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Attitude and Handling Improvements Through Gain-scheduled Suspensions and Brakes Control

C. Poussot-Vassal[1] * O. Sename * L. Dugard * P. Gáspár ** Z. Szabó ** J. Bokor **

* *GIPSA-lab, Control Systems Department (former LAG), INPG-CNRS-ENSIEG - Domaine Universitaire BP46, 38402 Saint Martin d'Hères - Cedex FRANCE*
** *Computer and Automation Research Institute, Hungarian Academy of Sciences, Kende u. 13-17, H-1111, Budapest, HUNGARY*

Abstract: In this paper, the problem of comfort and handling improvements of a ground vehicle is treated through the control of the suspension and braking systems. Two gain-scheduled controllers are synthesized (in the $\mathcal{H}_\infty$ framework) to achieve, in the frequency domain, comfort and yaw performances according to the driving situation, observed by the mean of a monitor. The proposed strategy tackles the nonlinear tire braking force in an original way and meets the situation dependent objectives of the vehicle in a unified synthesis. Simulation tests on a complex nonlinear full vehicle model shows improvements of the proposed approach.

Keywords: $\mathcal{H}_\infty$, Linear Parameter Varying, Suspensions, Brakes, Global Chassis Control, Anti-locking Braking System, Electronic Stability Control.

1. INTRODUCTION

In most control design approaches, suspension and braking control systems are synthesized separately and tuned using empirical rules, derived thanks to the global knowledge of automotive engineers. This kind of approach may lead to conflicting control objectives or suboptimal control choices. As an illustration, suspensions are usually designed to improve either comfort (ABC, Active Body Control) or road holding according to the kind of vehicle (Zin et al., 2006), and braking system, used to tackle emergency situations such as slipping (ABS, Anti-locking Braking System) (Tanelli et al., 2007; Botero et al., 2007) or important lateral and yaw accelerations, when the driver might loose control of the vehicle (ESC/ESP, Electronic Stability Control / Program). Nowadays, academic and industrial research communities are very active in the Global Chassis Control (GCC) fields, that aims to improve both comfort and security on commercial cars by the development of integrated control that could tackle many different driving situations (Chou and D'Andrea-Novel, 2005; Gáspár et al., 2007). The GCC leads to various control problems such as MIMO (Multiple Input Multiple Output) controller synthesis, robustness analysis, variation of performances, which are important properties in the academic and industrial field, especially for implementation issues.

In this paper the focus is put on vertical and yaw motion control. Through the LPV (Linear Parameter Varying) theory, we propose a design methodology for an integrated global chassis controller, that can be viewed as an extension of (Gáspár et al., 2007) where focus was done on roll-over prevention, to overcome the usually conflicting comfort/security trade-off.

The here proposed LPV/$\mathcal{H}_\infty$ based strategy involves active suspensions and rear brakes to guarantee comfort in normal cruise situations and improves vehicle stability when emergency situations are detected (through a monitor), such as undesirable yaw rates. The developed strategy aims at simplifying engineer design, reducing development time in making actuators cooperate, guaranteing robustness properties with respect to model uncertainties, and internal stability (while the vehicle model is stable, see Section 2). As the controller is built to handle the strong nonlinearities of the tire braking forces, reproducing in an original way the ABS principle, the validation of the proposed design on a complex full vehicle model makes the solution complete and tractable for implementation issues.

The paper is structured as follows: Section 2 introduces the involved nonlinear model, then, both attitude and handling control designs, based on robust gain-scheduling techniques, are given in Section 3. In Section 4, typical driving situations are simulated using real vehicle data in the nonlinear model to show the improvements obtained by the proposed approach (and results are compared with fixed gain approach). Conclusions and perspectives are discussed in Section 5.

2. FULL VEHICLE MODEL AND NOTATIONS

The vehicle model involved in the sequel is the so called full vehicle model (Chou and D'Andrea-Novel, 2005; Gillespie, 1992; Smith and Wang, 2002; Zin, 2005). This model reproduces the vertical, longitudinal, lateral, roll, pitch and yaw dynamics of the chassis and the wheels, and the

[1] Corresponding author: charles.poussot@gipsa-lab.inpg.fr

$$
\left\{
\begin{array}{rcl}
\ddot{x}_s &=& \big((F_{tx_{fr}}+F_{tx_{fl}})cos(\delta)+(F_{tx_{rr}}+F_{tx_{rl}})-(F_{ty_{fr}}+F_{ty_{fl}})sin(\delta)+m\dot{\psi}\dot{y}_s-F_{dx}\big)/m \\
\ddot{y}_s &=& \big((F_{ty_{fr}}+F_{ty_{fl}})cos(\delta)+(F_{ty_{rr}}+F_{ty_{rl}})+(F_{tx_{fr}}+F_{tx_{fl}})sin(\delta)-m\dot{\psi}\dot{x}_s-F_{dy}\big)/m \\
\ddot{z}_s &=& -\big(F_{sz_{fl}}+F_{sz_{fr}}+F_{sz_{rl}}+F_{sz_{rr}}+F_{dz}\big)/m_s \\
\ddot{z}_{us_{ij}} &=& \big(F_{sz_{ij}}-F_{tz_{ij}}\big)/m_{us_{ij}} \\
\ddot{\theta} &=& \big((F_{sz_{rl}}-F_{sz_{rr}})t_r+(F_{sz_{fl}}-F_{sz_{fr}})t_f-mh\ddot{y}_s+(I_y-I_z)\dot{\psi}\dot{\phi}+M_{dx}\big)/I_x \\
\ddot{\phi} &=& \big((F_{sz_{rr}}+F_{sz_{rl}})l_r-(F_{sz_{fr}}+F_{sz_{fl}})l_f+mh\ddot{x}_s+(I_z-I_x)\dot{\psi}\dot{\theta}+M_{dy}\big)/I_y \\
\ddot{\psi} &=& \big((F_{ty_{fr}}+F_{ty_{fl}})l_f cos(\delta)-(F_{ty_{rr}}+F_{ty_{rl}})l_r+(F_{tx_{fr}}+F_{tx_{fl}})l_f sin(\delta)+(F_{tx_{rr}}-F_{tx_{rl}})t_r \\
&& +(F_{tx_{fr}}-F_{tx_{fl}})t_f cos(\delta)-(F_{tx_{fr}}-F_{tx_{fl}})t_f sin(\delta)+(I_x-I_y)\dot{\theta}\dot{\phi}+M_{dz}\big)/I_z
\end{array}
\right. \quad (1)
$$

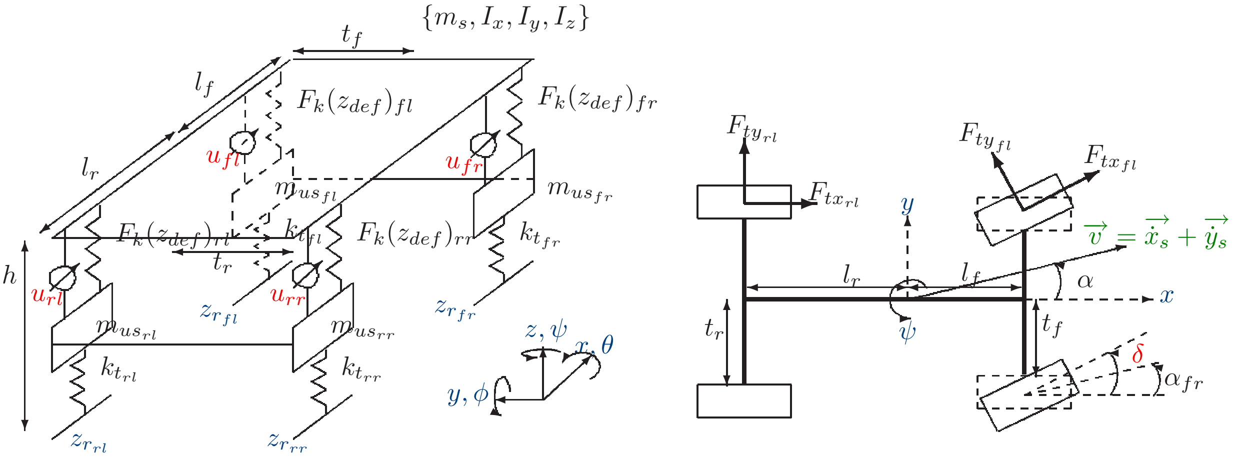


Fig. 1. Full vertical vehicle model (left) and Full lateral vehicle model (right).

rotational motions of the wheels (see equations (1), (3) and Figure 1). The dynamical equations are,

where z_s (resp. $z_{us_{ij}}$) denotes the chassis (resp. unsprung masses) bounce; θ, ϕ, ψ represent the chassis roll, pitch and yaw dynamics. Lateral and longitudinal dynamics are given by y_s and x_s. m_s and $m_{us_{ij}}$ hold for the chassis and suspended masses respectively; m is the total mass of the vehicle. I_x, I_y, I_z hold for the vehicle inertia in the x, y, z-axis. $F_{d(x,y,z)}$ (resp. $M_{d(x,y,z)}$) are external longitudinal, lateral and vertical (resp. roll, pitch and yaw) disturbances. l_f, l_r, t_f and t_r hold for the vehicle dimensions. Finally, the force provided by each suspension ($F_{sz_{ij}}$) and the tire vertical (longitudinal, lateral) forces $F_{tz_{ij}}$ ($F_{tx_{ij}}$, $F_{ty_{ij}}$) are defined by :

$$
\left\{
\begin{array}{rcl}
F_{tx_{ij}} &=& f(s_{ij}, F_{n_{ij}}) \text{ , Pacejka formulae} \\
F_{ty_{ij}} &=& f(\alpha_{ij}, F_{n_{ij}}) \text{ , Pacejka formulae} \\
F_{tz_{ij}} &=& k_{t_{ij}}(z_{us_{ij}}-z_{r_{ij}})+c_{t_{ij}}(\dot{z}_{us_{ij}}-\dot{z}_{r_{ij}}) \\
F_{sz_{ij}} &=& F_{k_{ij}}+u_{ij} \\
u_{ij} &=& F_{c_{ij}} \text{ , in the passive case} \\
u_{ij} &=& c_{0_{ij}}(.)+u_{ij}^{\mathcal{H}_\infty} \text{ , in the controlled case}
\end{array}
\right. \quad (2)
$$

$F_{k_{ij}} = k_{ij}(.)(z_{def_{ij}})$ and $F_{c_{ij}} = c_{ij}(.)(\dot{z}_{def_{ij}})$ denote the stiffness and damping nonlinear forces of the $\{i,j\}^{th}$ suspension (see Figure 2) ($i = f, r$ and $j = l, r$ give the position of each corner, obtained through geometrical base change). $k_{t_{ij}}$ is the tire stiffness and $c_{t_{ij}}$ the damping coefficient (negligible). s_{ij} defines the longitudinal slip coefficient and α_{ij}, the lateral slip angle; see (Canudas et al., 2003; Denny, 2005; Tanelli et al., 2007) and references therein. $z_{def_{ij}} = z_{s_{ij}} - z_{us_{ij}}$ (resp. $\dot{z}_{def_{ij}} = \dot{z}_{s_{ij}} - \dot{z}_{us_{ij}}$) are the suspensions relative deflections (resp. deflection speed) and u_{ij} holds for the suspension control input (i.e. the suspension added force). The slip ratio, s_{ij}, and the drift angle, α_{ij}, (angle between the longitudinal tire force and the vehicle speed direction) of each wheel are defined as :

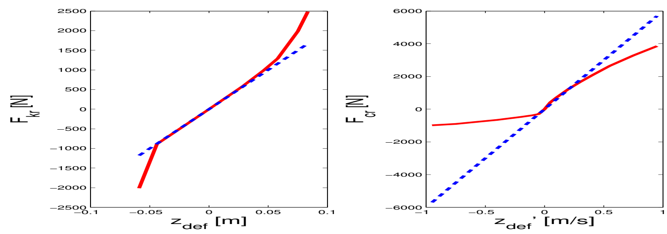


Fig. 2. Rear suspension stiffness (left) and damping (right) forces. Linear (dashed) and nonlinear (solid).

$$
\left\{
\begin{array}{rcl}
s_{ij} &=& 100(1-\frac{R\omega_{ij}}{v_{ij}})[\%] \\
I_w\dot{\omega}_{ij} &=& RF_{tx_{ij}}-T_{b_{ij}} \\
\alpha_{fj} &=& arctg\big(\frac{\dot{y}_{ij}}{\dot{x}_{ij}}\big)-\delta \text{ and } \alpha_{rj} = arctg\big(\frac{\dot{y}_{ij}}{\dot{x}_{ij}}\big)
\end{array}
\right. \quad (3)
$$

where ω_{ij} is the rotational speed of each wheel, I_w (resp. R) is the wheel inertia (resp. radius) and $T_{b_{ij}}$, the braking torque given by the braking controller (if any). The parameters of this model have been identified on a Renault Mégane Coupé (Zin et al., 2004). In Section 4, this model will be used as a reference ($u_{ij}^{\mathcal{H}_\infty} = 0$ and $T_{b_{ij}} = 0$) in order to evaluate the proposed control approach. In the controlled case, $u_{ij}^{\mathcal{H}_\infty}$ and $T_{b_{rj}}$ will be given by K_{susp} and K_{brake} (suspension and rear braking controllers respectively).

3. GLOBAL CHASSIS CONTROL DESIGN

For control design purpose, the full model can be split into two separate models, and the coupling phenomenon be considered as disturbances that the control design has to take into account:

(1) the vertical model, that is mainly used when comfort specifications and road holding in normal cruise situations are to be considered, involving vertical, roll and pitch dynamics (attitude).
(2) the lateral-longitudinal model that is used when yaw control and handling are studied, involving longitudinal, lateral and yaw dynamics (handling).

3.1 General structure

The main idea is to synthesize two controllers. The first is an active suspension one (based on the full vertical model) that focuses on vehicle attitude behavior in normal driving situations and improves road holding and handling in emergency or critical cases. The second is a rear braking system controller (based on the lateral-longitudinal model) that is activated in emergency situations to prevent critical yaw rate and lateral acceleration situations (in case of loss of manoeuvrability). As the two controllers performance objectives might vary, according to the driving situations (normal, emergency, critical) detected by a monitor, these controllers are designed using LPV methods. On Figure 3, the whole structure of the control strategy is given,

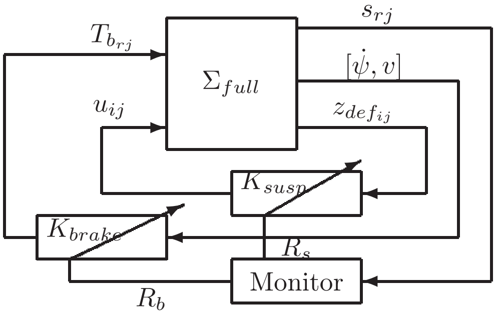


Fig. 3. General vehicle control structure.

where v denotes the vehicle speed, R_b (resp. R_s) is the scheduling parameter associated to K_{brake}, the braking (resp. K_{susp}, the suspension) controller.

3.2 Monitor

The aim of the monitoring is to give an image of the driving situation and to tune brake and suspension control objectives to overcome conflicting effects. On real vehicle, such a block may be much more complex, but here, as we focus on attitude and yaw stability, we will only consider the following strategy, based on the measurement of the longitudinal slip ratio of the rear wheels (s_{rj}). As introduced, two monitor variables are computed :

(1) **Braking monitor** $R_b = \min_{j=l,r}(r_{b_j})$, is a function of the absolute value of the slip ratio ($|s_{rj}|$). r_{b_j} is defined as a relay (hysteresis like) function: $\rightarrow 0$ when 'on', $\rightarrow 1$ when 'off'. Switch 'on' (resp. 'off') threshold equal to s^+ (resp s^-) (see Figure 4). When slipping is low, the vehicle is in normal situation, hence $R_b \rightarrow 1$. When the slip ratio raises and became greater than s^+, critical situation is detected, then $R_b \rightarrow 0$. As R_b is function of the slip ratio, the choice of s^+ (resp. s^-) is done according to the tire friction curve. Here (and in a general case), $s^+ = 9\%$ and $s^- = 8\%$, in order to delimitate the linear and peak tire friction force with the unstable part of the tire. Refer to Section 3.3 to see how the braking controller is tuned according to the R_b parameter and its role in the control strategy.

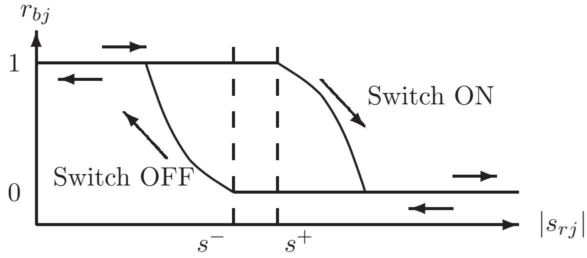


Fig. 4. r_{bj} as a function of the rear slip $|s_{rj}|$.

(2) **Suspension monitor** R_s, is defined as :

$$R_s \begin{cases} \rightarrow 1 & \text{when } 1 > R_b > R^2_{crit} \\ = \dfrac{R_b - R^1_{crit}}{R^2_{crit} - R^1_{crit}} & \text{when } R^1_{crit} < R_b < R^2_{crit} \\ \rightarrow 0 & \text{when } 0 < R_b < R^1_{crit} \end{cases} \tag{4}$$

when $R_b > R^2_{crit} (= 0.9)$, i.e. when low slip ($< s^-$) is detected, the vehicle is not in an emergency situation. When $R_b < R^1_{crit} (= 0.7)$, i.e. when high slip ($> s^+$), we reach a critical situation. Intermediate values of R_b will give intermediate diving situation. Hence as explained in Section 3.4, in the first case, suspension will be set to comfortable and in the second one to road-holding (intermediate performances will be reached for R_b values in between).

In the following, both controllers are derived thanks to the LPV/$\mathcal{H}_\infty$ methodology in order to meet the monitor requirements. Such a synthesis makes possible to smoothly change control performances thanks to parameters (here R_b and R_s), guaranteeing internal stability (avoiding switching) and minimizing the $\mathcal{H}_\infty$ norm (Bruzelius, 2004; Scherer et al., 1997).

3.3 Braking controller

The braking system aims at improving handling, avoiding emergency situations such as yaw rate error and large lateral accelerations. But one of the main problem in brake control is to provide an optimal force with respect to the nonlinear tire characteristics. For this purpose, many works concerning tire and brake control have been done in the last decade. Here, this problem is tackled by using the LPV/$\mathcal{H}_\infty$ design methodology where the varying parameter is the brake monitor which is a new contribution. R_b aims at ensuring that the required braking force remains in the linear stable zone of the tire characteristic and close to the maximal braking force. As the optimal braking strategy is not the main purpose of this paper (i.e. the estimation of the wheel slip ratio), the slip ratio of the rear wheels is assumed to be known. Here, as we aim at attenuating yaw rate error, caused by lateral forces and yaw moment disturbances that can lead the vehicle to instability, one introduces the following weighting functions and generalized plant, based on the longitudinal lateral model (Figure 5) :

where $W_{e_{\dot{\psi}}} = 10^{-2}\frac{s+1000}{s+1}$ gives the performance with respect to the yaw rate error (reference being pro-

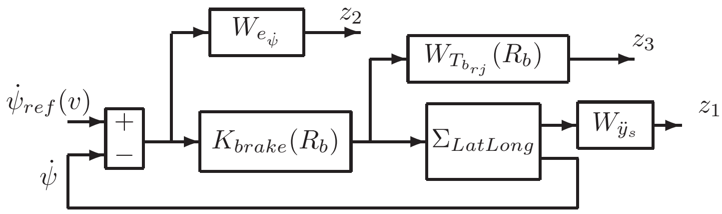


Fig. 5. Braking system generalized plant.

vided by a nonlinear undisturbed bicycle model), $W_{\ddot{y}_s} = 10^{-2}\frac{s/2000+1}{s/12+1}$ is devoted to lateral acceleration attenuation and $W_{T_{b_{rj}}}(R_b) = 3R_b 10^{-4}$ is a parameter dependent weigh acting on the controller gain. When $R_b \rightarrow 1$, which corresponds to a low slip ratio, the control signal gain would be high. Conversely, when the slip ratio is higher and enters the unstable tire zone, $R_b \rightarrow 0$, then the control signal will be attenuate. This mechanism will lead to bring back the slip ratio to lower values. As the R_b parameter is varying in an hysteresis way, the slip ratio will be 'trapped' between s^- and s^+ when high torque is required, which guarantees good braking torque and avoids slipping, reproducing the ABS working principle (see Figure 9).

Remark : As long as the controller design is done on a linear model, in case of high braking control, the force required may exceed the achievable one, given by the tire characteristic, and lock the wheel, which is dangerous for the passengers and the tire system. Hence, R_b monitoring step is essential to obtain good braking performances.

3.4 Suspension controller

Attitude control is ensured through suspension system, in order to achieve frequency specifications performances (Poussot-Vassal et al., 2006; Sammier et al., 2003). This controller is tuned thanks to the LPV/$\mathcal{H}_\infty$ techniques using a full linear vertical model and the following generalized plant (Figure 6) and parameterized weighting functions :

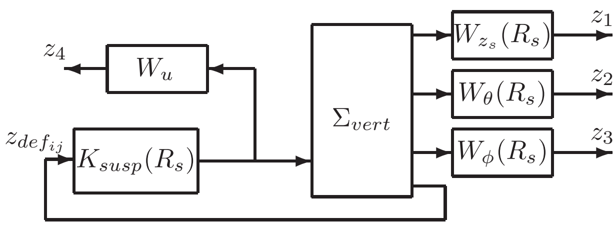
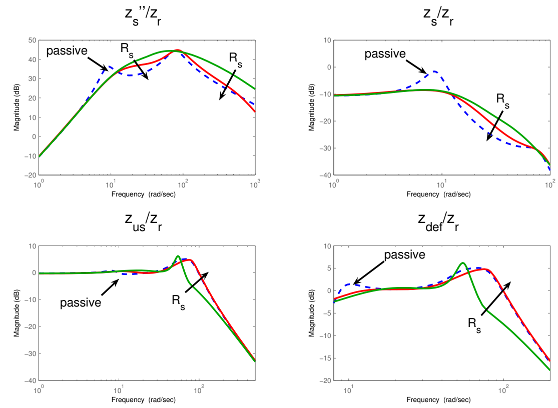
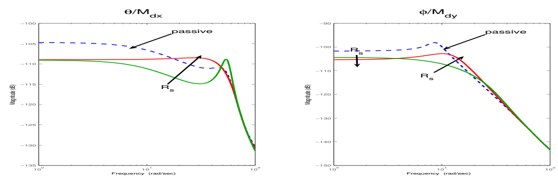


Fig. 6. Suspension system generalized plant.

where $W_{z_s}(R_s) = \frac{2}{s/(2\pi f_1)+1}R_s$ is shaped in order to reduce bounce amplification of the suspended mass (z_s) between $[0, 8]$Hz ($f_1 = 10$Hz), $W_\theta(R_s) = \frac{2}{s/(2\pi f_3)+1}(1 - R_s)$ attenuates the roll amplification in low frequency and the frequency peak at $9Hz$ ($f_2 = 2$Hz) and $W_\phi(R_s) = \frac{2}{s/(2\pi f_2)+1}R_s$ reduces the pitching moment especially in low frequency ($f_3 = 8$Hz). $W_u = 3.10^{-2}$ is set to shape the control signal. When $R_b > R^2_{crit}$, the braking is in the linear zone (tire stable zone), hence, suspensions are tuned to improve comfort (i.e. $R_s \rightarrow 1$). Conversely, when $R_b < R^1_{crit}$, braking became critical, hence, suspensions are set to road holding (i.e. $R_s \rightarrow 0$). The applied control law is given by: $u_{ij} = c_0(1 - R_s) + u_{ij}^{\mathcal{H}_\infty}$ where $u_{ij}^{\mathcal{H}_\infty}$ is obtained by the $\mathcal{H}_\infty$ design. Figure 7 gives the closed-loop Bode diagram for $R_s \in]0, 1[$ (and compare it to the passive reference Renault Mégane Coupé model).



(a) Displacements



(b) Angles

Fig. 7. Frequency responses of the suspension system

When $R_s \rightarrow 1$ (resp. $\rightarrow 0$), the suspension tends to improve comfort while deteriorating road-holding (and reciprocally). For deeper insight on the design, see earlier papers of the authors (Gáspár et al., 2007). The originality relies on the scheduling of the weighting functions that make the controllers function of the suspension monitor.

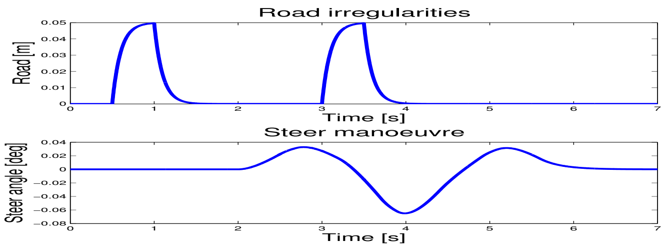
4. SIMULATION EXAMPLES

4.1 Scenario & Monitored signals

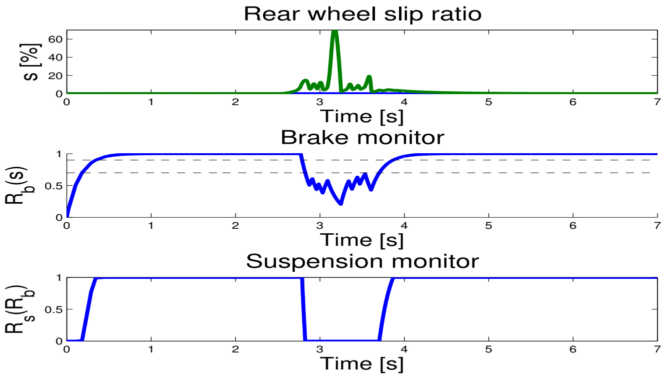
Simulations are performed on the full nonlinear vehicle model given in Section 2, including nonlinear suspensions forces, that uses values identified on a Renault Mégane Coupé. In the sequel, the performances obtained by the proposed gain-scheduled controller, denoted as 'LPV', are analyzed and compared to the Renault Mégane Coupé car (without control, denoted as 'Reference car') and, for sake of completeness, with a simple LTI/$\mathcal{H}_\infty$ controller (without scheduled gains), denoted as 'LTI'. The following scenario is used (see also Figure 8.a):

(1) the vehicle runs at 130km/h in straight line
(2) 5cm bump on the left wheels (from $t = 0.5$s to $t = 1$s)
(3) double line change manoeuvre is performed (from $t = 2$s to $t = 6$s)
(4) lateral wind occurs at vehicle's front, generating an undesirable yaw moment (from $t = 2.5$s to $t = 3$s)
(5) 5cm bump on the left wheels, during the manoeuvre (from $t = 3$s to $t = 3.5$s)

The following input signals and resulting monitored signals are obtained (see Figure 8).



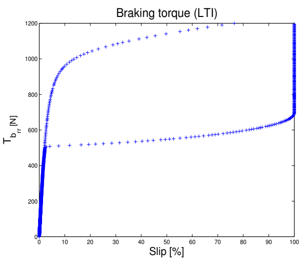
(a) Input signals



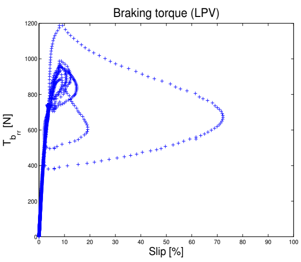
(b) Slip ratio & Monitored signals

Fig. 8. Input signals and Monitored signals.

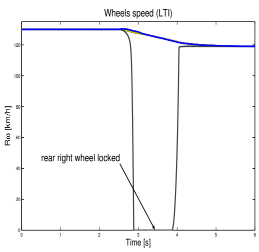
At time $t = 2.5$s, a positive yaw moment disturbance is generated, a yaw rate error is detected, hence the rear right wheel brake is activated to overcome the loss of manoeuvrability and the suspensions (usually tuned to improve comfort) are set to road holding mode ($R_s = 0$). As the braking force required is higher than the limit of the actuator (1200 Nm) and will tend to lock the rear right wheel, the R_b monitor (thanks to the slip ratio measure) attenuates the control gain of the braking controller in order to reduce the torque control and brings back the slip ratio close to the linear and maximum braking forces. It results an anti-locking wheel system (Figure 9)



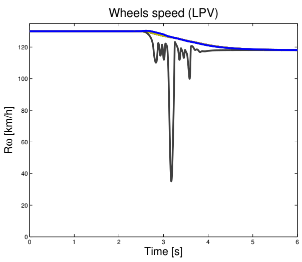
(a) LTI design



(b) LPV design



(c) LTI design



(d) LPV design

Fig. 9. (a)-(b) Rear right brake control (function of the slip ratio). (c)-(d) Wheel & Vehicle speed.

Thanks to the braking monitored value (R_b), that reproduces the ABS principle, the vehicle using the gain-scheduled controller provides a braking force that stays in the optimal zone and avoids slipping (at the end of the manoeuvre the LPV controlled system speed is 118km/h and the LTI one 120km/h).

4.2 Vehicle attitude & Handling analysis

The vertical behavior is given on Figure 10,

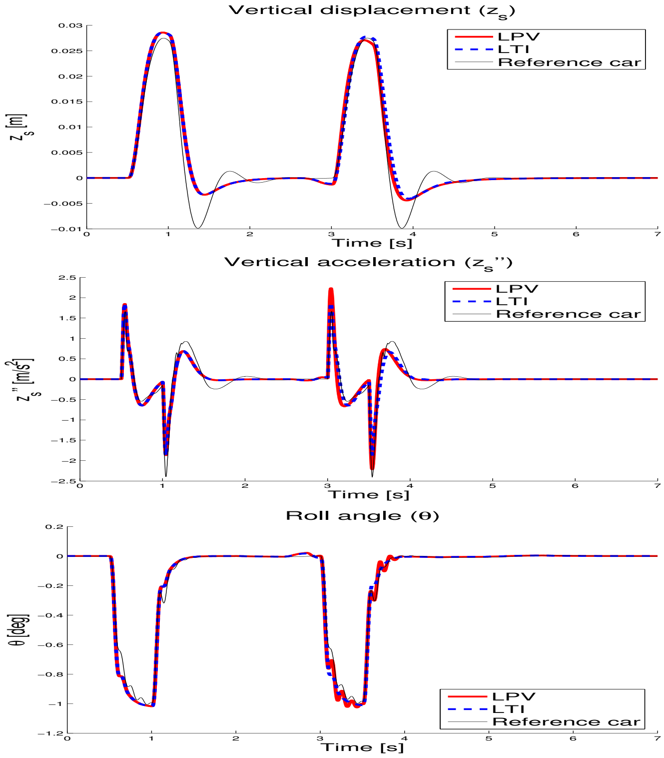


Fig. 10. Chassis attitude (bounce and orientation).

The results obtained are consistent with the Bode diagrams obtained in Section 3. It is interesting to note here that at the first bump, vertical bounce and acceleration are much more improved than during the second one. Note also that while the LTI controller gives the same performances for the two bumps, the LPV one degrades its comfort performances during the manoeuvre (after 2s), in order to focus on road-holding. On Figure 11, the handling performances are analyzed.

As previously observed, the brakes controlled system significantly improves the vehicle's handling by attenuating the yaw moment disturbance, allowing the vehicle to limit lateral and yaw errors which may lead to a road exit. As long as the gain-scheduled controller brakes in a better way, it gives an even better result.

5. CONCLUSIONS

In this paper, a global chassis strategy is introduced, involving brake and suspension control systems, in order to handle the compromises between driving situations in a unified way. This work extends previous results obtained in (Gáspár et al., 2007) where rollover prevention was

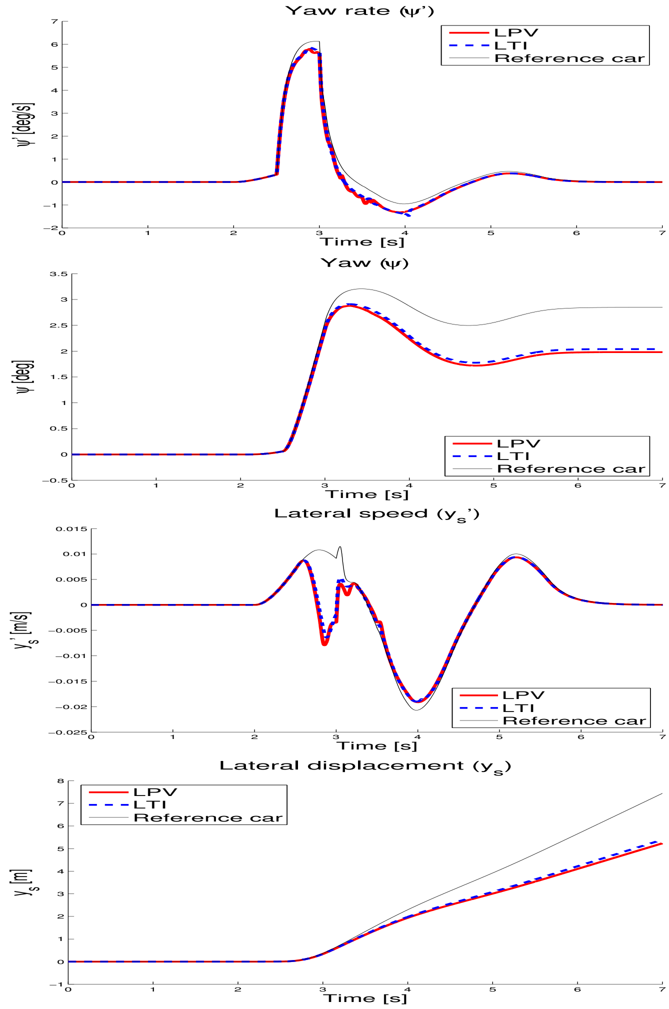


Fig. 11. Handling performances.

to be handled. Here, both yaw stability and comfort are improved using gain-scheduled robust methodology. The braking strategy, that reproduces the ABS working principle, is also designed in an original way. The authors stress that an advantage of such a method is that the exact knowledge of the tire force curve is not needed to guarantee good performances. A rough idea of the linear and peak value zone is enough to obtain good results. By the way, as far as this part is not the main point of the paper, one can use the brake controller (K_{brake}) as a reference torque control for an even much more efficient braking system (Falcone et al., 2007). Simulations of a consistent driving situation, performed on a complex nonlinear model, have shown the efficiency of the proposed approach. In practice, the slip ratio is not always available, therefore future works should include its estimation or sensitivity analysis of the control approach according to this parameter.

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