Rollover Avoidance for Steer-by-Wire Vehicles by Using Linear Parameter Varying Methods

Peter Gaspar, Istvan Szaszi, Jozsef Bokor

Abstract— **In the paper a combined control structure is proposed for tracking the path of the vehicle. Besides tracking this structure also guarantees rollover avoidance. First, a Linear Parameter Varying (LPV) model is constructed, in which both the performance specifications and the model uncertainties are taken into consideration. Then a parameter dependent LPV controller based on the H_∞ design method is designed. Since a chattering problem may occur when using this combined control, one possible extension of the control structure is also proposed. The control mechanism is demonstrated in various maneuver situations.**

Keywords— **vehicle dynamics, active control, robustness, linear parameter varying control, nonlinear control systems.**

I. Introduction

These days there is a growing demand for vehicles with ever better driving characteristics in which efficiency, safety, and performance are ensured. In order to meet these demands the research and development of vehicle navigation systems and path tracking systems play an important role. The basis of this research is the drive-by-wire systems, which replace the traditional mechanical lineages with electrical equivalents in the next generation of vehicles. This upgrading of components includes steer-by-wire and brake-by-wire systems. This will result in a variety of architectures for enhancing steering characteristics and automatic tracking. There are many papers concerning different approaches that develop steering and braking systems, see e.g. [1], [8], [14].

In our paper a method for the tracking problem is proposed. Besides tracking this method also guarantees the rollover avoidance. When the vehicle is travelling on the road there are maneuvers, e.g. a double lane change or a cornering, which may result in an emergency situation. Besides active steering this method applies an active brake. In a normal driving situation the brake should not be activated. The brake is only activated when the vehicle comes close to a rollover.

To perform tracking and rollover prevention at the same time poses a difficult problem since these tasks are in contradiction with each other. The tracking problem is solved by using active steering and in this operation the objective is to minimize the tracking error. When the vehicle body rolls out of the corner and the center of mass shifts outboard of the centerline then a destabilizing moment is created and the vehicle is in an emergency situation. Several schemes concerned with the possible active intervention into the vehicle dynamics have been proposed: active anti-roll bars, active steering and an active braking, e.g. [2], [7], [11], [15]. Moreover, different control structures are combined in one control mechanism in order to create fault-tolerant systems and enhance safety. In Odenthal et al. the linear steering control is extended by nonlinear emergency steering and braking control, see [9]. In terms of the autonomic vehicle control the combination of the brake and the throttle has been proposed, see [6], [12].

In an earlier work of our project a combined control mechanism, in which both the active anti-roll bars and the active brake control are applied, was developed, see [5]. The purpose of the compensator is to reduce the lateral tire forces acting on the outside wheel. These compensators, however, do not only have effects on the roll dynamics of the vehicle but they also modify the desired path of the vehicle, so they affect the yaw motion. Choosing any solution to the rollover prevention, the tracking error is increased.

Thus, in this paper a combined control mechanism is applied, which only guarantees the tracking in a normal situation. However, it also guarantees rollover avoidance in an emergency situation. 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 side 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 end its model. The active brake is switched on in an emergency situation and it is switched off after the emergency. Using such switching structures a chattering phenomenon may occur, and it may degrade the performance properties of the vehicle. A possible extension of the combined control structure is proposed for solving the chattering problem.

The structure of the paper is as follows. In Section 2 the LPV structure of the combined yaw-roll model is constructed. In Section 3 the LPV model for control design is constructed and the solution to the chattering problem is also presented. In Section 4 the the method of the LPV control design is presented. In Section 5 the combined control mechanism is demonstrated. Finally, Section 5 contains some concluding remarks.

J. Bokor and P. Gaspar are with the Systems and Control Laboratory, Computer and Automation Research Institute, Hungarian Academy of Sciences, Budapest, Hungary. E-mail: gaspar@sztaki.hu.

I. Szaszi is with the Department of Control and Transport Automation, Budapest University of Technology and Economics, Budapest, Hungary. E-mail: szaszi@kaut.kka.bme.hu.

II. The LPV 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.

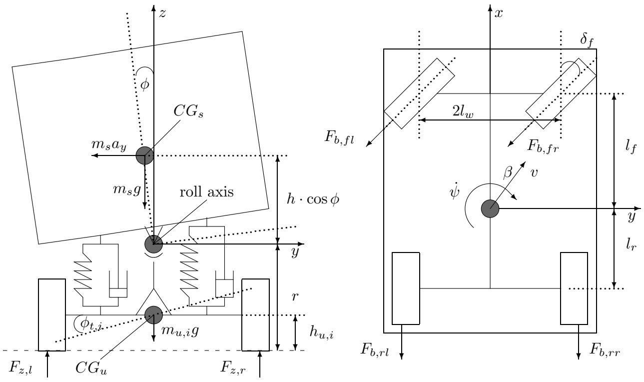

Fig. 1. Vehicle model of yaw-roll dynamics

In the vehicle modelling the motion differential equations of the yaw-roll dynamics of the single unit vehicle are formalized, i.e. 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 symbols of the combined yaw-roll model are found in Table I.

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

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

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

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

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

where $Y_{\beta,i}$, $Y_{\dot\psi,i}$, Y_{δ_f}, N_β, $N_{\dot\psi}$, N_{δ_f} are the tyre coefficients.

TABLE I
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
C_i	tire cornering stiffness
F_{zi}	total axle load
R_i	normalized load transfer
k_i	suspension roll stiffness
b_i	suspension roll damping
$k_{t,i}$	tire roll stiffness
I_{xx}	roll moment of inertia of sprung mass
I_{xz}	yaw-roll product of inertial of sprung mass
I_{zz}	yaw moment of inertia of sprung mass
l_i	length of the axle from the CG
l_w	half of the vehicle width
μ	road adhesion coefficient

These equations can be expressed in the state space representation. 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:

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

The state equation is formalized in the following form:

$$\dot x = A(v)x + B\begin{bmatrix}\delta_f\\ \Delta F_b\end{bmatrix} \tag{7}$$

$$y = Cx \tag{8}$$

where the control inputs 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. In our case it is assumed that the difference in the brake forces ΔF_b provided by the compensator is applied on the rear axle. The measured outputs are the lateral acceleration of the sprung mass a_y and the derivative of the roll angle $\dot\phi$.

The form of the LPV system is as follows:

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

where ρ vector contains the scheduling parameters. 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 equation (7)). Choosing the forward velocity v as a scheduling parameter, the differential equations of the yaw-roll motion are linear in the state variables.

The aim of the control design is to minimize the tracking error and to avoid the rolling over in an emergency situation. The roll-over situation can be detected if the lateral load transfers for both axles are calculated. The lateral load transfer can be given:

$$\Delta F_{z,i} = \frac{k_{t,i}\phi_{t,i}}{l_w} \tag{10}$$

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.

$$R_i = \frac{\Delta F_{z,i}}{F_{z,i}} \tag{11}$$

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. Using the brake system of the vehicle a yaw moment can be generated by unilateral brake forces, which can reduce the lateral acceleration directly.

III. The construction of the LPV model for control design

In this section the control design for the combined tracking and the roll stability is discussed. Consider the closed-loop system in Figure 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. The control inputs are the front wheel steering angle, δ_f, and the difference in brake forces, ΔF_b. The measured outputs are the lateral acceleration of the sprung mass a_y and the yaw rate $\dot{\psi}$. The noises, n_{ay} and $n_{\dot{\psi}}$, are from the measurements. The performance signals z are the tracking error, e_r, the load transfer $\Delta F_{z,i}$ and the control inputs δ_f and ΔF_b. In this representation the uncertainty is the unmodelled dynamics.

The uncertainties of the model are represented by W_r and Δ_m. Design models used for tracking and roll stability control typically exhibit high fidelity at lower frequencies ($\omega < 10$ Hz), but they degrade rapidly at higher frequencies due to poorly modelled or neglected effects. Thus, W_r is selected as

$$W_r = 0.1\frac{s/2+1}{s/40+1} \tag{12}$$

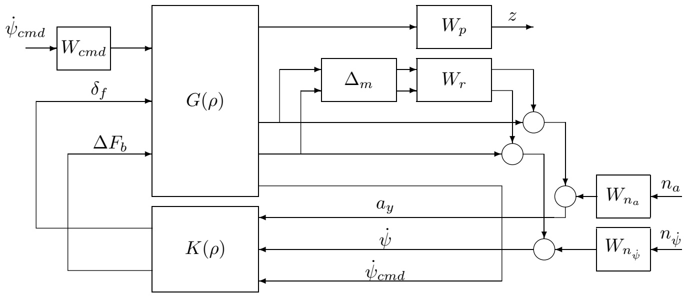

Fig. 2. The closed-loop interconnection structure

The input scaling weight W_{cmd} normalizes yaw rate command to the maximum expected command. It is selected 15 deg/sec yaw rate command. The dynamics of the reference input is as follows: $T = \frac{\omega^2}{s^2+2\zeta\omega s+\omega^2}$ with $\omega = 12$ and $\zeta = 1$.

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 deg/sec for the yaw rate $\dot{\psi}$.

The weighting function W_p represents the performance outputs: the tracking error, the lateral acceleration, and the control inputs. The weighting function of the tracking error is selected as:

$$W_{p_e} = 100\frac{(s/20+1)}{(s/0.1+1)} \tag{13}$$

The weighting function of the lateral acceleration is selected as:

$$W_{p_a} = \phi_a\frac{(s/100+1)}{(s+1)}. \tag{14}$$

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 weights of control inputs are: W_u for the steering angle is 1/20, and W_{F_b} for the brake force is 1/20. The reason for keeping the control signals small is to avoid the actuator saturation.

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 roll over 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.

ϕ_a is chosen to be parameter-dependent, i.e., the function of R_r. When R_r is small, i.e., 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 guarantee the yaw rate tracking by steering angle. 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 $\phi_a(R_r)$ is as follows:

$$\phi_a(R_r) = \begin{cases} 0 & \text{if } \ |R_r| < R_1 \\ \frac{2}{R_2 - R_1}(|R_r| - R_1) & \text{if } \ R_1 \le |R_r| \le R_2 \\ 2 & \text{otherwise} \end{cases} \tag{15}$$

R_1 defines the critical status when the vehicle is close to the rollover situation i.e. all wheels are in the ground but the lateral tire force of the inner wheels tends to zero. 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 control mechanism proposed in the combined control structure can be considered as a switching system. The active steering is primarily used to minimize the tracking error of the vehicle; the brake system is only activated when the vehicle is close to the roll-over. In practice, such switching structures are used a chattering may occur. Chattering causes small amplitude oscillations 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 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

$$R_1 = R_{nom} + \frac{\text{sgn}(\dot{R}_r)}{w_h} \tag{16}$$

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 equation (16), 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 sign$(\dot{R}_r)$ can be used as an additional scheduling parameter in the control design. In the design procedure, the possible values of sign$(\dot{R}_r)$ are selected $\{-1, 0, 1\}$.

In the LPV model of the yaw-roll motion three parameters are selected as scheduling parameters: the forward velocity v, the the normalized lateral load transfer at the rear side R_r, and the derivative of lateral load transfer at rear $\dot{R}_r$:

$$\rho = \begin{bmatrix} v & R_r & \dot{R}_r \end{bmatrix}^T \tag{17}$$

The parameter v is measured directly, while the parameter R_r can be calculated by using the measured roll angle of the unsprung mass $\phi_{t,r}$, and $\dot{R}_r$ can be calculated by numerical differentiation from R_r.

IV. Solution of the LPV-based control design

The LPV model to be controlled has a partitioned representation in the following way:

$$\begin{bmatrix} \dot{x}(t) \\ z(t) \\ y(t) \end{bmatrix} = \begin{bmatrix} A(\rho) & B_1(\rho) & B_2(\rho) \\ C_1(\rho) & D_{11}(\rho) & D_{12}(\rho) \\ C_2(\rho) & D_{21}(\rho) & D_{22}(\rho) \end{bmatrix} \begin{bmatrix} x(t) \\ w(t) \\ u(t) \end{bmatrix} \tag{18}$$

where ρ is the scheduling vector.

The induced $\mathcal{L}_2$ norm of the LPV system $G_{\mathcal{F}_\mathcal{P}}$, with zero initial conditions, is defined as

$$\|G_{\mathcal{F}_\mathcal{P}}\|_\infty = \sup_{\rho \in \mathcal{F}_\mathcal{P}} \ \sup_{\|w\|_2 \neq 0, w \in \mathcal{L}_2} \frac{\|z\|_2}{\|w\|_2} \tag{19}$$

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

Definition 1: Given a compact set $\mathcal{P} \subset \mathcal{R}^S$, and a function $A : \mathcal{R}^S \to \mathcal{R}^{n \times n}$, the function A is parametrically dependent stable over $\mathcal{P}$ if there exist a continuously differentiable function $X : \mathcal{R}^S \to \mathcal{R}^{n \times n}$, $X(\rho) = X^T(\rho) > 0$ such that

$$A^T(\rho)X(\rho) + X(\rho)A(\rho) + \sum_{i=1}^{s} \left(\nu_i \frac{\partial X}{\partial \rho_i} \right) < 0 \tag{20}$$

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

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 then the quadratic stability because the equation (20) is solved by finding a parameter dependent $X(\rho)$ instead of a single X.

The quadratic LPV γ-performance problem is to choose the parameter-varying controller matrices $A_K(\rho)$, $B_K(\rho)$, $C_K(\rho)$, $D_K(\rho)$ such that the resultant 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 controller $K(\rho)$ is as follows

$$\begin{bmatrix} \dot{x}_K(t) \\ u(t) \end{bmatrix} = \begin{bmatrix} A_K(\rho(t)) & B_K(\rho(t)) \\ C_K(\rho(t)) & D_K(\rho(t)) \end{bmatrix} \begin{bmatrix} x_K(t) \\ y(t) \end{bmatrix} \tag{21}$$

The quadratic LPV γ-performance problem is solvable if there exist a matrix W, $W(\rho) = W^T(\rho) > 0$ such that

$$\begin{bmatrix} A_{clp}^T(\rho)W(\rho) + W(\rho)A_{clp}^T(\rho) & W(\rho)B_{clp}(\rho) & \gamma^{-1}C_{clp}^T(\rho) \\ B_{clp}(\rho)W(\rho) & -I_{n_d} & \gamma^{-1}D_{clp}^T(\rho) \\ \gamma^{-1}C_{clp}^T(\rho) & \gamma^{-1}D_{clp}^T(\rho) & -I_{n_e} \end{bmatrix} < 0 \tag{22}$$

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

Theorem 1: Given a compact set $\mathcal{P} \subset \mathcal{R}^S$, the performance level γ and the LPV system (18), 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 $Y : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$, such that for all $\rho \in \mathcal{P}$, $X(\rho) = X^T(\rho) > 0$, $Y(\rho) = Y^T(\rho) > 0$ and

$$\begin{bmatrix} \hat{A}X + X\hat{A}^T - \sum_{i=1}^{s}\left(\nu_i \frac{\partial X}{\partial \rho_i}\right) - B_2 B_2^T & X C_{11}^T & \gamma^{-1} B_1 \\ C_{11} X & -I_{n_e} & 0 \\ \gamma^{-1} B_1^T & 0 & -I_{n_d} \end{bmatrix} < 0 \tag{23}$$

$$\begin{bmatrix} \tilde{A}^T Y + Y\tilde{A} + \sum_{i=1}^{s}\left(\nu_i \frac{\partial Y}{\partial \rho_i}\right) - C_2^T C_2 & Y B_1 & \gamma^{-1} C_{11}^T \\ B_1^T Y & -I_{n_d} & 0 \\ \gamma^{-1} C_{11} & 0 & -I_{n_e} \end{bmatrix} < 0 \tag{24}$$

$$\begin{bmatrix} x & \gamma^{-1} I_n \\ \gamma^{-1} I_n & Y \end{bmatrix} \geq 0 \tag{25}$$

where $\hat{A}(\rho) = A(\rho) - B_2(\rho) C_{11}(\rho)$ and $\tilde{A}(\rho) = A(\rho) - B_1(\rho) C_2(\rho)$. The state space representation of the LPV controller $K(\rho)$ is constructed from the solutions $X(\rho)$ and $Y(\rho)$ of the LMI optimization problem.

The constraints given 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 infinite-dimensionality of the constraints is relieved by approximating the parameter set $\mathcal{P}$ by a finite, sufficiently fine grid $\mathcal{P}_{grid} \subset \mathcal{P}$. Second, the variables $X, Y : \mathcal{R}^S \rightarrow \mathcal{R}^{n \times n}$ by restricting the search to the span of a collection of known scalar basis functions. Pick scalar continuously 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 parameterize as

$$X(\rho) = \sum_{i=1}^{N_x} g_i(\rho) X_i, \quad Y(\rho) = \sum_{j=1}^{N_y} f_j(\rho) Y_j. \tag{26}$$

Recently, there is no analytical method to choose the basis functions, namely g_i and f_i. Intuitive rules for basis function selection are applied.

V. Demonstration example

In the demonstration example, cornering responses of a single unit vehicle model travelling at 70 kph can be seen. Figure 3 shows the time responses of the roll over prevention system to the cornering. In this case chattering elimination is not applied in the control design. 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 reaches the maximum value (13 deg/s) in 0.5 s and filtered at 4 rad/s to represent the finite bandwidth of the driver. Since the brake system is only activated when the vehicle is close to the roll-over, the yaw rate command is generated in such a way that the normalized load transfer reaches its critical value R_1.

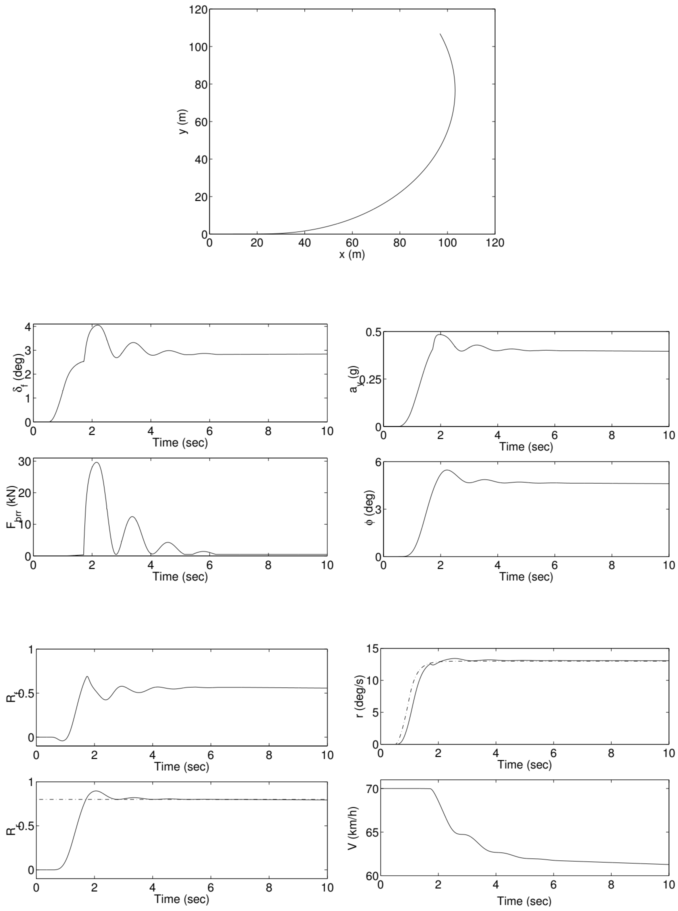

Fig. 3. Time responses to cornering when chattering elimination is not included

In the cornering situation as the lateral acceleration increases, the normalized load transfer lifts up the rear axle more quickly than the front one since the ratio of the effective roll stiffness to the axle load is greater at the driven axle. The relative roll angle does not exceed the acceptable limit, which is about $6 - 8$ degrees. Besides roll over avoidance, the controller also guarantees the tracking performance of yaw rate command. It can be seen that the tracking error is negligible in the yaw rate channel. The brake force is approximately 30 kN at the rear axle on the right-hand side. It can be observed that chattering in brake force occurs while the vehicle is decelerated. This phenomenon appears in such switching system when the controlled system reaches an equilibrium point with small amplitude oscillation. This small oscillation causes the brake system to switched on and off around the critical value R_1, and this degrades the roll stability of the vehicle.

In the next example, the extension of the roll over avoidance system with chattering elimination is analyzed (see Figure 4). 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. The yaw rate command and the initial forward velocity of the vehicle is exactly the same as in the previous maneuver.

Using the controller with chattering elimination the control algorithm is activated and the active brake system reduces the lateral acceleration when the normalized load transfer reaches the upper limit of the hysteresis characteristic. Due to the hysteresis characteristic the oscillation of brake force is disappeared. It can be seen that the vehicle is decelerated until the normalized load transfer reaches the lower limit of the hysteresis. When the brake control is activated, the rear right-hand-side wheel is braked to avoid the rollover of the vehicle, as shown in the brake force plot. Approximately 30 kN control force is required for the rear-right wheel during this maneuver. It can be stated that the tracking performance requirement associated with the yaw rate channel is fulfilled.

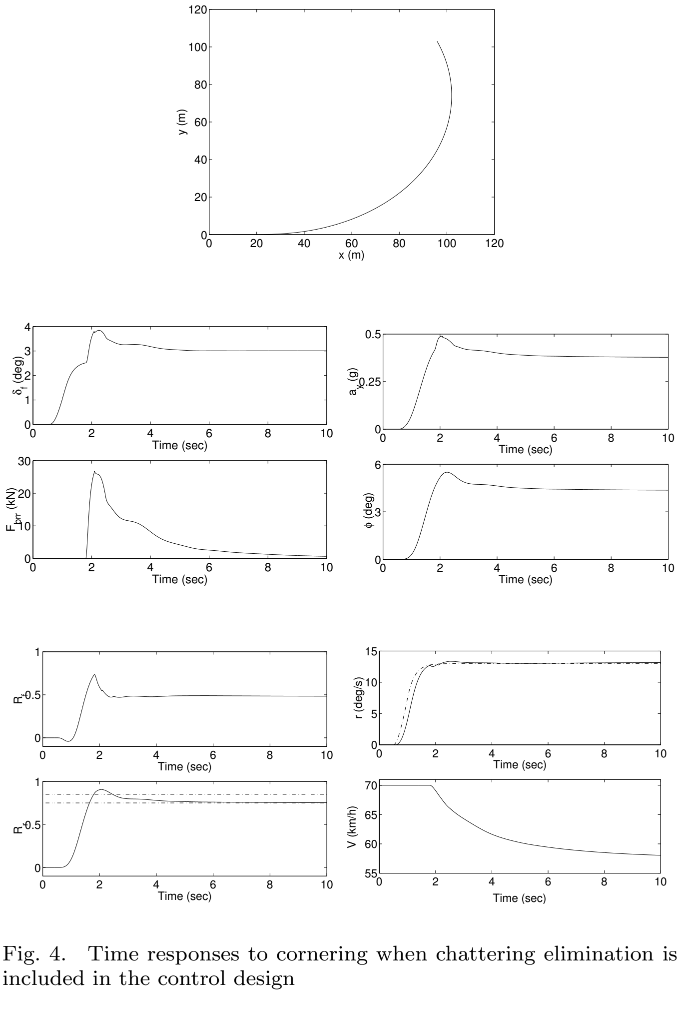

Fig. 4. Time responses to cornering when chattering elimination is included in the control design

VI. Conclusion

In the paper a combined control structure has been proposed for tracking the path of the vehicle and preventing rollover. In normal situations 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. The modelling and the control design are based on the LPV method. In the LPV model the forward velocity, the normalized lateral load transfer at the rear, and the critical value of the rollover are chosen as scheduling parameters. The LPV controller is able to handle the highly nonlinear model, as well as the performance demands and the model uncertainties. Since a chattering phenomenon may occur when using this combined control, the control structure is modified to eliminate this.

Acknowledgements

This work was supported by the Hungarian National Science Foundation (OTKA) under the grant $T-043111$ which is gratefully acknowledged.

References

[1] J. Ackermann, D. Odenthal, "Damping of vehicle roll dynamics by gain scheduled active steering", *Proc. European Control Conference*, 1999.

[2] J. Ackermann, T. Bnte, D. Odenthal, T. Bunte, "Advantages of active steering for vehicle dynamics control", *Proc. Int. Symposium on Automotive Technology and Automation*, Vienna, 1999.

[3] G. Balas, J.C. Doyle, "Robustness and performance tradeoffs in control design for flexible structures", *IEEE Trans. Control Systems Technology*, Vol. 2, No. 4, 352-361, 1994.

[4] Becker, G., A. Packard. " Robust performance of linear parametrically varying systems using parametrically-dependent linear feedback ", *System Control Letters*, 23, 205-215, 1994.

[5] Gaspar P., I. Szaszi, J. Bokor. "Reconfigurable control structure to prevent the rollover of heavy vehicles", *Safeprocess*, Washington, 2003.

[6] Hedrick J.K., J.C. Gerdes, D.B. Maciuca, D. Swaroop. "Brake system modelling, control and integrated brake/throttle switching", *PATH Project Proposal*, Berkeley, USA, 1997.

[7] Lin, R.C., D. Cebon, D.J. Cole. "Optimal roll control of a single-unit lorry", *Proc. IMechE, Journal of Automobile Engineering*, 210, 44-55, 1996.

[8] Moklegaard L., A.G. Stefanopoulou. "Advanced braking methods for longitudinal control of commercial heavy vehicles", *PATH Project Proposal*, Berkeley, USA, 2000.

[9] D. Odenthal, T. Bnte, J. Ackermann, "Nonlinear steering and braking control for vehicle rollover avoidance", *Proc. European Control Conference*, 1999.

[10] Palkovics, L., P. Michelberger, J. Bokor, P. Gaspar. "Adaptive on line identification for heavy-truck stability control", *Proc. of the 14th IAVSD Symposium*, Ann Arbor, Michigan, 44-46, 1995.

[11] Palkovics, L., A. Semsey, E. Gerum. "Roll-over prevention system for commercial vehicles - Additional sensorless function of the electronic brake system", *Vehicle System Dynamics*, 32, 285-297, 1999.

[12] Prohaska, R., P. Devlin. "Combined brake and steering actuator for automatic vehicle control", *PATH Project Proposal*, Berkeley, USA, 1998.

[13] Sampson, D.J.M., D. Cebon. "An investigation of roll control system design for articulated heavy vehicles", *Proc. 4th Int. Symposium on Advanced Vehicle Control*, Nagoya, 311-316, 1998.

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

[15] D. Wollherr, J. Mareczek, M. Buss, G. Scmidt, "Rollover avoidance for steerable vehicles by invariance control", *Proc. of the European Control Conference*, Porto, Portugal, 2001

[16] F. Wu, "A generalized LPV system analysis and control synthesis framwework ", *Int.J. Control*, 74, 745-759, 2001.