Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

AN INTEGRATED VEHICLE CONTROL WITH ACTUATOR RECONFIGURATION ⋆

P. Gáspár Z. Szabó J. Bokor

Computer and Automation Research Institute, Hungarian Academy of Sciences, Hungary (Tel: 3612796171; e-mail: gaspar@sztaki.hu).

Abstract: The aim of this paper is to present an algorithm that, by an actuator reconfiguration, performs tracking and rollover prevention at the same time. Using an active steering control a path following task can be realized. However during operational time maneuvers might occur when overturning moments are generated. By the brake mechanism rollover prevention can be ensured but the real path will significantly deviate from the desired one and this effect on the yaw motion has to be compensated using active steering. The integrated control of the steering and braking actuators that realizes the balance between the tracking task and rollover prevention is designed based on a Linear Parameter Varying (LPV) model. The conflict between these performance demands can be avoided by a suitable reconfiguration of the active suspension system by generating roll moments to reduce the rollover risk without affecting the yaw dynamics. Thus the critical level when braking must be applied to prevent rollover situation is increased by an active suspension mechanism.

Keywords: linear parameter varying control, chassis control, reconfiguration, robust control

1. INTRODUCTION

Recently, there has been a growing demand for vehicles with ever better driving characteristics in which efficiency, safety, and performance are ensured. In meeting these demands the research and development of vehicle navigation systems and path tracking systems play an important role. The tracking problem is solved by using active steering. When the vehicle is traveling along the road there are maneuvers, e.g. a double lane change or a cornering, during which overturning moments are generated. The role of the brake mechanism is to reduce the lateral tire forces and decelerate the vehicle. Using the brake the real path is significantly deviates from the desired path due to the brake moment which affects the yaw motion. This deviation must be compensated by the active steering system. To perform tracking and rollover prevention at the same time poses a difficult problem since these tasks are in contradiction with each other.

There are many papers concerned with different approaches that develop steering systems, see e.g. Kim et al. [2002], Setlur et al. [2002]. Moreover, different control structures are combined in one control mechanism in order to create fault-tolerant systems and enhance safety. In Ackermann et al. the linear steering control is extended by nonlinear emergency steering and braking control, see Odenthal et al. [1999]. In Nagai et al. a control system is proposed by the front steering angle and the distribution of braking forces, Nagai et al. [2002].

In this paper a integrated control mechanism is applied. When the vehicle is cruising it only performs the tracking. In an emergency when a rollover is imminent, it also performs the prevention of rollovers. The combined yaw-roll model, which is the basis of the control design, is nonlinear with respect to the forward velocity of the vehicle. The control design is based on the LPV model, which is adjusted continuously by the forward velocity of the vehicle in real-time. The normalized lateral load transfer at the rear is also applied as another scheduling parameter in order to focus on performance specifications. The model is augmented with the signals defined by the performance specifications and the uncertainty structure defined by the difference between the plant and its model. The active brake switches on in an emergency and it switches off when the emergency is over. Using such switching structures a chattering phenomenon may occur, and it may degrade the performance properties of the vehicle. In the combined control structure a solution is also proposed to solve the chattering problem. To compensate the deviation of the yaw motion caused by braking a slight modification of the tracking command for the steering subsystem is needed to avoid under or over–steering.

Active suspensions are used to provide good handling characteristics and to improve ride comfort while harmful vibrations caused by road irregularities and on-board

⋆ This work was supported by the Hungarian National Office for Research and Technology through the project "Advanced Vehicles and Vehicle Control Knowledge Center" (OMFB-01418/2004) which is gratefully acknowledged. This research work has been partially supported by Control Engineering Research Group, Hungarian Academy of Sciences at the Budapest University of Technology and Economics. The partial support of the grant no. $T-048482$ of the Hungarian Research Fund is also acknowledged. Dr Gáspár and Dr Szabó were supported by the János Bolyai Research Scholarship of the Hungarian Academy of Sciences.

excitation sources act upon the vehicle, Alleyne and Hedrick [1995], Hrovat [1997]. However, emergency situations, when the application of the active braking system is needed for rollover prevention, can also be delayed, moreover the necessary brake moments can be reduced, by using a suitable designed suspension system, Gáspár and Bokor [2006].

The proposed method is based on the fact that when active suspension systems are used not only the effects of road irregularities can be eliminated but road holding can also be improved by generating roll moments. The design of active suspension in this problem significantly differs from that of the conventional active suspension design where the performance specifications for passenger comfort, suspension deflections and tire deflections are met simultaneously. However, in this case the controller is no longer able to focus only on one of the performance specifications and to ignore other performances. When the vehicle is cruising, the performances are the same as in the conventional system. When the vehicle is coming close to rolling over, the performance demands significantly differ from those in the conventional case and in emergency a stabilizing moment is generated to balance the overturning moment in such a way that the control torque leans the vehicle into the bend.

The structure of the paper is as follows. In Section 2 the LPV structure of the vertical, yaw and roll model is constructed. In Section 3 the performance specifications and the uncertainty structures are formalized in an LPV design framework. In Section 4 the integrated control mechanism is demonstrated. Finally, Section 5 contains some concluding remarks.

2. THE LPV MODEL OF THE VERTICAL, YAW AND ROLL DYNAMICS

Figure 1 illustrates the dynamics of the vehicle, which is modeled by a three-body system. Here m_s is the sprung mass, m_{uf} is the unsprung mass at the front including the front wheels and axle, and m_{ur} is the unsprung mass at the rear with the rear wheels and axle. All suspensions consist of a spring, a damper and an actuator to generate a pushing force between the body and the axle. In the following the motion differential equations of the dynamics of single unit vehicle are formalized. The dynamic equation of the first system considers forces for the lateral dynamics and the torque balance equations for yaw and roll moments. The second system describes the equations containing the forces and moments for the vertical dynamics.

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 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 Becker and Packard [1994], Leith and Leithead [2000], Rough and Shamma [2000], Wu [2001]. 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

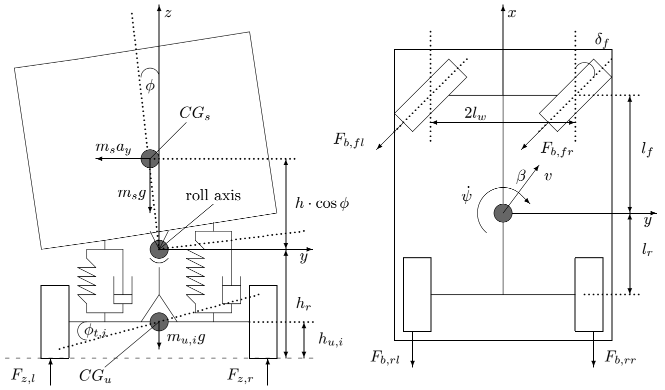

Fig. 1. Vehicle model for control design

matrices A and B are generally nonlinear functions of the scheduling vector ρ.

The adhesion coefficients in the lateral tire forces depend on the various types of road surface. These values can be approximated by using Burckhardt's method. In this paper, the nonlinear effects of the forward velocity and that of the adhesion coefficient are taken into consideration in the yaw and roll dynamics. Suspension systems also have nonlinear damping characteristics. For example when moving upwards the wheel generates a smaller damping force than when moving downwards. These nonlinear effects are considered by selecting the square of the relative displacement and the signum of the relative velocity as scheduling variables in the corresponding LPV model.

The equations of the yaw and roll dynamics are expressed in the state space representation, where 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 $\phi_{t,r}$. Let the state vector be the following:

$$x_r = \begin{bmatrix}\beta \ \dot{\psi} \ \phi \ \dot{\phi} \ \phi_{t,f} \ \phi_{t,r}\end{bmatrix}^T \tag{1}$$

Using the state vector, the differential algebraic model is:

$$\dot{x}_r = A(\rho_r)x_r + B(\rho_r)u_r. \tag{2}$$

The components of the control inputs u_r are the front wheel steering angle δ_f, and the difference in brake forces between the left and right-hand sides of the vehicle ΔF_b. For the details of the dynamic equations see Gáspár et al. [2003b]. It is assumed that the difference in brake forces ΔF_b provided by the compensator is applied to 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 between the front and the rear wheels and the two sides. The reason for distributing the control force between the front and rear wheels is to optimize the wear of the tires. In this case a sharing logic is required which calculates the brake forces for the wheels.

The scheduling vector ρ_r is selected with four scheduling variables $\rho_r = [\rho_1 \ \rho_2 \ \rho_3 \ \rho_4]$ with $\rho_1 = \mu$, $\rho_2 = \frac{\mu}{v}$, $\rho_3 = \frac{\mu}{v^2}$, $\rho_4 = \frac{1}{v}$. In the LPV model both the forward velocity and the adhesion coefficient are varying. It is assumed that the forward velocity and the adhesion coefficient are measured or available. Several papers have proposed estimation method for the vehicle velocity, see e.g Song et al. [2002]. A grey-box identification method based on an observer design was proposed in Gáspár et al. [2005b].

Next, the suspension dynamics is also formalized. The state vector x_s uses in the suspension system is selected as follows:

$$x_s = [q \; x_u \; \dot{q} \; \dot{x}_u]^T, \tag{3}$$

with $q = [x_1 \; \theta \; \phi]^T$ and $x_u = [x_{2fl} \; x_{2fr} \; x_{2rl} \; x_{2rr}]^T$. Here x_1, θ and ϕ are the vertical displacement at the center of gravity, the pitch angle and the roll angle of the sprung mass, respectively. The front and rear displacement of the unsprung mass on the left and right side be denoted by x_{2fl}, x_{2rl}, x_{2fr}, and x_{2rr}.

Two expressions concerning the front and rear displacement of the unsprung mass on the left and right side and their velocities are selected as the components of the scheduling vector:

$$\rho_{bij} = sgn(\dot{x}_{2ij} - \dot{x}_{1ij}), \tag{4}$$

$$\rho_{kij} = (x_{2ij} - x_{1ij})^2. \tag{5}$$

with $ij \in \{fl, fr, rl, rr\}$. Parameter ρ_{bij} depends on the relative velocity, parameter ρ_{kij} is equal to the relative displacement. In practice, the relative displacement is a measured signal. The relative velocity is then determined by numerical differentiation from the measured relative displacement. Thus, in the LPV model of the active suspension system eight scheduling variables are selected.

The state space representation of the LPV model is as follows:

$$\dot{x}_s = A(\rho_s)x_s + B(\rho_s)u_s, \tag{6}$$

where the vector of the actuator forces is
$u_s = [f_{fl} \; f_{fr} \; f_{rl} \; f_{rr}]^T$. For the details of the dynamic equations see e.g. Gáspár et al. [2003a]. Note that for the sake of simplicity in this paper the dynamics of the actuators are ignored.

3. THE CONSTRUCTION OF THE LPV MODEL FOR CONTROL DESIGN

The objective of the control design is to minimize the tracking error and prevent rollovers. The chassis control integrates the active suspension system with the active steering and the active brake. The mechanism of the control system is the following. When the vehicle is in a normal cruising mode the active steering minimizes the error between the predefined and the actual paths of the vehicle while the suspension system guarantees passenger comfort and road holding. In normal cruising the brake is not activated. When the vehicle is in an imminent rollover, the suspension system generates a stabilizing moment to balance the overturning moment in such a way that the control torque leans the vehicle into the corner. If emergency persists the brake system is also activated to reduce the rollover risk.

Roll stability is achieved by limiting the lateral load transfers on both axles, $\Delta F_{z,l}$ and $\Delta F_{z,r}$, to below the levels for wheel lift-off. The lateral load transfer is calculated:

$$\Delta F_{z,i} = \frac{k_{t,i}\phi_{t,i}}{l_w}, \tag{7}$$

where $k_{t,i}$ is the stiffness of the 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, and $i = f, r$ denotes the front and rear of the vehicle.

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:

$$R_i = \frac{\Delta F_{z,i}}{m_i g}. \tag{8}$$

where m_i is the mass of the vehicle in the front and the rear. 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. The limit cornering condition occurs when the load on the inside wheels has dropped to zero and all the load has been transferred onto the outside wheels.
Let $R = max\{R_f, R_r\}$.

The roll angles of the unsprung masses have an important role in the monitoring of rollovers, since the calculation of the normalized load transfers is based on these signals. In practice the roll rates of the unsprung masses are measured and the roll angles are calculated by using a numerical integration. A method was proposed for the estimation of the roll angles of the unsprung masses based on an observer design, see Gáspár et al. [2005a].

In the next subsections the control design both for the combined tracking and the roll stability and for the reconfigurable suspension system are discussed.

3.1 Control design for the steering and braking system

The closed-loop system applied for the combined tracking and the roll stability tasks includes the feedback structure of the model $G(\rho)$, the compensator, and elements associated with the uncertainty models and performance objectives, see Figure 2. The command signal is a pre-defined yaw rate signal. The performance and the measured signals are:

$$z_r = \begin{bmatrix} e_{\dot{\psi}} \; a_y \; u_r \end{bmatrix}^T, \tag{9}$$

$$y_r = \begin{bmatrix} a_y \; \dot{\psi} \; \dot{\psi}_{\text{cmd}} \end{bmatrix}^T, \tag{10}$$

where $e_{\dot{\psi}}, a_y, \dot{\psi}$ are the tracking error, the lateral acceleration and the yaw rate, respectively. n_a and $n_{\dot{\psi}}$ are measurement noises. In order to solve the yaw rate tracking problem, the command signal must be fed forward to the controller. The tracking error is the difference between the actual yaw rate and the yaw rate command. Hence, the controller also uses the yaw rate command signal $\dot{\psi}_{\text{cmd}}$.

The input scaling weight W_{cmd} normalizes the yaw rate command to the maximum expected command. The yaw rate command is selected 15 deg/sec. The dynamics of the reference input is as follows:

$$T = \frac{\omega^2}{s^2 + 2\zeta\omega s + \omega^2} \tag{11}$$

with $\omega = 12$ and $\zeta = 1$. The dynamics of the yaw rate command is defined by the designer by using parameters ω and ζ.

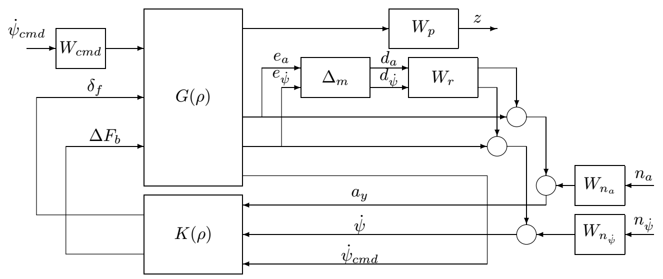

Fig. 2. The closed-loop interconnection structure

The weighting function W_p represents the performance outputs. The weighting functions of the tracking error and of the lateral acceleration are selected as:

$$W_{p_e} = 100\frac{(T_{b1}s+1)}{(T_{b2}s+1)} \tag{12}$$

$$W_{p_a} = \phi_a\frac{(T_{b3}s+1)}{(T_{b4}s+1)}. \tag{13}$$

where T_{bi} are time constants. Here, it is required that in the steady state value of the tracking error should be below 1% and the lateral accelerations of the body should be rejected by a factor of ϕ_a. The reason for keeping the control signals small is to avoid actuator saturation. Thus, the weights of control inputs are: W_u for the steering angle is 1/20, and W_{Fb} for the brake force is 1/20.

ϕ_a is a gain, which reflects the relative importance of the lateral acceleration in the LPV control design and it is chosen to be parameter-dependent, i.e., the function of R. When R is small, i.e., when the vehicle is not in an emergency, ϕ_a is small, indicating that the LPV control should not focus on minimizing acceleration, it should only guarantee the yaw rate tracking by setting the steering angle. On the other hand, when R approaches the critical value, ϕ_a is large, indicating that the control should focus on preventing the rollover. As the gain ϕ_a increases the lateral acceleration decreases, since the active brake affects the lateral acceleration directly.

In the control design the parameter dependence of the gain is characterized by the constants R_b and $R_{b,2}$. The parameter dependent gain ϕ_a is as follows:

$$\phi_a(R) = \begin{cases} 0 & \text{if } \quad |R| < R_b \\ \dfrac{1}{R_{b,2}-R_b}(|R|-R_b) & \text{if } \quad R_b \le |R| \le R_{b,2} \\ 1 & \text{otherwise} \end{cases} \tag{14}$$

R_b 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_b is to 1 the later the control will be activated. Parameter $R_{b,2}$ shows how fast the control should focus on minimizing the lateral acceleration. Hence the signal R should be included in the set of scheduling variables.

The uncertainties of the model, which is represented by unmodeled dynamics, are represented by W_r and Δ_m. Design models used for tracking and roll stability control typically exhibit high fidelity at lower frequencies, but they degrade rapidly at higher frequencies due to poorly modeled or neglected effects. Thus, the weighting of the uncertainty is selected as $W_r = 0.1\frac{s/2+1}{s/40+1}$. W_n is selected as a diagonal matrix, which accounts for sensor noise models in the control design: $W_n = diag[W_{na}, W_{nu}]$. The noise weights W_{na} and W_{nu} are chosen 0.01 m/s^2 for the lateral acceleration and 0.01 deg/sec for the yaw rate.

The closed-loop system $M(\varrho)$ is given by a lower linear fractional transformation (LFT) structure:

$$M(\varrho) = \mathcal{F}_\ell(P(\varrho), K(\varrho)) \tag{15}$$

where $P(\varrho)$ is the augmented plant. The goal of the control design is to minimize the induced $\mathcal{L}_2$ norm of a LPV system $M(\varrho)$, with zero initial conditions, which is given by

$$\|M(\varrho)\|_\infty = \sup_{\varrho\in\mathcal{F}_\mathcal{P}} \sup_{\|w\|_2\neq 0, w\in\mathcal{L}_2} \frac{\|z_r\|_2}{\|w\|_2} \tag{16}$$

3.2 Control design for the reconfigurable suspension system

In the suspension system the goals are to keep the heave accelerations, suspension deflections, wheel travels, and control inputs small over the desired operation range. The performance signals in the suspension design are:

$$z_s = [a_z \ s_d \ t_d \ u_s]^T \tag{17}$$

Here $a_z = \ddot{q}, s_d = x_{1ij} - x_{2ij}, t_d = x_{2ij} - w_{ij}, u_s$ with $ij \in \{fl, fr, rl, rr\}$ are the acceleration of the sprung mass, the suspension deflection, the tire deflection and the control forces at the front and rear on both sides (left and right), respectively. Disturbance w_{ij} is caused by road irregularities. The measured signals are the relative displacements between the sprung mass and the unsprung mass at the front and rear on both sides.

The performance weighting functions can be considered as penalty functions, i.e. weights should be large in a frequency range where small signals are desired and small where larger performance outputs can be tolerated. Thus, $W_{p,az}$ and $W_{p,sd}$ are selected as

$$W_{p,az}(\rho_{kij}) = \phi_{az}(\rho_{kij}) \cdot \frac{T_{s1}+1}{T_{s2}+1}, \tag{18}$$

$$W_{p,sd}(\rho_{kij}) = \phi_{sd}(\rho_{kij}) \cdot \frac{T_{s3}+1}{T_{s4}+1}. \tag{19}$$

where T_{si} are time constants. Here, it is assumed that in the low frequency domain disturbances at the heave accelerations of the body should be rejected by a factor of ϕ_{az} and at the suspension deflection by a factor of ϕ_{sd}.

The trade-off between passengers comfort and suspension deflection is due to the fact that is not possible to keep them together simultaneously. A large gain ϕ_{az} and a small gain ϕ_{sd} correspond to a design that emphasizes passenger comfort while choosing ϕ_{az} small and ϕ_{sd} large corresponds to a design that focuses on suspension deflection.

The idea of the reconfigurable suspension system is based on the fact that active suspension systems are used not only to eliminate the effects of road irregularities but also to generate roll moments to improve road holding. For reconfigurable suspension system the gains are selected as functions of the suspension deflection and the normalized lateral load transfer. In normal cruising, i.e. when $R < R_s$ the parameter dependence of the gains is characterized by the constants ρ_1 and ρ_2 in the following way:

$$\phi_{az}(\rho_{kij}) = \begin{cases} 1 & \text{if} \quad |\rho_{kij}| < \rho_1 \\ \dfrac{|\rho_{kij}| - \rho_2}{\rho_1 - \rho_2} & \text{if} \quad \rho_1 \leq |\rho_{kij}| \leq \rho_2 \\ 0 & \text{otherwise} \end{cases}, \tag{20}$$

$$\phi_{sd}(\rho_{kij}) = \begin{cases} 0 & \text{if} \quad |\rho_k| < \rho_1 \\ \dfrac{|\rho_{kij}| - \rho_1}{\rho_2 - \rho_1} & \text{if} \quad \rho_1 \leq |\rho_{kij}| \leq \rho_2 \\ 1 & \text{otherwise} \end{cases}, \tag{21}$$

while in emergency, i.e. when $R \geq R_s$, the suspension system should be reducing the rollover risk and guaranteeing passenger comfort is no longer a priority:

$$\phi_{az}(\rho_{kij}) = 0 \tag{22}$$

$$\phi_{sd}(\rho_{kij}) = 1. \tag{23}$$

Therefore the set of scheduling variables should be augmented with the signal R.

The uncertainties of the model are represented by W_r and Δ_m. W_r is assumed to be known, and Δ_m is assumed to be unknown with $\|\Delta_m\|_\infty < 1$. Design models used for active suspension control typically exhibit high fidelity at lower frequencies, but they degrade at higher frequencies. Thus, W_r is selected as $W_r = 2.25\frac{S+20}{S+450}$.

The LPV problem setting is analogous with the one presented for the control of yaw and roll dynamics.

4. SIMULATION EXAMPLES

In this section the operation of the integrated control is illustrated in a cornering maneuver in which the vehicle is traveling at 70 km/h. The tracking task is to follow a predefined yaw rate. In the first experiment only the active brake is used to reduce the rollover risk (solid line) while in the second simulation a reconfigurable suspension control is also applied (dashed line).

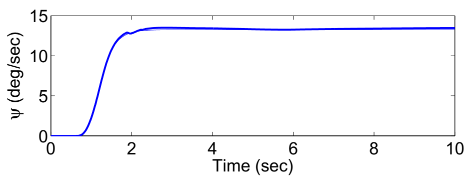

Fig. 3. Reference signal

Figure 3 shows the time responses of the rollover prevention system to the cornering. The yaw rate command applied in the simulation is a step signal. In order to avoid the unrealistic change in the yaw rate command, a ramp signal is applied when the signal has reached the maximum value (15 deg/s) in 0.5 s and filtered at 4 rad/s to represent the finite bandwidth of the driver.

In the simulation example it is assumed that the difference in the brake forces ΔF_b provided by the compensator is applied to the rear axle. This means that only one wheel is decelerated at the rear axle. This deceleration generates an appropriate yaw moment. The relative roll angle does not exceed the acceptable limit, which is about $6-8$ degrees. Besides rollover prevention, the controllers also guarantees the tracking performance of yaw rate command.

Figure 4 shows the measured values of the lateral acceleration and the values of the scheduling variables v and R. For the sake of simplicity the value of the adhesion constant μ was maintained constant during the simulation.

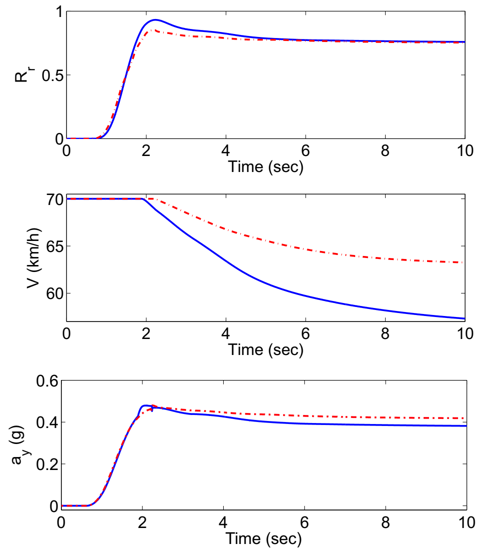

Fig. 4. Measured signals

Figure 5 shows that the tracking error is below an acceptable limit in the yaw rate channel in both cases.

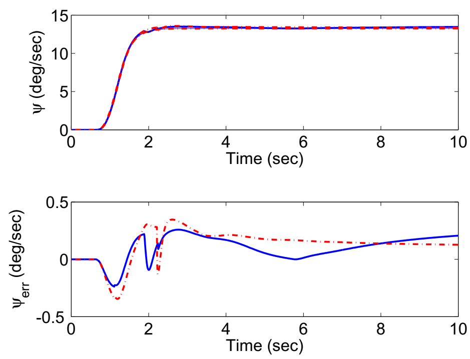

Fig. 5. Tracking performance

During the cornering maneuver the lateral acceleration increases and in the same time the lateral load transfers also increase. Thus at the 2nd second the normalized load transfer R reaches its critical value, i.e. an imminent rollover occurs, and the brake is activated. The values of the actual braking forces are depicted in the second plot of the Figure 6. The braking action modifies the yaw rate dynamics therefore to fulfill the tracking requirement the steering command must be modified.

It can be observed that for the controller in which the suspension system is also actuated the necessary braking forces are considerably smaller (dashed line) than in the case when only the brakes are actuated (solid line). Thus the induced compensation in the steering command is significantly smaller – around 7% – than for the first case – around 15%, see the upper plot in Figure 6.

The suspension forces are depicted in the bottom plots of Figure 6.

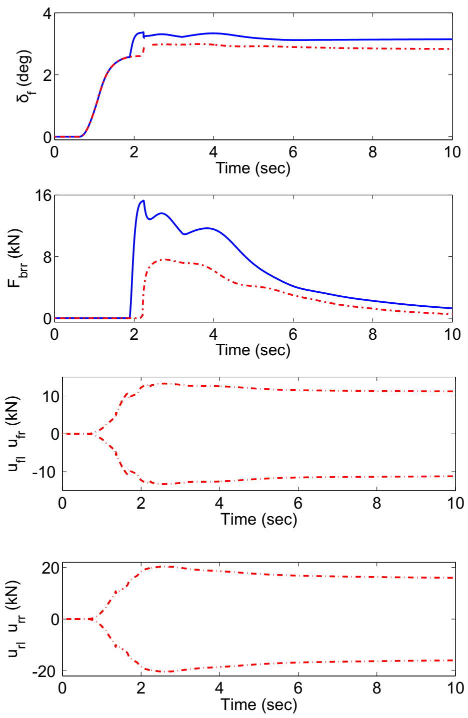

Fig. 6. Control signals

5. CONCLUSIONS

In the paper a integrated control structure has been proposed for tracking the path of the vehicle and preventing rollovers. In cruising mode the controller minimizes the tracking error and when the normalized load transfer has reached its critical value the brake control is also activated in order to prevent the rollover. In order to reduce the steering action necessary to compensate the side effect of braking on the yaw rate dynamics a reconfigurable control of the suspension system is applied. When the vehicle is cruising, the performances of the suspension system are the same as in the conventional suspension controls while in emergency a stabilizing moment is generated to balance the overturning moment in such a way that the control torque leans the vehicle into the bend.

The modeling and the control design are based on the LPV method. In the LPV model the forward velocity, the adhesion coefficient and the normalized lateral load transfer are chosen as scheduling variables. The LPV controller is able to handle the nonlinear model, as well as the performance demands and the model uncertainties.

REFERENCES

A. Alleyne and J.K. Hedrick. Nonlinear adaptive control of active suspensions. *IEEE Transactions on Control Systems Technology*, pages 94–101, 1995.

G. Becker and A. Packard. Robust performance of linear parametrically varying systems using parametrically-dependent linear feedback. *System Control Letters*, 23: 205–215, 1994.

P. Gáspár and J. Bokor. Fault-tolerant rollover prevention system based on an LPV method. *International Journal of Vehicle Design*, 42(3-4):392–412, 2006.

P. Gáspár, Z. Szabó, and J. Bokor. Brake control combined with prediction to prevent the rollover of heavy vehicles. *Proc. of the IFAC World Congress, Praha*, 2005a.

P. Gáspár, Z. Szabó, and J. Bokor. Estimation of the friction coefficient for road vehicles. *Report, submitted, American Control Conference*, 2005b.

P. Gáspár, I. Szászi, and J. Bokor. Active suspension design using the mixed μ synthesis. *Vehicle System Dynamics*, 40(4):193–228, 2003a.

P. Gáspár, I. Szászi, and J. Bokor. The design of a combined control structure to prevent the rollover of heavy vehicles. *Proc. of the European Journal of Control*, 10 (2):1–15, 2003b.

D. Hrovat. Survey of advanced suspension developments and related optimal control applications. *Automatica*, 33:1781–1817, 1997.

H.J. Kim, H.S. Yang, and Y.P. Park. Improving the vehicle performance with active suspension using road-sensing algorithm. *Computers and Structures*, 80:1569–1577, 2002.

D.J. Leith and W.E. Leithead. Survey of gain-scheduling analysis and design. *International Journal of Control*, 73:1001–1025, 2000.

M. Nagai, M. Shino, and F. Gao. Study on integrated control of active front steer angle and direct yaw moment. *JSAE Review*, 23:309–315, 2002.

D. Odenthal, T. Bünte, and J. Ackermann. Nonlinear steering and bring control for vehicle rollover avoidance. *Proc. European Control Conference*, 1999.

W.J. Rough and J.S. Shamma. Research on gain scheduling. *Automatica*, 36:1401–1425, 2000.

P. Setlur, D.Dawson, J. Wagner, and Y. Fang. Nonlinear tracking controller design for steer-by-wire automotive systems. *Proc. of the American Control Conference*, pages 280–285, 2002.

C.K. Song, M. Uchanski, and J.K. Hedrick. Vehicle speed estimation using accelerometer and wheel speed measurements. *Proc. of the SAE Automotive Transportation Technology, Paris*, 2002.

F. Wu. A generalized LPV system analysis and control synthesis framework. *International Journal of Control*, 74:745–759, 2001.