selected $5\pi/180$, which corresponds to a 5° steering angle command. W_n is selected as a diagonal matrix, which accounts for sensor noise models in the control design.

The noise weights is chosen 0.01 m/s^2 for the lateral acceleration and $0.01^\circ/\text{sec}$ for the derivative of roll angle $\dot{\phi}$.

The weighting function W_p represents the performance outputs, and it contains the components W_{p_a} , $W_{p_{F_z}}$, $W_{p_{F_b}}$, and W_{p_u} . The purpose of the weighting functions is to keep the lateral acceleration, the lateral load transfers and the control inputs 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:

$$W_{p_a} = \phi_a \frac{(s/2000 + 1)}{(s/12 + 1)}, \quad (22)$$

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 $W_{p_{F_z}}$ is selected as $\text{diag}(1/10^2, 1/10^3)$ for control design, which means that the maximal gain of the lateral load transfers can be 10^2 in a frequency domain for front axle and 10^3 for rear axle. The W_{p_u} is a diagonal matrix with diagonal entries $1/20$, which correspond to the front and rear control torque generated by active anti-roll bars. The weight $W_{p_{F_b}}$ for the brake force is $1/10$. The reason for keeping the control signals small is to avoid the actuator saturation.

The ϕ_a is gain, which reflects the relative importance of the lateral acceleration in the LPV control design. A large gain ϕ_a corresponds to a design that avoids the rollover situation. Choosing ϕ_a small corresponds to a vehicle in a normal driving situation in which the minimization of lateral acceleration is not needed. Consequently, when the acceleration is not critical the weighting function should be small and when the acceleration has reached the critical value the weight should be large to avoid the rollover.

In order to take into consideration such a nonlinear function of control 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. Basically, the rollover 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 lift off first. 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, that is, the function of R_r . The parameter-dependent gain ϕ_a captures the relative importance of the acceleration response. When R_r is small, that is, when the vehicle is

not in an emergency, $\phi_a(R_r)$ is small, indicating that the LPV control should not focus on minimizing acceleration, it should only generate lateral displacement moment by active anti-roll bars. On the other hand, when R_r approaches the critical value or the suspension has reached its physical limit, $\phi_a(R_r)$ is large, indicating that the control 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 $\phi_a(R_r)$ in Eq. (22), is as follows:

$$\phi_a(R_r) = \begin{cases} 0 & \text{if } |R_r| < R_1, \\ (2/(R_2 - R_1))(|R_r| - R_1) & \text{if } R_1 \leq |R_r| \leq R_2, \\ 2 & \text{otherwise.} \end{cases} \quad (23)$$

R_1 defines the critical status when the vehicle is close to the rollover situation, that is, all wheels are in the ground but the lateral tire force of the inner wheels tends to zero or the suspension has reached its physical limit and the active anti-roll bars are not capable of generating more stabilizing moment. The closer R_1 is to 1 the later the control will be activated. parameter R_2 shows how fast the control 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. The function $\phi_a(R_r)$ is illustrated in Fig. 3.

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 parameter v is measured directly, while the parameter R_r can be calculated by using the measured roll angle of the unsprung mass $\phi_{t,r}$.

The quadratic LPV γ -performance problem is to choose the parameter-varying control matrices $A_K(\rho)$, $B_K(\rho)$, $C_K(\rho)$, $D_K(\rho)$ in such a way that the

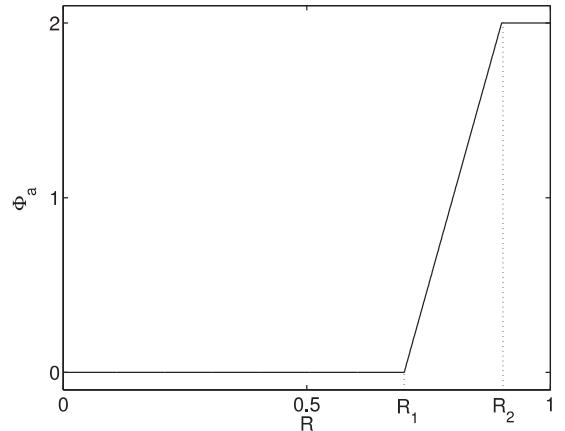


Fig 3. Parameter-dependent gain $\Phi_a(R)$.

resulting closed-loop system is quadratically stable and the induced \mathcal{L}_2 norm from w to z is less than γ . The form of LPV control $K(\rho)$ is as follows:

$$\begin{bmatrix} \dot{x}_K(t) \\ u(t) \end{bmatrix} = \begin{bmatrix} A_K(\rho(t)) & B_K(\rho(t)) \\ C_K(\rho(t)) & D_K(\rho(t)) \end{bmatrix} \begin{bmatrix} x_K(t) \\ y(t) \end{bmatrix}, \quad (24)$$

where $(A_K(\rho), B_K(\rho), C_K(\rho), D_K(\rho)) : \mathcal{R}^S \rightarrow (\mathcal{R}^{m \times m}, \mathcal{R}^{m \times n_y}, \mathcal{R}^{n_u \times m}, \mathcal{R}^{n_y \times n_y})$ continuous bounded matrix functions.

The quadratic LPV γ -performance problem is solvable if there exist an integer $m \geq 0$, a matrix $W : \mathcal{R}^S \rightarrow \mathcal{R}^{(n+m) \times (m+n)}$, $W(\rho) = W^T(\rho) > 0$ and continuous bounded matrix functions $(A_K(\rho), B_K(\rho), C_K(\rho), D_K(\rho))$ in such a way that

$$\begin{bmatrix} A_{\text{clp}}^T(\rho)W(\rho) + W(\rho)A_{\text{clp}}(\rho) & W(\rho)B_{\text{clp}}(\rho) & \gamma^{-1}C_{\text{clp}}^T(\rho) \\ B_{\text{clp}}(\rho)W(\rho) & -I_{n_d} & \gamma^{-1}D_{\text{clp}}^T(\rho) \\ \gamma^{-1}C_{\text{clp}}^T(\rho) & \gamma^{-1}D_{\text{clp}}^T(\rho) & -I_{n_e} \end{bmatrix} < 0, \quad (25)$$

for all $\rho \in \mathcal{P}$, where the matrices $A_{\text{clp}}, B_{\text{clp}}, C_{\text{clp}}, D_{\text{clp}}$ are the state space matrices in the closed-loop system.

The existence of a compensator that solves the quadratic LPV γ -performance problem can be expressed as the feasibility of a set of linear matrix inequalities (LMIs), which can be solved numerically.

Theorem 1. Given a compact set $\mathcal{P} \subset \mathcal{R}^S$, the performance level γ and the LPV system (19), with restriction $D_{11}(\rho) = 0$, the parameter-dependent γ -performance problem is solvable if and only if there exist a continuously differentiable function $X : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$, and $Y : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$ in such a way that for all $\rho \in \mathcal{P}$, $X(\rho) = X^T(\rho) > 0$, $Y(\rho) = Y^T(\rho) > 0$ and

$$\begin{bmatrix} \hat{A}(\rho)X(\rho) + X(\rho)\hat{A}^T(\rho) - \sum_{i=1}^s \left(\nu_i \frac{\partial X}{\partial \rho_i} \right) - B_2(\rho)B_2^T(\rho) & X(\rho)C_1^T(\rho) & \gamma^{-1}B_1(\rho) \\ C_1(\rho)X(\rho) & -I_{n_e} & 0 \\ \gamma^{-1}B_1^T(\rho) & 0 & -I_{n_d} \end{bmatrix} < 0, \quad (26)$$

$$\begin{bmatrix} \tilde{A}^T(\rho)Y(\rho) + Y(\rho)\tilde{A}(\rho) + \sum_{i=1}^s \left(\nu_i \frac{\partial Y}{\partial \rho_i} \right) - C_2^T(\rho)C_2(\rho) & Y(\rho)B_1(\rho) & \gamma^{-1}C_1^T(\rho) \\ B_1^T(\rho)Y(\rho) & -I_{n_d} & 0 \\ \gamma^{-1}C_1(\rho) & 0 & -I_{n_e} \end{bmatrix} < 0, \quad (27)$$

$$\begin{bmatrix} x(\rho) & \gamma^{-1}I_n \\ \gamma^{-1}I_n & Y(\rho) \end{bmatrix} \geq 0, \quad (28)$$

where $\hat{A}(\rho) = A(\rho) - B_2(\rho)C_1(\rho)$, $\tilde{A}(\rho) = A(\rho) - B_1(\rho)C_2(\rho)$.

The state space representation of the LPV control $K(\rho)$ is constructed from the solutions $X(\rho)$ and $Y(\rho)$ of the LMI optimization problem. Theorem 1 and its proof are found in [16,17].

The constraints set by the LMIs in Theorem 1 are infinite dimensional, as is the solution space. Therefore some approximations are needed in order to compute solutions. First, the variables are $X, Y : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$, which restricts the search to the span of a collection of known scalar basis functions. Select scalar continuous differentiable basis functions $\{g_i : \mathcal{R}^S \rightarrow \mathcal{R}\}_{i=1}^{N_x}, \{f_j : \mathcal{R}^S \rightarrow \mathcal{R}\}_{j=1}^{N_y}$, then the variables in Theorem 1 can be parameterized as

$$X(\rho) = \sum_{i=1}^{N_x} g_i(\rho)X_i, \quad (29)$$

$$Y(\rho) = \sum_{j=1}^{N_y} f_j(\rho)Y_j. \quad (30)$$

Currently, there is no analytical method to select the basis functions, namely g_i and f_j . An intuitive rule for basis function selection is to use those present in the open-loop state space data. In our case, several power series $\{1, \rho^2\}$ of the scheduling parameters are chosen based on the lowest closed-loop \mathcal{L}_2 norm achieved.

Second, the infinite-dimensionality of the constraints is relieved by approximating the parameter set \mathcal{P} by a finite, sufficiently fine grid $\mathcal{P}_{\text{grid}} \subset \mathcal{P}$. For the interconnection structure, \mathcal{H}_∞ compensators are synthesized for several value of velocity in a range $v = [40, \dots, 100] \text{ kmph}$. The spacing of the grid points is based upon how well the \mathcal{H}_∞ point designs perform for plants around the design point. Seven grid points

are selected for the velocity scheduling parameter design space. The rear load transfer parameter space is gridded as $R_r = [0, R_1, R_2, 1]$. Weighting functions for both the performance and robustness specifications

are defined at all of the grid points. With respect to the robustness requirement, the same frequency weighting functions are applied in the entire parameter space and the effect of the scheduling parameter is ignored. It is a reasonable engineering assumption, since the uncertainty (i.e. unmodelled dynamics, parametric uncertainties) does not depend on the forward velocity. In the LPV control design based on the H_∞ method, weighting functions for both the performance and robustness specifications are defined. If weighting functions are selected and Eqs (26)–(28) are applied, the gamma iteration results in an optimal gamma value and an optimal controller. Obviously, if the weighting functions were changed, another optimal gamma and another optimal controller would be gained. By assuming an uncertainty structure and its magnitude the weighting for uncertainty is fixed. The weighting for performances are defined by achievable intervals using the performance demands. After a tuning process, such weighting functions for performances which result in the best roll stability are selected. In this paper the optimal solution is examined, and the suboptimal solutions are not analyzed in detail.

4. Simulation Examples

4.1. Examinations of Various Vehicle Maneuvers

In this section, illustrative examples are shown for the combined control mechanism, which is based on the LPV control. The results of the combined control are compared with the controlled systems which only use active anti-roll bars and which only use an active brake mechanism. In order to compare the combined control mechanism with the other solutions, two control systems are designed. In the case when the control only uses active anti-roll bars with control inputs u_f and u_r . In the case of an active brake, the control is designed by using control input ΔF_b . The performance specifications and the model uncertainty are selected the same way in all cases. Various vehicle maneuvers are examined: a double lane change and a cornering maneuver. The values of the vehicle parameters are found in Table 2.

In the first simulation example, a double lane change maneuver is performed. This maneuver is often used to avoid an obstacle in an emergency. 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 in an emergency on a road. The vehicle velocity is 75 km/h. The steering angle input is generated in such a way that the vehicle with no roll

Table 2. Parameters of the yaw–roll model.

Parameter	Value
m_s	12487 kg
m_{uf}, m_{ur}	706 kg, 1000 kg
m	14193 kg
h	1.15 m
$h_{uf} h_{ur}$	0.53, 0.53 m
r	0.83 m
C_f, C_r	$582 \times 10^3, 783 \times 10^3$ kN/rad
k_f, k_r	$380 \times 10^3, 684 \times 10^3$ kNm/rad
b_f, b_r	$100 \times 10^3, 100 \times 10^3$ kNm/rad
$k_{t,f}, k_{t,r}$	$2060 \times 10^3, 3337 \times 10^3$ kNm/rad
I_{xx}	24201kgm^2
I_{xz}	4200kgm^2
I_{zz}	34917kgm^2
l_f, l_r	1.95, 1.54 m
l_w	0.93 m
μ	1

control comes close to a rollover situation during the maneuver and its normalized load transfers are above the value ± 1 . Note that the yaw – roll model is only valid if the normalized load transfers are below the value ± 1 for both axles. In order to avoid the unrealistic change in the steering angle, a ramp signal is applied which reaches the maximum value (3.5°) in 0.5 s and filtered at 4 rad/s to represent the finite bandwidth of the driver.

Figure 4 shows the time responses in cases when active anti-roll bars (dash-dot), an active breaking mechanism (dash) and a combined roll control system (solid) are used. These figures also show the lateral acceleration and the roll angle of the sprung mass. Using only active anti-roll bars the control is not able to decrease the lateral acceleration, that is, it does not have a direct effect on the change in acceleration. In the case of the active anti-roll bars the vehicle rolls into the corner. Hence, this motion of the vehicle generates a stabilizing lateral displacement moment, which balances the destabilizing overturning moment caused by lateral acceleration. It can be seen that using advanced suspension, the roll angle has a -180° phase shifting compared to the passive suspension. Note, that the relative roll angle of the suspension ($\phi - \phi_{t,i}$) is not within the acceptable limit, which is about 7° – 8° . In the case of an active brake, the lateral acceleration is the same as when the normalized load transfers do not reach the critical value, which is determined by R_1 but once the critical value is exceeded the control algorithm is activated and the active brake system reduces the lateral acceleration. In the example the constants are selected as $R_1 = 0.85$ and $R_2 = 0.95$. The time when the brake control is activated can be seen in the brake force figure, which

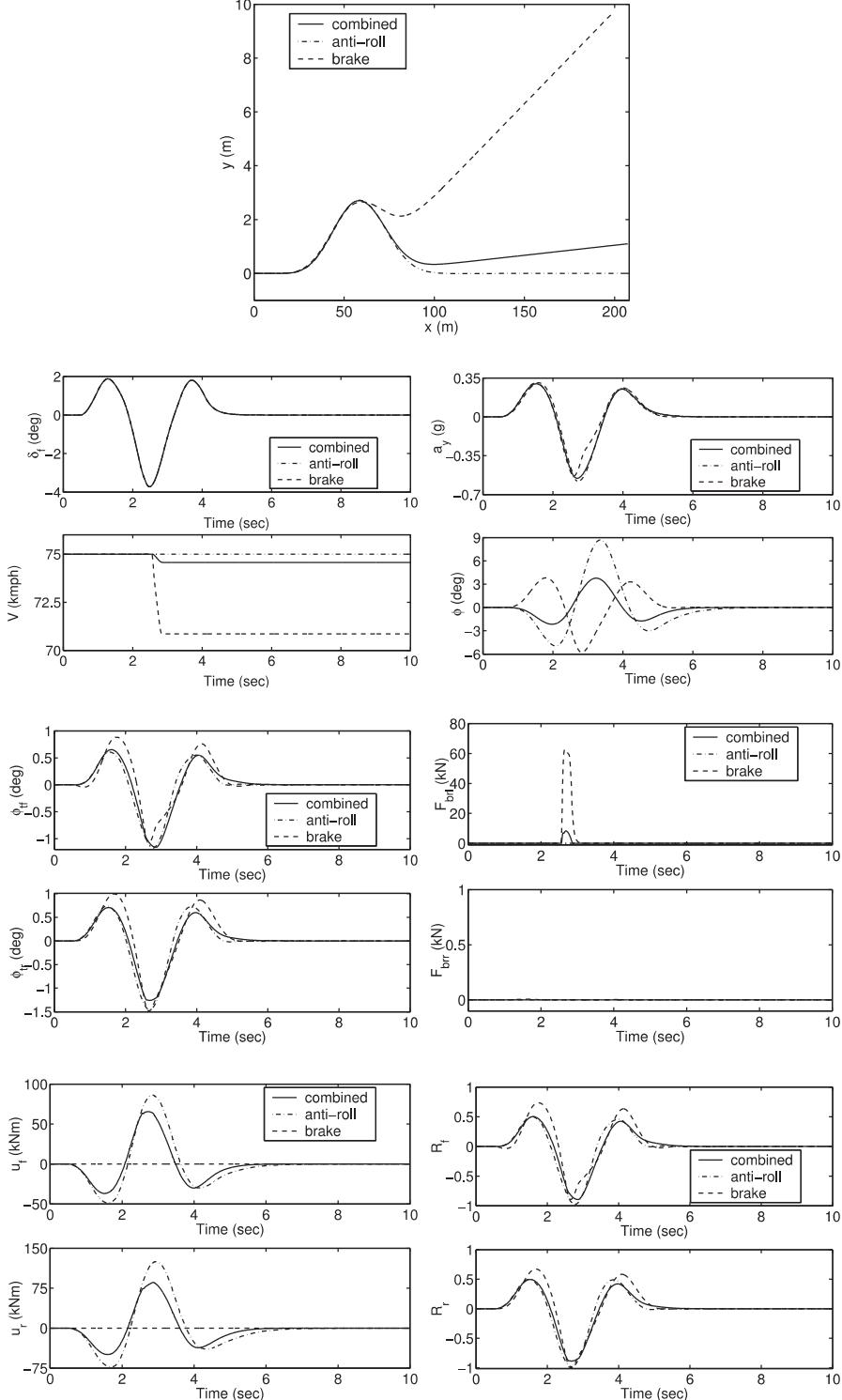


Fig. 4. Time responses to double lane change steering input when different control methods are applied.

shows that the rear-left wheel is braked to avoid the rollover of the vehicle.

In the case of the combined control the magnitude of the acceleration is similar to the previous cases. The

reason for this is that only the active anti-roll bars decrease the acceleration and the brake control does not work until the normalized load transfer has reached the critical value R_1 . Hence the combined

control can be considered as simple active anti-roll bars when the normalized load transfer is less than R_1 . However, the required control action with respect to brake force is less than the active brake control. In the case of the combined roll control, only the active anti-roll bars work and generate a stabilizing lateral displacement moment when the normalized load transfer does not reach the critical value and the brake system is not activated. Hence, the brake force required to prevent the rollover of the vehicle is less than when using only the brake system. When the combined control and the brake mechanism are applied the roll angle of the sprung mass is in the same phase as in the uncontrolled case and the physical limit of the suspension travel is not exceeded. It can be observed that the critical value of the lateral acceleration is the second peak from the rollover point of view. The engineering interpretation of this phenomenon is that the vehicle generates larger lateral acceleration when the vehicle starts returning into the lane because the driver must set a double steering angle with -180° phase shifting to steer back the vehicle into its original position. As far as all of the three control structures are concerned, the roll angle of the unsprung masses at the front and rear axle is slightly different due to the different suspension parameters and the stiffness to load ratio.

The path of vehicle for all control can be seen in Fig. 4. In case of the advanced suspension the vehicle keeps the desired path. In the case of the brake control the real path is significantly different from the desired path due to the brake moment, which affects the yaw motion. The only limit to using the active brake mechanism is the saturation of the brake actuators. This means that the minimization of the lateral acceleration is restricted by the physical limit of the actuator. However, the only problem is that if too much unilateral brake force is applied the stability of the yaw motion is degraded. In order to avoid the degradation of yaw dynamics the combined control is used, in which case the real path is slightly different from the desired path because the brake force is less than in the case of the active brake control. In the case of braking control or in the case of combined control the deviation in path can be corrected by using the steering angle. In the case of combined control the deviation of the real and the desired path can be corrected by a small change in the steering angle. It is noted that changing the steering angle has negative effects on the roll movement. The greater the difference between the desired path and the real path is, the larger steering angle is needed, which increases the overturning moment. As in the case of combined control mechanism the real path is slightly different

from the desired path, a small correction is needed. Thus, the effect of the steering angle does not result in an overturning situation.

In the next example, the cornering responses of a single unit vehicle model travelling at 75 km/h can be seen. Fig. 5 shows the time responses in case when active anti-roll bars (dash-dot), an active breaking mechanism (dash) and a combined roll control system (solid) are used. The steering angle applied in the simulation is a step signal. The forward velocity is not constant during the maneuver because in the case of brake and combined control the brake force provided by the compensator decelerates the vehicle. In simulation the driver does not push down on the brake pedal, hence the only change in forward velocity is caused by the compensator. As the lateral acceleration increases, the normalized load transfer lifts up the rear axle more quickly than the front axle since the ratio of the effective roll stiffness to the axle load is greater at the drive axle. The normalized load transfers do not exceed the value ± 1 in all cases, which means that the lateral force on one of the curve inner side wheels will not become zero.

In the case of the combined control the vehicle rolls into the corner. This motion of the vehicle generates a stabilizing lateral displacement moment, which balances the destabilizing overturning moment caused by lateral acceleration. However, active anti-roll bars are not enough to prevent the rollover of the vehicle. Consequently, the normalized load transfers reach the critical value R_1 during the cornering maneuver so the brake control is activated and the brake system reduces the lateral acceleration. The time when the brake control is activated can be seen in the brake force figure, which shows that the rear right-hand side wheel is braked to avoid the rollover of the vehicle. In the case of active anti-roll bars the control torque required to prevent the rollover of the vehicle is much larger than in the case of the combined control. Using the combined active anti-roll bars and active brake control the relative roll angle is reduced significantly and it stays within the acceptable limits. However, it can be seen that when only active anti-roll bars are used the relative roll angle of the suspension ($\phi - \phi_{t,i}$) is not within the acceptable limits. The unacceptable suspension travel is caused by the increased control torque. The roll angle of the unsprung masses is slightly different due to the different suspension parameters and the stiffness to load ratio. The control torque is approximately -60 kNm at the front axle and -90 kNm at the rear axle and the brake force is 15 kN. In case of active anti-roll bars the required control action is -150 kNm at front and -250 kNm at rear axle.

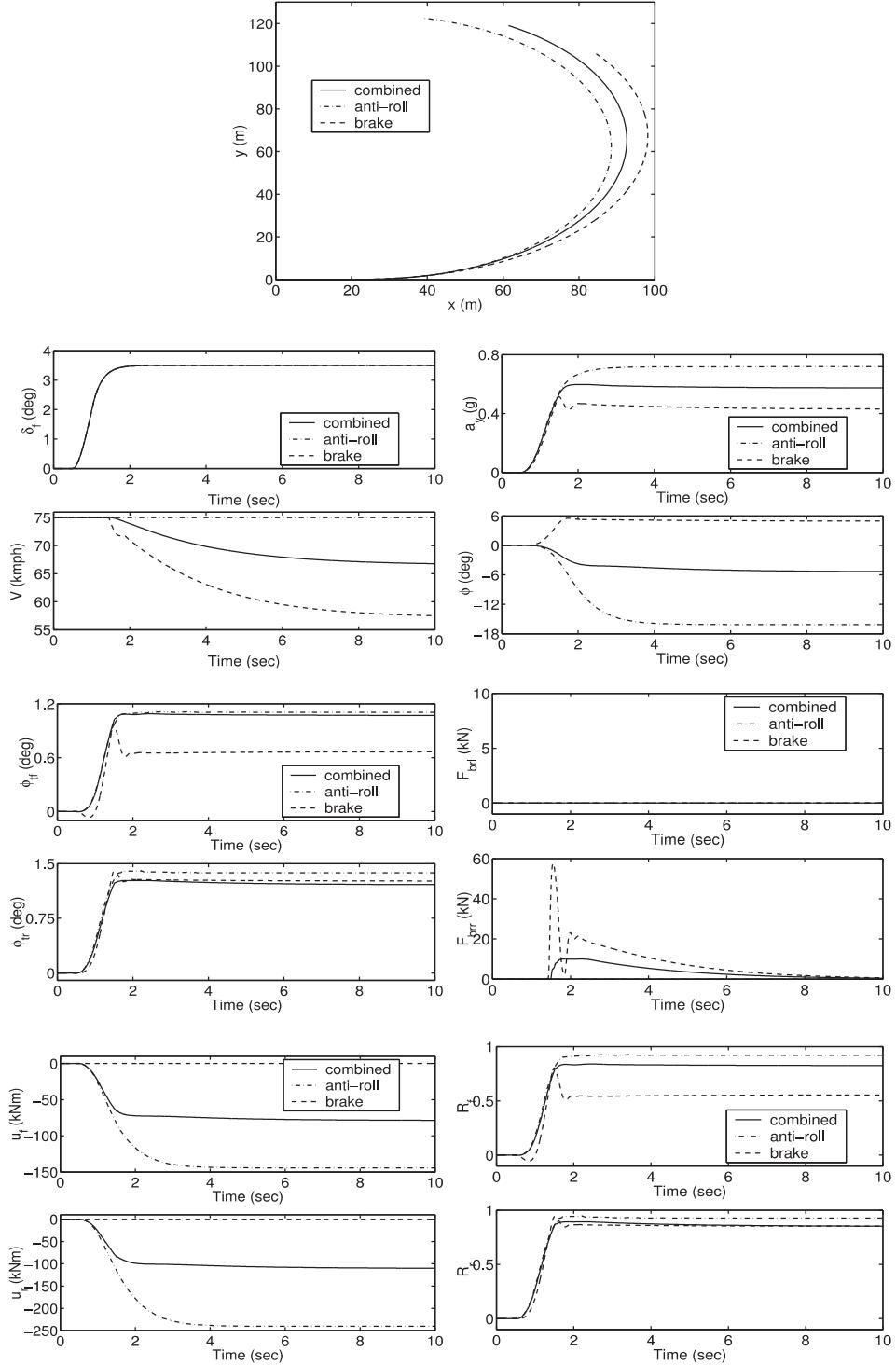


Fig. 5. Time responses to cornering when different control methods are applied.

4.2. Comparison of Combined Compensators

In this section the controlled systems using different design methods are compared. In the case of the control design based on the single Lyapunov function

(SLF), it is assumed that the closed-loop system is also stable for infinite fast parameter variations. In practice, the derivatives of the LPV parameters are finite. Hence, the control design based on SLF leads to a conservative control. The conservatism of the control

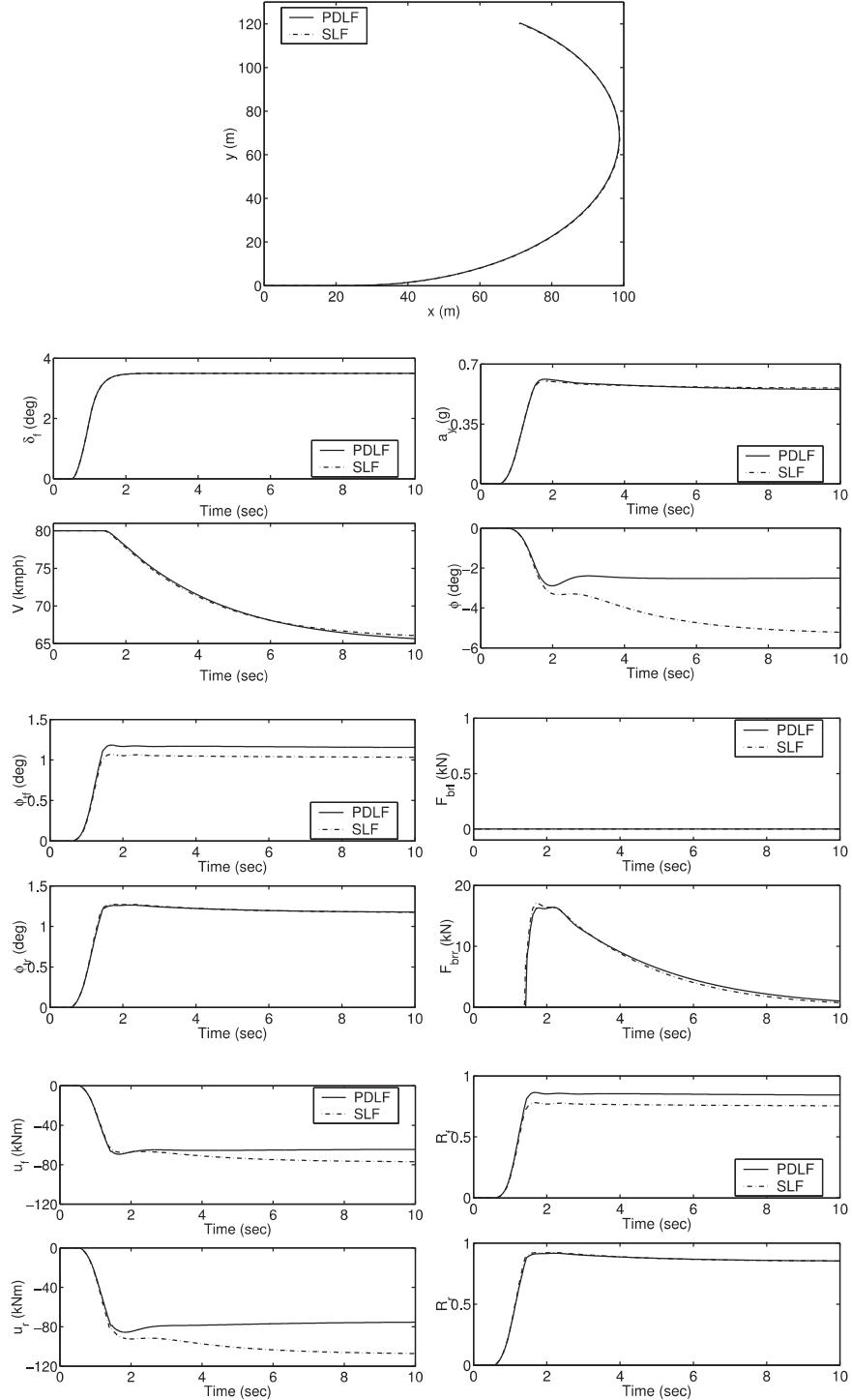


Fig. 6. Time responses to cornering when PDLF and SLF are applied.

can be reduced by applying parameter-dependent stability concept, in which the control is constructed by parameter-dependent Lyapunov function (PDLF). In this case it is assumed that the derivative of parameters can be measured in real time and the control also receives the derivatives of the scheduling parameters.

In the following, the cornering responses of the closed-loop system, which apply compensators based on SLF and PDLF, can be seen.

Fig. 6 shows the time responses when these compensators are used. The initial velocity of the vehicle is 75 km/h in the simulations. The steering angle applied

in the simulation is a step signal. The normalized load transfers do not exceed the value ± 1 in all cases, which means that the vehicle does not lose roll stability. In the case of the SLF compensator the required control torque is larger than in the case of the PDLF compensator. The control torque difference between the two compensators is approximately -20 kNm at the front axle and -30 kNm at the rear axle. The brake forces are the same in both cases, approximately 15 kN. The relative roll angle of the suspension ($\phi - \phi_{t,i}$) is within the acceptable limits, but using the SLF compensator it has a larger value caused by larger control torque at the front axle. The simulation results show conservative behavior of the SLF compensator because in this case the compensator needs larger control torque than in the case of PDLF compensator in the same situation to prevent the rollover of the single vehicle.

4.3. The Elimination of Chattering

The control mechanism proposed in the combined control structure can be considered as a switching system. The active anti-roll bars are primarily used to prevent the rollover of the vehicle; the brake system is only activated when the vehicle is close to the rollover situation. In practice, using such switching structures a chattering may occur. Chattering causes small amplitude oscillation with high frequency around the switching point, which may degrade the performance properties of the vehicle. In our case the switching point is the critical normalized load transfer defined as R_1 , and the brake system is switched on and off at this value.

In the following, one possible extension of the combined control structure is proposed for the chattering. In order to eliminate chattering, a hysteresis characteristic is applied with respect to the critical value of the load transfer, R_1 . It means that the value of R_1 must be larger when the brake system is switched on than when it is switched off. Such a load transfer hysteresis is defined as

$$R_1 = R_{\text{nom}} + \text{sgn}(\dot{R}_r)/w_h, \quad (31)$$

where R_{nom} is a nominal value of the switching point. w_h is the parameter with the width of hysteresis window can be adjusted.

In Eq. (31), it is assumed that, the derivative of load transfer \dot{R}_r is also computed in real time. The sign of \dot{R}_r can be used to deduce the direction of the load transfer change. If the sign of \dot{R}_r is positive the load transfer is increasing and the brake system is switched on above the R_{nom} . However, when the sign of \dot{R}_r is negative, that is the load transfer is decreasing, the

brake system is switched off at a smaller value than its nominal value. The $\text{sgn}(\dot{R}_r)$ can be used as an additional scheduling parameter in the control design. In the design procedure, the possible values of $\text{sgn}(\dot{R}_r)$ are selected $\{-1, 0, 1\}$.

4.4. Fault Adaptive Control

Now, we shall examine the effect of varying the design parameter R_1 on the nonlinear response of combined rollover control. Note that for a fixed value of the parameter R_2 , the parameter R_1 determines at what point and how fast the brake system is activated to prevent the rollover. Small values of R_1 correspond to a brake system that is activated early and gradually, whereas large values of R_1 correspond to a brake system that is activated rapidly when the normalized load transfer is close to its critical value. In Fig. 7 the peak lateral acceleration against forward velocity is plotted during double lane change maneuver. The R_2 is fixed at 0.95 and R_1 varies. The dash-dot, dashed and solid lines correspond to $R_1 = 0.7$, $R_1 = 0.8$ and $R_1 = 0.9$, respectively. With $R_1 = 0.7$ in the combined roll control system there is a gradual brake control, whereas when $R_1 = 0.9$, the brake system is not used until the normalized load transfer R_r equals 0.9, and the response of the yaw-roll model is the same as in the case when only active anti-roll bars are used. However the brake system prevents rapidly the rollover after $R \geq 0.9$. Thus the design parameters R_1 and R_2 can be used to shape the nonlinear response characteristics of the control.

One possible extension of the framework proposed here is to design a fault adaptive rollover control system. At the end of the previous section we showed how the design parameters R_1 and R_2 can be used to

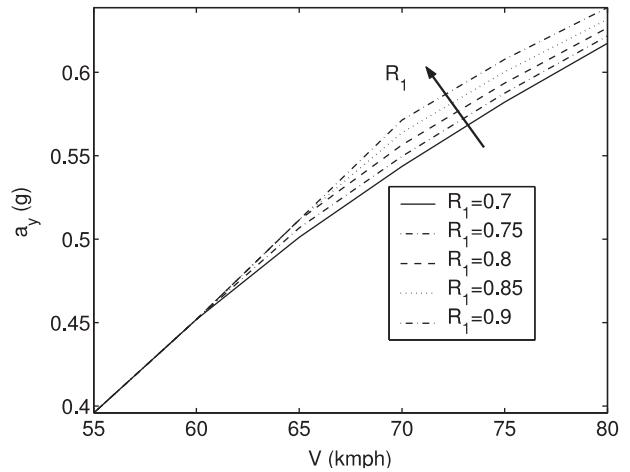


Fig. 7. Effect of parameter R_1 on lateral acceleration.

shape the nonlinear response characteristics of the control. In a non-faulty case, which means both the active anti-roll bars work well, it would be preferable to choose R_1 large, which is shown by the dashed line in Fig. 7. This corresponds to an active brake system that is not used for most of the time and activated very rapidly as the normalized load transfer exceeds the critical value determined by R_1 . In a non-faulty case it is assumed that the controlled brake system is used in an emergency most cases. However, this would result in a large lateral acceleration until the critical R_1 is reached. This would be a small price for the stability of yaw motion. Because until the critical R_1 has been reached only the active anti-roll bars, which does not affect directly the yaw dynamics of the vehicle, are used. On the other hand, if a hydraulic actuator fault occurs in the system it would be preferable to choose R_1 small and R_2 fixed (the dash-dot line in Fig. 7). This corresponds to a combined control where the range of operation of the brake system is extended and the wheels are decelerated gradually rather than rapidly if the normalized load transfer has reached R_1 . It is assumed that the actuator fault can occur as a loss of effectiveness, that is, its power is reduced by some percent. It means that both control inputs (the active anti-roll bars and the brake system) are able to work simultaneously but the hydraulic actuator does not have maximum performance. It is a reasonable assumption in many cases because the failure appearance indicates an effectiveness failure in an early stage. Thus, the design parameter R_1 can be chosen as a scheduling parameter based on the fault information. The fault information is provided by a fault detection filter. This adaptive feature would lead to an enhanced roll stability which uses fault information when a fault occurs in the hydraulic actuators.

5. Conclusion

This paper is concerned with a combined control structure with active anti-roll bars and an active brake control. The control design is based on the LPV modelling, in which the forward velocity and the normalized lateral load transfer at the rear are chosen as scheduling parameters. In the designed control the velocity of the vehicle, the performance specifications and the model uncertainties are taken into consideration. In this control structure the active anti-roll bars generate stabilizing lateral moment and when the normalized load transfer has reached a critical value the brake control is also activated in order to prevent the rollover situation. This control structure can be considered as a fault-tolerant system. Different

maneuver situations are analyzed and the combined control structure is compared with simple active anti-roll bars and an active brake control. An extension of the combined control structure is proposed for the solution of chattering problem. The advantage of this structure is that it incorporates a fault adaptive control mechanism, in which the design parameters can be tuned when a fault occurs in the hydraulic actuator.

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