

TOWARDS GLOBAL CHASSIS CONTROL BY INTEGRATING THE BRAKE AND SUSPENSION SYSTEMS

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Abstract: A control structure that integrates active suspensions and an active brake is proposed to improve the safety of vehicles. The design is based on an \mathcal{H}_∞ control synthesis extended to LPV systems and uses a parameter dependent Lyapunov function. In an emergency, such as an imminent rollover, the safety requirement overwrites the passenger comfort demand by tuning the performance weighting functions associated with the suspension systems. If the emergency persists active braking is applied to reduce the effects of the lateral load transfers and thus the rollover risk. The solution is facilitated by using the actual values of the so-called normalized lateral load transfer as a scheduling variable of the integrated control design. The applicability of the method is demonstrated through a complex simulation example containing vehicle maneuvers.

Keywords: LPV modeling and control, performance specifications, uncertainty, safety operation, passenger comfort, automotive.

1. INTRODUCTION

These days road vehicles contain several individual active control mechanisms that solve a large number of required control tasks. These control systems contain a lot of hardware components, such as sensors, actuators, communication links, power electronics, switches and micro-processors. In traditional control systems the vehicle functions to be controlled are designed and implemented separately. This means that control hardware is grouped into disjoint subsets with sensor information and control demands handled in parallel processes. However, these approaches can lead to unnecessary hardware redundancy. Al-

though in the design of the individual control components only a subset of the full vehicle dynamics is considered these components influence the entire vehicle. Thus in the operation of these autonomous control systems interactions and conflicts may occur that might overwrite the intentions of the designers concerning the individual performance requirements.

The aim of the integrated control methodologies is to combine and supervise all controllable subsystems affecting the vehicle dynamic responses in order to ensure the management of resources. The flexibility of the control systems must be improved by using plug-and-play extensibility, see e.g. (Gordon *et al.*, 2003). The central purpose of vehicle control is not only to improve functionality, but also simplify the electric architecture of the vehicle. Complex and overloaded networks are the bottle-neck of functional improvements

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and high complexity can also cause difficulties in reliability and quality. The solution might be the integration of the high level control logic of sub-systems. It enables designers to reduce the number of networks and create a clear-structured vehicle control strategy. Several schemes concerned with the possible active intervention into vehicle dynamics to solve different control tasks have been proposed. These approaches employ active anti-roll bars, active steering, active suspensions or active braking, see e.g. (Alleyne and Hedrick, 1995; Fialho and Balas, 2000; Hedrick *et al.*, 1997; Kim and Park, 2004; Nagai *et al.*, 1998; Odenthal *et al.*, 1999; Sampson and Cebon, 2003; Shibahata, 2005; Trächtler, 2004).

In this paper a control structure that integrates active suspensions and an active brake is proposed to improve the safety of vehicles. The active suspension system is primarily designed to improve passenger comfort, i.e. to reduce the effects of harmful vibrations on the vehicle and passengers. However, the active suspension system is able to generate a stabilizing moment to balance an overturning moment during vehicle maneuvers in order to reduce the rollover risk, (Gáspár and Bokor, 2005). Although the role of the brake is to decelerate the vehicle, if the emergency persists, the effects of the lateral tire forces can be reduced directly by applying unilateral braking and thus reducing the rollover risk (Gáspár *et al.*, 2005; Palkovics *et al.*, 1999). This paper is an extension of the principle of the global chassis control, which has been proposed in (Zin *et al.*, 2006).

The controller uses the actual values of the so-called normalized lateral load transfer R as a scheduling variable of the integrated control design. When a rollover is imminent the values of R increase and reach a lower critical limit, and then suspension forces must be generated to create a moment at the front and the rear to enhance the stability of the vehicle. When this dangerous situation persists and R reaches the upper critical limit the active brake system must generate unilateral brake forces in order to reduce the risk of the rollover. The goal of the control system is to use the active suspension system all the time to improve passenger comfort and road holding and activate the braking system only when the vehicle comes close to rolling over. In an emergency the safety requirement overwrites the passenger comfort demand by tuning the performance weighting functions associated with the suspension systems. Then a functional reconfiguration of the suspension system is carried out in order to generate stabilizing moments to balance an overturning moment during vehicle maneuvers.

In this paper the control-oriented model design has been carried out in a Linear Parameter Varying (LPV) framework that allows us to take into consideration the nonlinear effects in the state space description. Thus the model structure is nonlinear in the parameter functions, but it remains linear in the states. In the control design the performance specifications for rollover and suspension problems, and also the model uncertainties are taken into consideration. The design is based on an \mathcal{H}_∞ control synthesis extended to LPV systems that use parameter dependent Lyapunov functions, (Balas *et al.*, 1997; Wu *et al.*, 1996).

The structure of the paper is as follows. After a short introduction in Section 2 the control oriented modeling for rollover prevention and suspension problems is presented. In Section 3 the weighting strategy applied for the parameter-dependent LPV control is presented. In Section 4 the operation of the integrated control system is demonstrated through a simulation example. Finally, Section 5 contains some concluding remarks.

2. AN LPV MODELING FOR THE CONTROL DESIGN

The combined yaw-roll dynamics of the vehicle is modeled by a three-body system, where m_s is the sprung mass, m_f and m_r are the unsprung masses at the front and at the rear including the wheels and axles and m is the total vehicle mass.

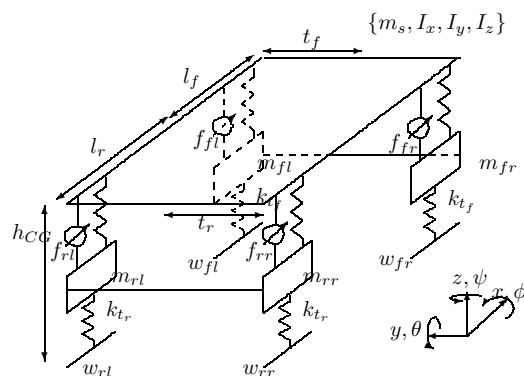


Fig. 1. Vertical dynamics of the full-car model.

The suspension system, which is shown in Figure 1, contains springs, dampers and actuators between the body and the axle on both sides at the front and rear. The suspension stiffnesses, the tire stiffnesses and the suspension dampers at the front and rear are denoted by k_f , k_r , k_{t_f} , k_{t_r} , b_f , b_r , respectively. The front and rear displacements at both sides of the sprung and the unsprung masses are denoted by x_{1fl} , x_{1fr} , x_{1rl} , x_{1rr} and x_{2fl} , x_{2fr} , x_{2rl} , x_{2rr} , respectively. In the model,

the disturbances w_{fl} , w_{fr} , w_{rl} , w_{rr} are caused by road irregularities.

The yaw and roll dynamics of the vehicle is shown in Figure 2. The roll moment of the inertia of the sprung mass and of the yaw-roll product is denoted by I_{xx} and I_{xz} while I_{yy} is the the pitch moment of inertia and I_{zz} is the yaw moment of inertia. The total axle loads are F_{zl} and F_{zr} . The lateral tire forces in the direction of the wheel-ground contact are denoted by F_{yf} and F_{yr} . h is the height of CG of the sprung mass and h_{uf} , h_{ur} are the heights of CG of the unsprung masses, ℓ_w is the half of the vehicle width and r is the height of the roll axis from the ground. β denotes the side slip angle of the sprung mass, ψ is the heading angle, ϕ is the roll angle, $\dot{\psi}$ denotes the yaw rate and θ the pitch angle. The roll angle of the unsprung mass at the front and at the rear axle are denoted by $\phi_{t,f}$ and $\phi_{t,r}$, respectively. δ_f is the front wheel steering angle, a_y denotes the lateral acceleration and z_s is the heave displacement while v stands for the forward velocity.

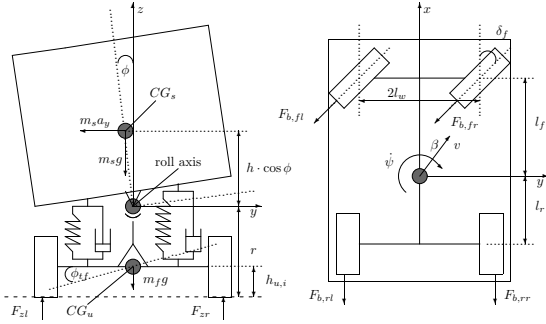


Fig. 2. Yaw and roll dynamics of the full-car model

First the modeling for suspension purposes is formalized. The vehicle dynamical model, i.e. the heave, pitch and roll dynamics of the sprung mass and the front and rear dynamics of the unsprung masses at both sides of the front and rear, is as follows:

$$\begin{aligned}
m_s \ddot{z}_s &= k_f(\Delta_{fl} + \Delta_{fr}) + k_r(\Delta_{rl} + \Delta_{rr}) \\
&\quad + b_f(\dot{\Delta}_{fl} + \dot{\Delta}_{fr}) + b_r(\dot{\Delta}_{rl} + \dot{\Delta}_{rr}) \\
&\quad - f_{fl} - f_{fr} - f_{rl} - f_{rr} \\
I_{yy} \ddot{\theta} &= k_f l_f(\Delta_{fl} + \Delta_{fr}) + k_r l_r(\Delta_{rl} + \Delta_{rr}) \\
&\quad + b_f l_f(\dot{\Delta}_{fl} + \dot{\Delta}_{fr}) - b_r l_r(\dot{\Delta}_{rl} + \dot{\Delta}_{rr}) \\
&\quad - (f_{fl} + f_{fr}) l_f + (f_{rl} + f_{rr}) l_r \\
I_{xx} \ddot{\phi} &= k_f \ell_w(\Delta_{fl} - \Delta_{fr}) + k_r \ell_w(\Delta_{rl} - \Delta_{rr}) \\
&\quad + b_f \ell_w(\dot{\Delta}_{fl} - \dot{\Delta}_{fr}) + b_r \ell_w(\dot{\Delta}_{rl} - \dot{\Delta}_{rr}) \\
&\quad - (f_{fl} - f_{fr}) \ell_w - (f_{rl} - f_{rr}) \ell_w \\
m_f \ddot{x}_{2fl} &= -k_f \Delta_{fl} + k_{tf} \Delta_{wfl} + b_f \dot{\Delta}_{fl} - f_{fl} \\
m_f \ddot{x}_{2fr} &= -k_f \Delta_{fr} + k_{tf} \Delta_{wfr} + b_f \dot{\Delta}_{fr} - f_{fr} \\
m_r \ddot{x}_{2rl} &= -k_r \Delta_{rl} + k_{tr} \Delta_{wrl} + b_r \dot{\Delta}_{rl} - f_{rl} \\
m_r \ddot{x}_{2rr} &= -k_r \Delta_{rr} + k_{tr} \Delta_{wrr} + b_r \dot{\Delta}_{rr} - f_{rr}
\end{aligned}$$

with the following notations: with $\Delta_{fl} = -x_{1fl} + x_{2fl}$, $\Delta_{fr} = -x_{1fr} + x_{2fr}$, $\Delta_{rl} = -x_{1rl} + x_{2rl}$, $\Delta_{rr} = -x_{1rr} + x_{2rr}$, $\Delta_{wfl} = x_{2fl} - w_{fl}$, $\Delta_{wfr} = x_{2fr} - w_{fr}$, $\Delta_{wrl} = x_{2rl} - w_{rl}$ and $\Delta_{wrr} = x_{2rr} - w_{rr}$.

The state space representation of the suspension system is the following:

$$\dot{x}_s = A_s x_s + B_{1s} d_s + B_{2s} u_s, \quad (1)$$

with the state vector $x_s = [x_1 \dot{x}_1]^T$, where $x_1 = [z_s \phi \theta x_{2fl} x_{2fr} x_{2rl} x_{2rr}]^T$. The input signals is $u_s = [f_{fl} f_{fr} f_{rl} f_{rr}]^T$ and $d_s = [w_{fl} w_{fr} w_{rl} w_{rr}]^T$ is the disturbance.

Second, the modeling for the rollover problem is formalized. This structure includes two control mechanisms which generate control inputs: the roll moments between the sprung and unsprung masses, generated by the active suspensions u_{af} , u_{ar} , and the difference in brake forces between the left and right-hand sides of the vehicle ΔF_b . The differential equations of the yaw-roll dynamics are formalized:

$$\begin{aligned}
m v(\dot{\beta} + \dot{\psi}) - m_s h \ddot{\phi} &= F_{yf} + F_{yr} \\
-I_{xz} \ddot{\phi} + I_{zz} \ddot{\psi} &= F_{yf} l_f - F_{yr} l_r + l_w \Delta F_b \\
(I_{xx} + m_s h^2) \ddot{\phi} - I_{xz} \ddot{\psi} &= m_s g h \phi + m_s v h(\dot{\beta} + \dot{\psi}) \\
&\quad - k_f(\phi - \phi_{tf}) - b_f(\dot{\phi} - \dot{\phi}_{tf}) - k_r(\phi - \phi_{tr}) \\
&\quad - b_r(\dot{\phi} - \dot{\phi}_{tr}) + \ell_w u_{af} + \ell_w u_{ar} \\
-r F_{yf} &= m_f v(r - h_{uf})(\dot{\beta} + \dot{\psi}) + m_{uf} g h_{uf} \phi_{tf} \\
&\quad - k_{tf} \phi_{tf} + k_f(\phi - \phi_{tf}) + b_f(\dot{\phi} - \dot{\phi}_{tf}) \\
&\quad + \ell_w u_{af} \\
-r F_{yr} &= m_r v(r - h_{ur})(\dot{\beta} + \dot{\psi}) - m_{ur} g h_{ur} \phi_{tr} \\
&\quad - k_{tr} \phi_{tr} + k_r(\phi - \phi_{tr}) + b_r(\dot{\phi} - \dot{\phi}_{tr}) \\
&\quad + \ell_w u_{ar}.
\end{aligned}$$

The lateral tire forces F_{yf} and F_{yr} are approximated linearly to the tire slide slip angles α_f and α_r , respectively: $F_{yf} = \mu C_f \alpha_f$ and $F_{yr} = \mu C_r \alpha_r$, where μ is the side force coefficient and C_f and C_r are tire side slip constants. At stable driving conditions, the tire side slip angles α_f and α_r can be approximated as $\alpha_f = -\beta + \delta_f - \frac{l_f \dot{\psi}}{v}$ and $\alpha_r = -\beta + \frac{l_r \dot{\psi}}{v}$.

The differential equations depend on the forward velocity v of the vehicle nonlinearly. Choosing the forward velocity as a scheduling parameter $\rho_r = v$, an LPV model is constructed. Note, that the side force coefficient is another parameter which varies nonlinearly during operational time. In (Gáspár *et al.*, 2006) a method has been proposed for the estimation of this parameter. Hence, it can be considered as a scheduling variable of the LPV model, too. In this paper, for the sake of simplicity, the variation of the side force coefficient is ignored.

The equations can be expressed in the state space representation form as:

$$\dot{x}_r = A_r(\rho_r)x_r + B_{1r_v}(\rho_r)d_r + B_{2r_v}(\rho_r)u_r, \quad (2)$$

where $x_r = [\beta \ \dot{\psi} \ \phi \ \dot{\phi} \ \phi_{tf} \ \phi_{tr}]^T$ is the state vector, $u_r = \Delta F_b$ is the control input while $d_r = \delta_f$ is considered as a disturbance.

In this approach of the rollover problem the active suspensions generate two stabilizing moments at the front and the rear, which can be considered as the effects of the suspension forces $u_{af} = (f_{fl} - f_{fr})\ell_w$ and $u_{ar} = (f_{rl} - f_{rr})\ell_w$. The control input provided by the brake system generates a yaw moment, which affects the lateral tire forces directly. The difference between the brake forces ΔF_b provided by the compensator is applied to the vehicle: $\Delta F_b = (F_{brl} + d_2 F_{bfl}) - (F_{brr} + d_1 F_{bfr})$, where d_1 and d_2 are distances, which depend on the steering angle. In the implementation of the controller means that the control action be distributed at the front and the rear wheels at either of the two sides. The reason for distributing 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 the wheels.

3. INTEGRATED CONTROL DESIGN BASED ON THE LPV METHOD

3.1 Predicting emergencies by monitoring R

Roll stability is achieved by limiting the lateral load transfers for both axles, ΔF_{zl} and ΔF_{zr} , below the level for wheel lift-off. The lateral load transfers are given by $\Delta F_{zi} = \frac{k_{ti}\phi_{ti}}{\ell_w}$, where i denotes the front and rear axles. The tire contact force is guaranteed if $\frac{mg}{2} \pm \Delta F_z > 0$ for both sides of the vehicle. This requirement leads to the definition of the normalized load transfer, which is the ratio of the lateral load transfers at the front and rear axles: $r_i = \frac{\Delta F_{zi}}{m_i g}$, where m_i is the mass of the vehicle in the front and the rear. The scheduling parameter in the LPV model is the maximum value of the normalized load transfer $R = \max(|r_i|)$.

The limit of the cornering condition is reached when the load on the inside wheels has dropped to zero and all the load has been transferred onto the outside wheels. Thus, if the normalized load transfer R takes on the value ± 1 then the inner wheels in the bend lift off. This event does not necessary result in the rolling over of the vehicle. However, the aim of the control design is to prevent the rollover in all cases and thus the lift-off of the wheels must also be prevented. Thus, the normalized load transfer is also critical when the vehicle is stable but the tendency of the

dynamics is unfavorable in terms of a rollover. An observer design method has been proposed for the estimation of the normalized load transfers, see (Gáspár *et al.*, 2005).

In this paper the detection of an imminent rollover is based on the monitoring of the normalized lateral load transfers for both axles. In the control design the actual value of the normalized load transfer is used. In order to make an estimation of the lateral load transfers the roll angles of the unsprung masses $\phi_{t,i}$ must be estimated. For this purpose a Luenberger type observer

$$\dot{\eta} = (A(\rho) + K(\rho)C)\eta + B(\rho)u - K(\rho)y \quad (3)$$

is used. The observer is based on the measured signals, a_y , $\dot{\psi}$ and $\dot{\phi}$, where a_y is the lateral acceleration.

In order to obtain a quadratically stable observer the LMI $(A(\rho) + K(\rho)C)^T P + P(A(\rho) + K(\rho)C) < 0$ must hold for suitable $K(\rho)$ and $P = P^T > 0$ for all the corner points of the parameter space, see (Apkarian and Gahinet, 1995; Wu *et al.*, 1996). By introducing the auxiliary variable $G(\rho) = PK(\rho)$, the following set of LMIs on the corner points of the parameter space must be solved:

$$A(\rho)^T P + PA(\rho) + C^T G(\rho)^T + G(\rho)C < 0.$$

3.2 Weighting strategy for the control design

Based on the model of the suspension system a control is designed considering the suspension deflections at the suspension components as measured output signals and u_s as the control inputs. The performance outputs for control design are the passenger comfort (i.e. heave displacement and acceleration z_a and z_d), the suspension deflections $z_{si} = [z_{sfl} \ z_{sfr} \ z_{srl} \ z_{srr}]$ and the tire deflection $z_{ti} = [z_{tfl} \ z_{tfr} \ z_{trl} \ z_{trr}]$.

In an earlier paper of this project the design of a global chassis system is proposed, see (Zin *et al.*, 2006). Here the suspension forces on the left and right hand sides at the front and rear are designed in the following form:

$$u_a = u - b_0(\dot{z}_s - \dot{z}_{us}), \quad (4)$$

where b_0 is a damping coefficient and u is the active force. When the value b_0 is selected small the suspension system focuses on passenger comfort, while the system focuses on road holding when value b_0 is selected large. In this paper this experience is exploited when a parameter dependent weighting strategy is applied in the design of the suspension system.

Figure 3 shows the structure of the active suspension system incorporated into the integrated control. The inputs of the controller are the measured

relative displacements and their numerical differentiations. The controller uses the normalized lateral load transfer R and the so-called normalized moment $\chi = \phi_{az} \frac{M_{act}}{M_{max}}$ as scheduling variables. Here

$$\phi_{az} = \begin{cases} 1 & \text{if } |R| < R_s \\ 1 - \frac{|R| - R_s}{R_c - R_s} & \text{if } R_s \leq |R| \leq R_c \\ 0 & \text{if } |R| > R_c \end{cases},$$

where R_s is a warning level, while R_c is a critical value of the admissible normalized lateral load transfer.

The value of the damping b_0 is scheduled by the normalized lateral load transfer R . Its value must be selected in such a way that it improves passenger comfort in normal cruising, however, it enhances road holding in an emergency. With this selection the active suspension system focuses on passenger comfort and road holding due to the value of the normalized load transfer. The LPV controller C is designed to meet the same criteria but its scheduling variable also reflects the presence of the moment demand. This is achieved by using a look-up table that encodes the function ϕ_{az} .

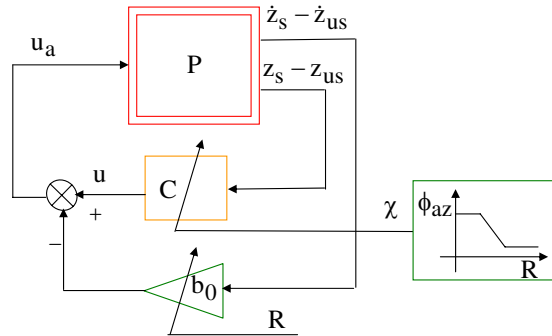


Fig. 3. Logical structure of the suspension controller

Figure 4 illustrates the effects of the tuning parameters b_0 and χ through the frequency responses of the closed loop system to the disturbances, i.e. the sprung mass acceleration, the sprung mass displacement, the displacement of the unsprung mass, and the relative displacement between the sprung and unsprung masses. This parameter represents the balance between road holding and passenger comfort. The active suspension of the closed-loop model presents better performances than the passive model. When a small value of the tuning parameter is selected a better ride comfort without the deterioration of road holding or the suspension deflection is achieved. On the other hand, when the value of the tuning parameter increases, passenger comfort deteriorates, while road holding improves. This emphasizes the trade-off between comfort and road holding and the significance of using b_0 as a varying coefficient.

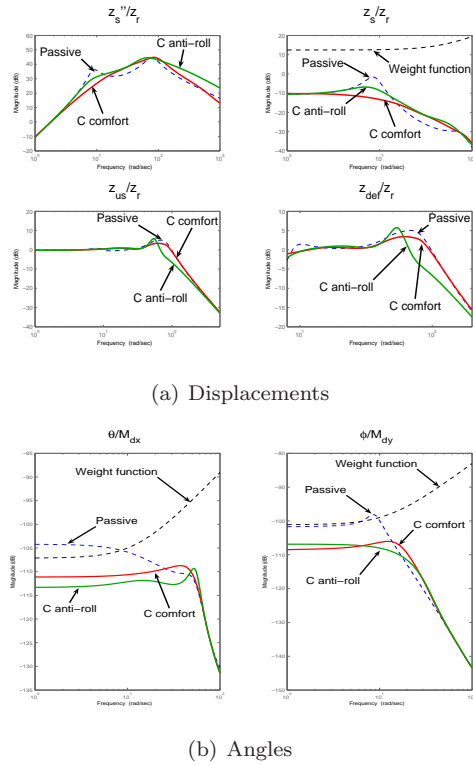


Fig. 4. Frequency responses of the suspension system

The weighting functions applied in the active suspension design are the following:

$$\left\{ \begin{array}{l} W_{z_s}(\chi) = \frac{3}{s/(2\pi f_1) + 1} \chi \\ W_{\theta}(\chi) = \frac{\chi}{s/(2\pi f_2) + 1} \\ W_{\phi}(\chi) = \frac{1 - \chi}{s/(2\pi f_3) + 1} \\ W_u = 10^{-2} \\ W_{z_r} = 7 \cdot 10^{-2} \\ W_{dx} = 10^5 \\ W_{dy} = 5 \cdot 10^4 \\ W_n = 10^{-3} \end{array} \right.$$

where W_{z_s} is shaped in order to reduce bounce amplification of the suspended mass (z_s) between $[0, 8]Hz$ ($f_1 = 8Hz$), W_{θ} attenuate amplification in low frequency and the frequency peak at $9Hz$ ($f_2 = 2Hz$) and W_{ϕ} reduces the rolling moment especially in low frequency ($f_3 = 2Hz$). Then W_{z_r} , W_{dx} , W_{dy} and W_n model ground, roll, pitch disturbances (z_r , M_{dx} and M_{dy}) and measurement noise (n) respectively, and W_u is used to limit the control signal. Note, that although the suspension model is a linear time invariant (LTI), the model of the augmented plant is LPV because of the weighting strategy. Thus, the control design is performed in an LPV setting.

The control of braking forces are designed in terms of the rollover problem. The measured outputs

are the lateral acceleration of the sprung mass, the yaw rate and the roll rate of the sprung mass while u_r are the control inputs. The performance outputs for the control design are the lateral acceleration a_y , the lateral load transfers at the front and the rear ΔF_{zf} and ΔF_{zr} . The lateral acceleration is given by $a_y = v\dot{\beta} + v\dot{\Psi} - h\ddot{\Phi}$.

The weighting function for the lateral acceleration is selected in such a way that in the low frequency domain the lateral accelerations of the body must be penalized by a factor of ϕ_{ay} .

$$W_{p,ay} = \phi_{ay} \frac{\frac{s}{2000} + 1}{\frac{s}{12} + 1},$$

$$\text{where } \phi_{ay} = \begin{cases} 0 & \text{if } |R| < R_s \\ \frac{|R| - R_s}{R_c - R_s} & \text{if } R_s \leq |R| \leq R_c \\ 1 & \text{if } |R| > R_c \end{cases},$$

R_c defines the critical status when the vehicle is in an emergency and the braking system must be activated. The gain ϕ_{ay} in the weighting functions is selected as a function of parameter $|R|$ in the following way. In the lower range of $|R|$ the gain ϕ_{ay} must be small, and in the upper range of $|R|$ the gains must be large. Consequently, the weighting functions must be selected in such a way that they minimize the lateral load transfers in emergencies. In normal cruising the brake is not activated since the weight is small. The weighting function for the lateral loads and the braking forces are the following:

$$W_{p,Fz} = \text{diag}\left(\frac{1}{7}, \frac{1}{5}\right)$$

$$W_{p,\Delta Fb} = 10^{-3} \phi_{ay}$$

The control design is performed based on an augmented LPV model of the yaw-roll dynamics where two parameters are selected as scheduling variables: the forward velocity and the maximum value of the normalized lateral load transfer either at the rear side or at the front $\rho_r = [v \ R]^T$.

In the design of rollover problem the difference in the braking forces is designed. Based on this fictitious control input the actual control forces at the front and rear on both sides generated in the braking system are calculated. Certainly, different optimization procedures, which distribute the fictitious force between the braking forces can be implemented. However, this problem is not within the scope of the paper.

The solution of an LPV problem is governed by the set of infinite dimensional LMIs being satisfied for all $\rho \in \mathcal{F}_P$, thus it is a convex problem. In practice, this problem is set up by gridding the parameter space and solving the set of LMIs that hold on the subset of \mathcal{F}_P . If this problem does not have a solution, neither does the original infinite dimension problem. Even if a solution is found, it does not guarantee that the solution satisfies

the original constraints for all ρ . However, it is expected since the matrix functions are continuous with respect to ρ . The number of grid points depends on the nonlinearity and the operation range of the system. For the interconnection structure, \mathcal{H}_∞ controllers are synthesized for 7 values of velocity in a range $v = [20\text{km/h}, 140\text{km/h}]$. The normalized lateral load transfer parameter space is selected $R = [0, R_s, R_c, 1]$.

4. A SIMULATION EXAMPLE

In the simulation example, a double lane change maneuver is performed. In this maneuver passenger comfort and road holding are guaranteed by the suspension actuators and the rollover is prevented by modifying the operation of the suspension actuators and using an active brake. When a rollover is imminent the values R increase and reach a lower critical limit (R_s) and suspension forces are generated to create a moment at the front and the rear. When this dangerous situation persists and R reaches the second critical limit (R_c) the active brake system generates unilateral brake forces.

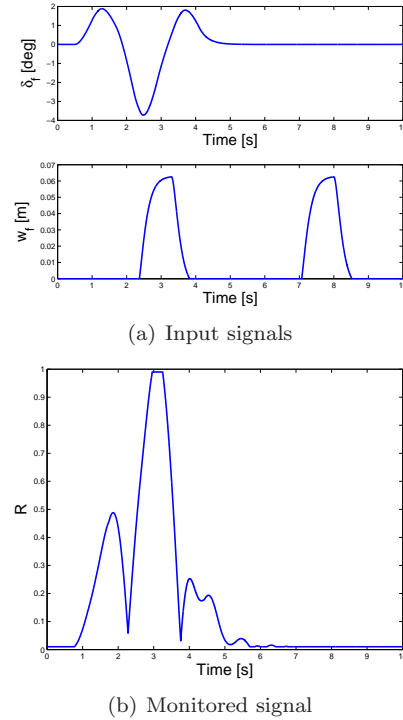


Fig. 5. Time responses in the double lane change maneuver

The velocity of the vehicle is 90 km/h . The maneuver starts at the 1^{st} second and at the 2.5^{th} and the 7^{th} seconds 6-cm -high bumps on the front wheels disturbs the motion of the vehicle. The steering angle is generated with a ramp signal with 3.5 degrees maximum value and 4 rad/s filtering, which represents the finite bandwidth of the

driver. The time responses of the steering angle, the road disturbance, the yaw rate, the roll rate, the lateral acceleration, the heave acceleration on the front-left side, the normalized load transfer at the rear and their maximum, the vehicle velocity, the roll moments at the front and the rear and the braking forces at the front and the rear are presented in Figure 5... Figure 7.

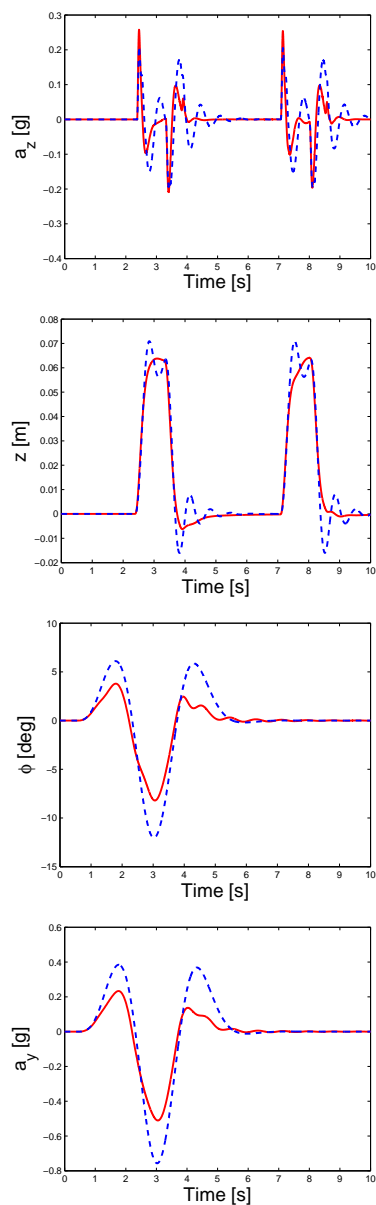


Fig. 6. Output signals in the double lane change maneuver

The effect of a 6-cm-high bump disturbs heave acceleration at the 2.5th second. The effect of this disturbance should be reduced by the suspension system, since it improves the passenger comfort and road holding. During the maneuver the lateral acceleration and the roll angles of the unsprung masses increase, thus the normalized load transfer also increases and reaches the critical value R_s . Control forces (0.5 kN and 0.5 kN at the front and at the rear, respectively) should also be generated

by the suspension forces so that the controller can prevent the rollover of the vehicle. Thus, during the maneuver the suspension system focuses on both the passenger comfort and the roll stability of the vehicle. The control moments are not sufficient to prevent rollovers, since the normalized lateral load transfers have achieved the critical value R_c . Thus the brake is also activated and unilateral braking forces (approximately 0.9 kN and 1 kN on the left and the right hand sides in the rear) are generated. As a result the velocity of the vehicle decreases and the normalized lateral load transfers stay below the critical value 1. After the double lane maneuver another 6-cm-high bump disturbs the motion. In this case a large suspension force generated by the suspension actuators is needed to reduce both the magnitude and the duration of the oscillation.

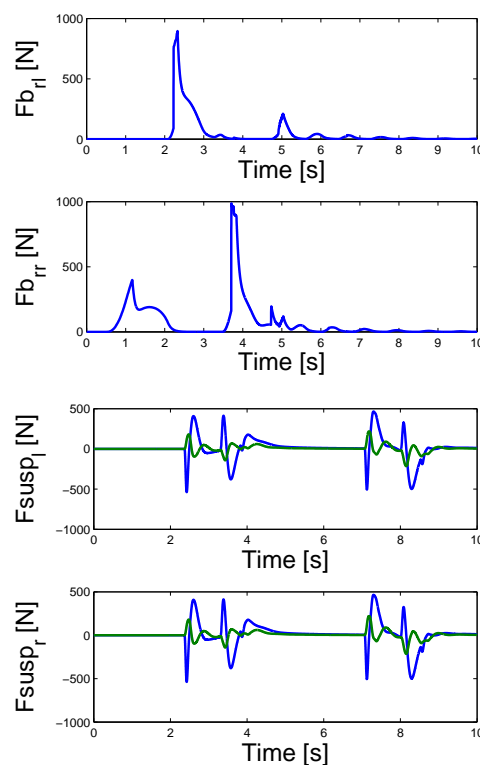


Fig. 7. Control signals in the double lane change maneuver

In the future it is possible to exploit the balance between the brake and suspension systems to enhance braking. During braking the real path might be significantly different from the desired path due to the brake moment which affects the yaw motion. Thus, the braking maneuver usually requires the drivers intervention. Applying the integrated control, the suspension system is able to focus on the emergency, consequently safety is improved.

5. CONCLUSION

In this paper an integrated control structure that uses active suspensions and an active brake is proposed to improve the safety of vehicles. In normal operation the suspension system focuses on passenger comfort and road holding, however, in an emergency the safety requirement overwrites the passenger comfort demand. When the emergency persists, the brake is also activated to reduce the rollover risk. The solution is based on a weighting strategy in which the normalized lateral load transfer is selected as a scheduling variable. The design is based on an \mathcal{H}_∞ control synthesis extended to LPV systems that uses a parameter dependent Lyapunov function. This control mechanism guarantees the balance between rollover prevention and passenger comfort. The applicability of the method is demonstrated through a complex simulation example containing vehicle maneuvers.

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