ADVANCED CONTROLLER DESIGN FOR ACTIVE BRAKING SYSTEMS

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ABSTRACT

This paper presents the synthesis of an advanced controller for an active braking system on trucks, by combining the H_∞ method with the Linear Parameter Varying system (LPV). The forward velocity and the normalized load transfer at the rear axle are considered as the two varying parameters. The grid-based LPV approach is used to synthesize the H_∞/LPV controller. In order to adapt the active braking system to the risk of the vehicle rollover, the parameter dependent weighting function for the lateral acceleration is used as the most important weighting function. The simulation results in the frequency domain show that the use of the H_∞/LPV active brake control system has significantly reduced the transfer function magnitude of the normalized load transfer at the two axles, which demonstrates the effectiveness of the proposed method.

Keywords: *Intelligent transportation systems; vehicle dynamics; active braking system; roll stability; H_∞ control; LPV system*

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THIẾT KẾ BỘ ĐIỀU KHIỂN NÂNG CAO CHO HỆ THỐNG PHANH CHỦ ĐỘNG

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TÓM TẮT

Bài báo này trình bày tổng hợp bộ điều khiển nâng cao cho hệ thống phanh chủ động trên ô tô tải, bằng cách kết hợp phương pháp H_∞ với mô hình hệ thống có thông số thay đổi tuyến tính LPV. Vận tốc chuyển động và hệ số chuyển tải ở cầu sau được xem xét là hai thông số thay đổi. Phương pháp LPV dựa trên cách tiếp cận chia lưới được sử dụng để tổng hợp bộ điều khiển H_∞/LPV. Để điều chỉnh hệ thống phanh chủ động đáp ứng được với các nguy cơ lật ngang của ô tô, hàm trọng số phụ thuộc vào gia tốc ngang được xem là thông số quan trọng nhất. Kết quả mô phỏng trong miền tần số cho thấy việc sử dụng hệ thống phanh chủ động điều khiển H_∞/LPV đã làm giảm đáng kể độ lớn hàm truyền từ góc đánh lái đến hệ số chuyển tải ở hai cầu xe, điều này đã chứng minh hiệu quả của phương pháp đề xuất.

Từ khóa: *Hệ thống giao thông thông minh; động lực của xe; hệ thống phanh chủ động; ổn định lật ngang; điều khiển bền vững H_∞; hệ thống LPV*

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1. Introduction

Active Braking System (ABS) is the general concept of controlled braking on vehicles, such as Electronic Brake System (EBS), Anti-lock Braking System (ABS), Advanced Emergency Braking System (AEBS), Autonomous Emergency Braking (AEB) [1], [2]. The active braking system was introduced to the automotive industry in the 1950s with the goal to improve braking performance. For a long time, hydraulic brake systems dominated the market; however, the main disadvantage is the noticeable oscillation of the wheel slip around a reference value. Today, electromechanical actuators are becoming common and will probably totally replace hydraulic brakes in the near future, along with the development of X-by-wire technology. These actuators allow the application of a smoother and continuous braking action on the brake pads [3]. The evolution of braking systems in the automotive field is well described in Figure 1 [4]. It indicates that, since electronics have been integrated into vehicles, the advances in the development of active vehicle control systems have been inextricably linked to advances in sensors and actuators technology [4]. The effect of the controlled suspension is only to keep the vehicle body perpendicular to the road, since it cannot reduce the lateral tyre force component. Therefore, the role of an active braking system in order to avoid the vehicle rollover situation is very important [5]-[7].

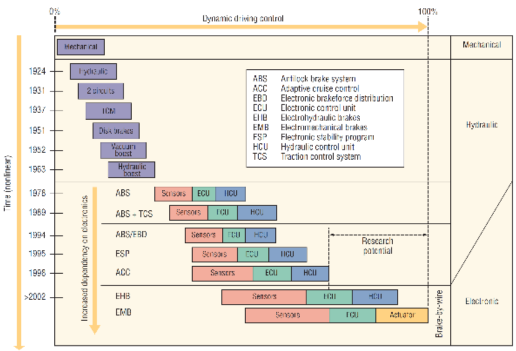

Figure 1. *The evolution of braking systems* [4]

LPV theory is closely related to gain scheduling via interpolation of point designs. LPV theory provides a mathematically rigorous approach to the design of gain-scheduled controllers. This includes powerful guarantees on the performance and robustness in the presence of time-varying dynamics and uncertainty. The LPV approach is considered in this paper. It is known to be able to handle system non-linearities by considering them as varying parameters and, as well, to make the controller performance varying through the linear introduction of smart (high-level) parameters in several chapters from the book [4].

In the literature, one can find the following references related to improved roll stability:

- In [8], a combined control structure between the active anti-roll bar system and the active braking system was proposed. The best part of this solution is that, in a normal driving situation, only the active anti-roll bar system is functional and the active braking system is only activated when the vehicle comes close to a rollover situation.

- In [3], a robust control algorithm for an anti-lock brake system is proposed. The method used is based on the static-state feedback of the longitudinal slip and does not involve controller scheduling with changing vehicle speed or road adhesion coefficient estimation.

Based on the idea in [8], here the authors would like to present preliminary research results on the active braking system with the aim of preventing the vehicle rollover phenomenon. Hence, this paper contributes the following elements:

- The active braking system is designed based on the control signal being the yaw moment control, which is generated by the difference in braking force at four wheels. This allows the controller to be synthesized more easily than when considering the braking force at each wheel.

- The grid-based LPV approach is used to set up an H_∞/LPV controller self-scheduled by two varying parameters: the forward velocity and the normalized load transfer of the rear axle. The parameter dependent weighting function for the lateral acceleration is used in order to adapt the vehicle performance to the risk of rollover.

The paper is organised as follows: Section 2 presents the yaw-roll model of a truck. Section 3 develops the H_∞/LPV control synthesis for an active braking system to prevent rollover. Section 4 introduces the grid-based LPV approach. Section 5 presents some simulation results in the frequency domain. Finally, some conclusions are drawn in section 6.

2. Vehicle modelling

The yaw-roll model of a truck for studying the active anti-roll bar system is presented in [9]. Here, this model is suitably modified for the active braking system by using the yaw moment control M_z as shown in Figure 2. The variables of the yaw-roll model are detailed in Table 1. The motion differential equations are formalized as follows:

$$\begin{cases} mv(\dot{\beta}+\dot{\psi}) - m_s h\ddot{\phi} = F_{yf} + F_{yr} \\ -I_{xz}\ddot{\phi} + I_{zz}\ddot{\psi} = F_{yf}l_f - F_{yr}l_r + M_z \\ (I_{xx} + m_s h^2)\ddot{\phi} - I_{xz}\ddot{\psi} = m_s gh\phi + m_s vh(\dot{\beta}+\dot{\psi}) \\ \qquad - k_f(\phi - \phi_{uf}) - b_f(\dot{\phi} - \dot{\phi}_{uf}) + M_{ARf} \\ \qquad - k_r(\phi - \phi_{ur}) - b_r(\dot{\phi} - \dot{\phi}_{ur}) + M_{ARr} \\ -rF_{yf} = m_{uf}v(r - h_{uf})(\dot{\beta}+\dot{\psi}) + m_{uf}gh_{uf}\phi_{uf} - k_{tf}\phi_{uf} \\ \qquad + k_f(\phi - \phi_{uf}) + b_f(\dot{\phi} - \dot{\phi}_{uf}) + M_{ARf} \\ -rF_{yr} = m_{ur}v(r - h_{ur})(\dot{\beta}+\dot{\psi}) - m_{ur}gh_{ur}\phi_{ur} - k_{tr}\phi_{ur} \\ \qquad + k_r(\phi - \phi_{ur}) + b_r(\dot{\phi} - \dot{\phi}_{ur}) + M_{ARr} \end{cases} \quad (1)$$

where $F_{yf,r}$ are the lateral tyre forces and $M_{ARf,r}$ the moments of the passive anti-roll bar system impact the unsprung and sprung masses at the two axles [10].

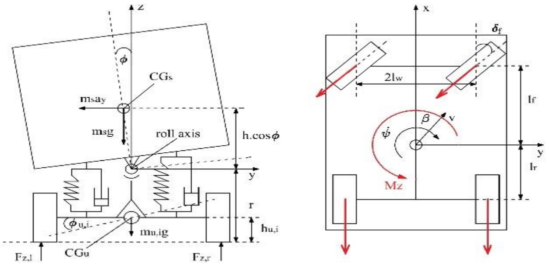

Figure 2. *Yaw-Roll model of a truck*

The yaw moment control M_z generated by the active braking system. In this case, we use an assumption that the driving throttle is constant during a lateral manoeuver and the forward velocity depends only on the brake forces at the four wheels (F_b). The differential equation of the forward velocity is: $m\dot{v} = -4F_b$ (2)

The motion differential equations (1)-(2) can be rewritten in the LPV state-space representation with the forward velocity as the varying parameter (ρ_1=v) as follows:

$$\dot{X} = A(\rho_1).X + B_1(\rho_1).W + B_2(\rho_1).U \quad (3)$$

with the state vector $X = \begin{bmatrix} \beta & \dot{\psi} & \phi & \dot{\phi} & \phi_{uf} & \phi_{ur} & \rho_1 \end{bmatrix}^T$, the exogenous disturbance $w = \begin{bmatrix} \delta_f \end{bmatrix}^T$ and the control input $u = \begin{bmatrix} M_z \end{bmatrix}^T$.

Table 1. *Variables and Parameters of the yaw-roll model*

Symbols	Description	Value	Unit
m_s	Sprung mass	12487	kg
$m_{u,f}$	Unsprung mass on the front axle	706	kg
$m_{u,r}$	Unsprung mass on the rear axle	1000	kg
m	The total vehicle mass	14193	kg
v	Forward velocity	-	km/h
v_{wi}	Components of the forward velocity	-	km/h
h	Height of CG of sprung mass from roll axis	1.15	m
$h_{u,i}$	Height of CG of unsprung mass from ground	0.53	m
r	Height of roll axis from ground	0.83	m
a_y	Lateral acceleration	-	m/s^2
β	Side-slip angle at center of mass	-	rad
ψ	Heading angle	-	rad
$\dot{\psi}$	Yaw rate	-	rad/s
α	Side slip angle	-	rad
ϕ	Sprung mass roll angle	-	rad
$\phi_{u,i}$	Unsprung mass roll angle	-	rad
δ_f	Steering angle	-	rad
C_f	Tire cornering stiffness on the front axle	582	kN/rad
C_r	Tire cornering stiffness on the rear axle	783	kN/rad

Symbols	Description	Value	Unit
k_f	Suspension roll stiffness on the front axle	380	kNm/rad
k_r	Suspension roll stiffness on the rear axle	684	kNm/rad
b_f	Suspension roll damping on the front axle	100	kN/rad
b_r	Suspension roll damping on the rear axle	100	kN/rad
k_{tf}	Tire roll stiffness on the front axle	2060	kNm/rad
k_{tr}	Tire roll stiffness on the rear axle	3337	kNm/rad
I_{xx}	Roll moment of inertia of sprung mass	24201	kgm^2
I_{xz}	Yaw-roll product of inertial of sprung mass	4200	kgm^2
I_{zz}	Yaw moment of inertia of sprung mass	34917	kgm^2
l_f	Length of the front axle from the CG	1.95	m
l_r	Length of the rear axle from the CG	1.54	m
l_w	Half of the vehicle width	0.93	m
μ	Road adhesion coefficient	1	-

3. The H∞/LPV controller synthesis for an active braking system

3.1. The H∞/LPV control design

In this section, the H∞/LPV control design is presented for the active braking system in heavy vehicles to prevent rollover in emergency situations. In Figure 3, the H∞/LPV control structure includes the nominal model $G(\rho_1)$, the controller $K(\rho_1, \rho_2)$, the performance output Z, the control input U, the measured output Y, and the measurement noise n. δ_f is the steering angle (disturbance signal), set by the driver. The measured output is $Y = [a_y, \dot{\phi}]$. The input scaling weight W_δ, chosen as $W_\delta = 0.02$. The weighting functions W_{n1} and W_{n2} are selected as: $W_{n1} = W_{n2} = 0.02$, which accounts for small sensor noise models in the control design.

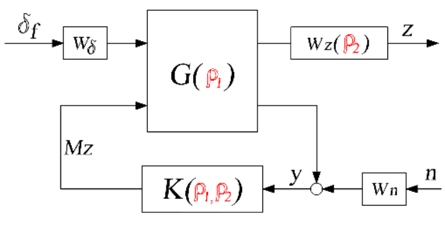

Figure 3. *Closed-loop of the active braking system*

The parameter dependent weighting function W_z represents the performance output and is chosen as

$$W_z = diag\left[\frac{\tau_1 s + \tau_2}{\tau_3 s + \tau_4}, \rho_2 \frac{\zeta_1 s^2 + \zeta_2 s + \zeta_3}{\zeta_4 s^2 + \zeta_5 s + \zeta_6}\right].$$

The purpose of this weighting function is to keep the yaw moment M_z and the lateral acceleration a_y as small as possible over the desired frequency range to over *4 rad/s*, which represents the limited bandwidth of the driver [8], [11]. The varying parameter is defined as $\rho_2 = f(|Rr|)$.

3.2. The solution of the H∞/LPV control problem

According to Figure 3, the concatenation of the nonlinear model (3) with the performance weighting functions has a partitioned representation in the following form:

$$\begin{bmatrix} \dot{X}(t) \\ Z(t) \\ Y(t) \end{bmatrix} = \begin{bmatrix} A(\rho) & B_1(\rho) & B_2(\rho) \\ C_1(\rho) & D_{11}(\rho) & D_{12}(\rho) \\ C_2(\rho) & D_{21}(\rho) & D_{22}(\rho) \end{bmatrix} \begin{bmatrix} X(t) \\ W(t) \\ U(t) \end{bmatrix} \quad (4)$$

where the exogenous input $W(t) = [\delta_f, n]$, the control input $U(t) = [M_z]$, the measured output vector $Y = [a_y, \dot{\phi}]$ and the performance output vector $Z(t) = [M_z\ a_y]^T$.

The LPV model of the active braking system (4) uses the varying parameters $\rho = [\rho_1; \rho_2]$, which are known in real time. The parameter $\rho_1 = v$ is measured directly, while the parameter $\rho_2 = f(|Rr|)$ can be calculated by using the measured roll angle of the unsprung mass at the rear axle ϕ_{ur}.

4. The grid-based LPV approach

The LPV system in the equation (3) is conceptually represented by a state-space system $S(\rho)$ that depends on a time varying parameter vector $\rho \in P_\rho$. A grid-based LPV model of this system is a collection of linearizations on a gridded domain of parameter values [12]. For general LPV systems, this conceptual representation requires storing the state-space system at an infinite number of points in the domain of ρ. For each grid point $\hat{\rho}_k$, there is a corresponding LTI system ($A(\hat{\rho}_k)$, $B(\hat{\rho}_k)$, $C(\hat{\rho}_k)$, $D(\hat{\rho}_k)$) which describes the dynamics of $S(\hat{\rho}_k)$ when $\hat{\rho}_k$ is held constant. It is worth noting that $\hat{\rho}_k$ represents a constant vector corresponding to the k^{th} grid point, while ρ_i is later used to denote the i^{th} element of the vector ρ. All the linearized systems on the grid have identical inputs u, outputs y and state vectors x. Together they form an LPV system approximation of $S(\rho)$ [13], [14].

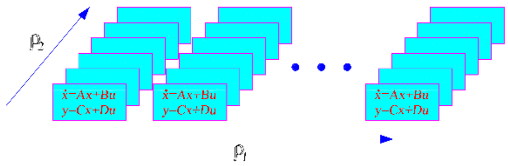

Figure 4. *LPV models defined on a rectangular grid*

The grid-based LPV approach is pictorially represented in Figure 4, with the example of such a system that depends on two parameters (ρ_1, ρ_2). The grid-based LPV approach approximates this conceptual representation by storing the LPV system as a state-space array defined on a finite, gridded domain. In this paper, we use the grid-based LPV approach and the LPVTools™ presented in [15], [16] to synthesize the H_∞/LPV active braking control system. It requires a gridded parameter space for the two varying parameters $\rho = [\rho_1, \rho_2]$.

The H_∞ controllers are synthesized for 10 grid points of the forward velocity in the range $\rho_1 = v = [40\ km/h;\ 130\ km/h]$ and 5 grid points of the normalized load transfer at the rear axle in a range $\rho_2 = f(|Rr|) = [0;\ 1]$. The grid points and the LPV controller synthesis using LPVTools™ are expressed by the following commands:

rho1 = pgrid('rho1',linspace(40/3.6,130/3.6,10));

rho2 = pgrid('rho2',linspace(0,1,5));

[Klpv,normlpv] = lpvsyn(H,nmeas,ncont).

5. Simulation results analysis

The parameters of the yaw-roll model of a truck are detailed in Table 1. To evaluate the effectiveness of the active braking system on the prevention of vehicle rollover in the frequency domain, the two following cases will be considered as:

- **1st case:** the varying parameters $\rho_1 = v$ varies from *40 km/h* to *130 km/h* and $\rho_2 = 0.8$;

- **2nd case**: the varying parameters $\rho_2 = [0, 0.8, 1]$ varies and $\rho_1 = v = 80\ km/h$.

5.1. 1st case: the varying parameters $\rho_1 = v$ varies from 40 km/h to 130 km/h and $\rho_2 = 0.8$

Vehicle rollover often occurs when the forward velocity is higher than *60 km/h*. Therefore, in this case, the author considers the varying parameter of the forward velocity $\rho_1 = v$ from *40 km/h* to *130 km/h*, while the varying parameter ρ_2 is kept constant at 0.8.

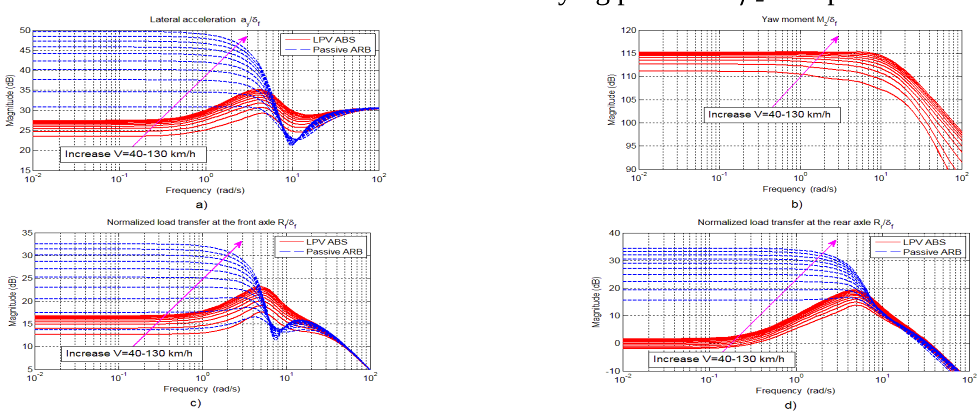

Figure 5. *The 1st case: transfer function magnitude of (a) the lateral acceleration* $\frac{a_y}{\delta_f}$*, (b) the yaw moment* $\frac{M_Z}{\delta_f}$*, (c, d) the normalized load transfers* $\frac{R_{f,r}}{\delta_f}$ *at the two axles*

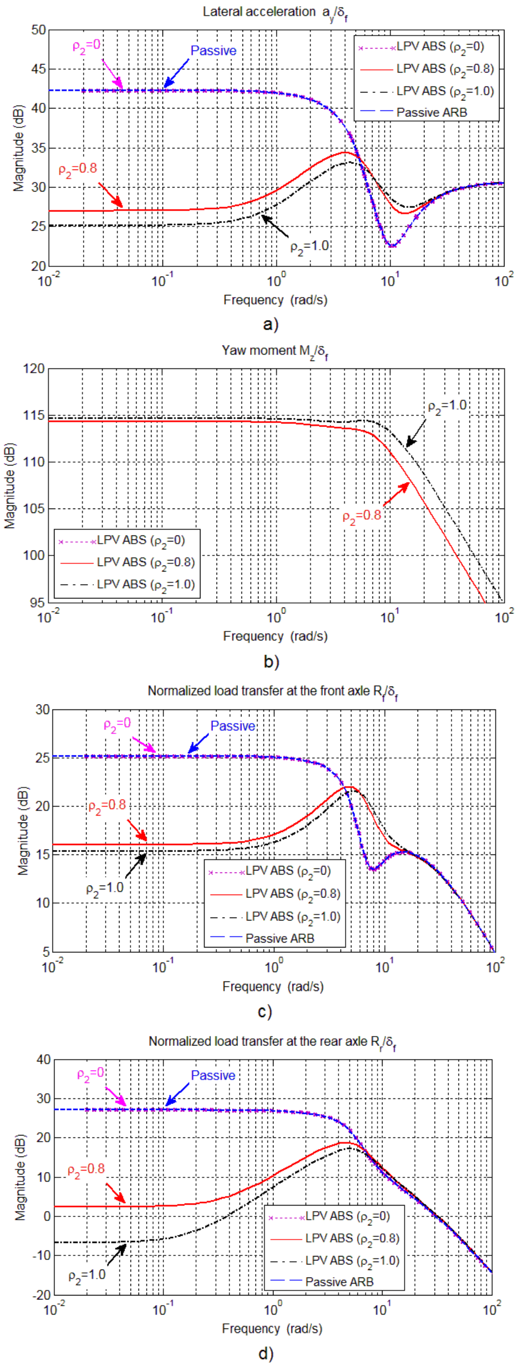

Figure 6. *2nd case: transfer function magnitude of (a) the lateral acceleration* $\frac{a_y}{\delta_f}$ *, (b) the yaw moment* $\frac{M_Z}{\delta_f}$ *, (c, d) the normalized load transfers* $\frac{R_{f,r}}{\delta_f}$ *at the two axles*

The objective of the H∞/LPV active braking control design is to prevent the vehicle rollover in an emergency situation with the considered frequency range to over *4 rad/s*. Figure 5 shows the transfer function magnitude of (a) the lateral acceleration, (b) the yaw moment, and (c, d) the normalized load transfers at the two axles. We can see that the H∞/LPV active braking control system reduces significantly the lateral acceleration and the normalized load transfers in the specified frequency range. By penalizing the lateral acceleration, the lateral tyre forces are reduced, therefore the normalized load transfers are also reduced.

5.2. 2nd case: the varying parameters ρ_2 = [0, 0.8, 1] varies and $\rho_1 = v$ = 80 km/h

In this case, the forward velocity is kept constant at $\rho_1 = v = 80$ *km/h*, while the varying parameter ρ_2 is surveyed at the three values: $\rho_2 = 0$, $\rho_2 = 0.8$, $\rho_2 = 1.0$. Figure 6 shows the simulation results in the frequency domain of the lateral acceleration, the yaw moment, as well as the normalized load transfers at the two axles. They show clearly the effect of the varying parameter ρ_2 to prevent vehicle rollover in the frequency range to over *4 rad/s*. When the varying parameter ρ_2 increases, the lateral acceleration and the normalized load transfers at the two axles decrease, which means that the active braking system can adapt to the rollover situation by increasing ρ_2. The reduction of the transfer function magnitude of the lateral acceleration and of the normalized load transfers when ρ_2= *[0, 0.8, 1]*, compared to the passive anti-roll bar system, is summarized in Table 2.

Table 2. *Reduction of the magnitude of the transfer functions compared to the passive anti-roll bar system*

Transfer functions	$\rho_2 = 0$	$\rho_2 = 0.8$	$\rho_2 = 1$
$\frac{a_y}{\delta_f}$	*0*	*16 dB*	*18 dB*
$\frac{R_f}{\delta_f}$	*0*	*9 dB*	*10 dB*
$\frac{R_r}{\delta_f}$	*0*	*25 dB*	*34 dB*

These simulation results indicated that the reductions of the normalized load transfers at the two axles are about 40% by using the active braking system. Moreover, with the two varying parameters, the controller can adapt to the vehicle rollover phenomenon in emergencies.

6. Conclusions

This paper proposes the first preliminary results on the combination of the active braking system and the passive anti-roll bar system for a truck. The grid-based LPV approach is used to synthesize the H_∞/LPV active braking controller, which is self-scheduled by two varying parameters. The parameter dependent weighting function for the lateral acceleration is used in order to allow for the vehicle performance adaptation to the risk of rollover. The simulation results in the frequency domain emphasize the efficiency of the proposed methodology.

In normal situations, the active braking system is in "off" mode, but when the normalized load transfer at the rear axle reaches its limit, the active braking system will be activated, thus improving the vehicle behaviour. Hence, in the future, a braking monitor will be needed in order that the H_∞/LPV active braking control system will satisfy simultaneously the two objectives, which are the prevention of vehicle rollover and the increased stability at the smooth switching points.

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