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A LPV suspension control with performance adaptation to roll behavior, embedded in a global vehicle dynamic control strategy

S. Fergani[1*], O. Sename[1], L. Dugard[1]

***Abstract*— This paper is concerned with a new LPV/H_∞ coordination strategy that improves the vehicle stability using suspension, active steering and electro-mechanical braking actuators. The new coordination technic aims, by monitoring the load transfer distribution of the vehicle while facing road irregularities, at tuning the suspensions in the four corners of the vehicle and improving vertical performances. At the same time, lateral and longitudinal car dynamics are improved using braking and steering actuators scheduled through the stability index, based on slip dynamics. The proposed GCC (Global Chassis Control) provides a load allocation on the four corners of the vehicle, ensures a good coordination between different actuators, and improves the vehicle stability and car dynamics. Simulations performed on a complex nonlinear full vehicle model, subject to critical driving situations, show the reliability and the robustness of the proposed solution.**

I. Introduction

Vehicle control systems aim at enhancing driving characteristics by ensuring stability in critical situations. Several automotive manufacturers have been interested by improving vehicle's stability to prevent them from drifting, spinning or rolling over. Many solutions to active chassis control for stability enhancing were then developed in the last years. Controlled car stability depends on several actuators; among them, the suspension system mainly acts on vertical dynamics, while lateral and longitudinal dynamics mainly depend on braking and steering systems. This led to the common feature that, in most of the proposed design approaches, suspension, steering and braking control systems are synthesized independently to solve local stability problems. Furthermore, since these different dynamics are strongly coupled, it is difficult to coordinate them efficiently. Thus, the global collaboration between each subsystem (sensors, controllers, and actuators) is achieved using empirical rules, thanks to the automotive engineers experience. This is of course not optimal and may induce poor performances.

Nowadays, the trend in modern automotive industry is the application of multivariable active safety systems to improve vehicle stability and handling. For this sake, many global chassis control systems have been developed and used in the automotive industry : *e.g.*, Electronic Stability Control that enhances vehicle lateral stability. Chassis control systems development is then an intense research field in industry and academic community.

Many papers have tackled this subject to bring innovative solutions. Most of the studies have been concerned with the lateral control using braking and steering actuators. A strategy using active steering for vehicle handling improvement was presented in [1]. In [2] an optimal non linear vehicle control based on individual braking torque and steering angle with online control allocation to improve vehicle performances is presented. A robust yaw control using an active differential and IMC techniques has been used to improve vehicle stability. In [3] and [4] a new design of actuator intervention for trajectory tracking and stability improvement is developed.

In [5], [6] the vehicle yaw control problem is tackled, considering steering and braking control actions coordinated through different LPV strategies. In [5] a specific LPV controller structure allows the activation of the actuators while handling the actuators limitation constraints. In [6] the coordination is ensured through a vehicle stability index, allowing to provide a hierarchical use of the actuators.
On the other hand there are few studies on the simultaneous use of 3 types of actuators, namely suspension, steering and braking control. Some of the authors have already proposed different LPV/$\mathcal{H}_\infty$ approaches to control vehicle dynamics. In [7] and [8] the coordination of active suspension and braking control actions is handled by monitoring the longitudinal slip dynamics. In [9] the vehicle dynamics are evaluated through the (suspension and braking) monitors given in (chapitre de livre) and the performance of the active suspension (from soft to hard) is coordinated accordingly.
In [10], the works recently developed in [9] have been extended to the case of cars equipped with semi active dampers (Magneto-Rheological dampers).
Finally in [11], the vehicle stability is monitored using the stability index considered in [6] and an additional parameter is defined to handle the load transfer distribution of the suspension control forces. The LPV/$\mathcal{H}_\infty$ strategy then allows to ensure the efficiency of the yaw controller and coordinates the use of the suspension actuator depending on the left/right load transfer.
The latter work is here extended providing two new main contributions:

- The suspension performance is here adapted to left/right load transfer which allows to coordinate the suspension behavior (from soft to hard) to the vehicle dynamical one.

[1] GIPSA-lab, Control Systems Department, Grenoble University - Grenoble INP, ENSE[3] - Domaine Universitaire BP46, 38402 Saint Martin d'Hères - Cedex FRANCE

* This work was supported by the French national project INOVE/ ANR 2010 BLAN 0308. Corresponding author: olivier.sename@gipsa-lab.grenoble-inp.fr

- The strategy developed in this paper allows to tune the suspension system to work either in a coordination mode with braking and steering systems to adapt to the driving situation, or alone when no other actions are required to satisfy the specified performances.

The strategy proposed for vehicle stabilization and performance improvement is presented (see Fig. 1), involving front active steering, rear braking and active suspension actuators, scheduled by two monitoring parameters (ρ_1 and ρ_2) provided by some supervision systems.

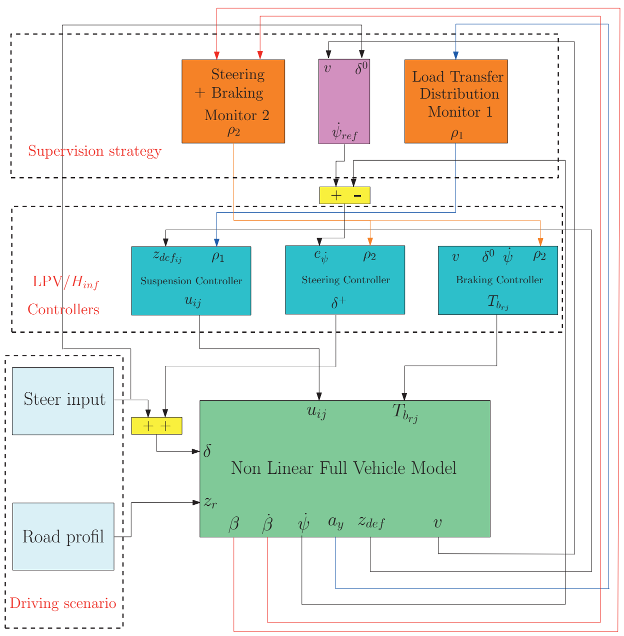

Fig. 1. Global chassis control Implementation scheme.

First, the load transfer distribution strategy allows to provide a varying parameter (ρ_1) that tunes the suspension behaviors at the four corners of the vehicle. The suspension is then set, based on this parameter, either to "soft" to enhance passengers comfort or to "hard" to keep the road holding and to help maintaining the stability of the car. The other varying parameter ρ_2 depends on the sideslip dynamics [6]; it ensures a good coordination between braking and steering actuators. It is used to generate the adequate braking torques and provides the additive steering angle to stabilize the vehicle in critical driving situations.

This whole supervision strategy induces a hierarchical activation of the different actuators. The LPV framework provides, in addition to stability, a smooth use of the coordination between steering, braking and suspension actuators, as well as a considerable help to the driver to overcome critical situations.

The paper is organised as follows: Section 2 briefly presents the vehicle model used for synthesis and validation purposes. Section 3 is devoted to the main contribution of the paper, which is to give a new LPV strategy for suspension control that better coordinates with the lateral control (using braking and steering actuators). The performance analysis is done in Section 4 with time domain simulations performed on a complex nonlinear full vehicle model. Conclusions are given in the last Section.

Paper notations:
Throughout the paper, the following notations will be adopted: indices $i = \{f, r\}$ and $j = \{l, r\}$ are used to identify vehicle front, rear and left, right positions respectively. Then, index $\{s, t\}$ holds for forces provided by suspensions and tires respectively. $\{x, y, z\}$ holds for forces and dynamics in the longitudinal, lateral and vertical axes respectively. Then let $v = \sqrt{v_x^2 + v_y^2}$ denote the vehicle speed, $R_{ij} = R - (z_{us_{ij}} - z_{r_{ij}})$ the effective tire radius, $m = m_s + m_{us_{fl}} + m_{us_{fr}} + m_{us_{rl}} + m_{us_{rr}}$ the total vehicle mass, $\delta = \delta_d + \delta^+$ is the steering angle (δ_d, the driver steering input and δ^+, the additional steering angle provided by the steering actuator, see Section III) and $T_{b_{ij}}$ the braking torque provided by the braking actuator (see Section III). The model parameters are those of a Renault Mégane Coupé, obtained during a collaborative study with the MIPS laboratory in Mulhouse, through identification with real data.

II. Background on vehicle modeling

A. Full vehicle model including active suspension, braking and steering actuators

The model (1) used for this work is a nonlinear full vehicle model. This model and the corresponding parameters can be found in [7]. It involves several car dynamics: vertical (z_s), longitudinal (v_x), lateral (v_y), roll (θ), pitch (ϕ) and yaw (ψ) dynamics of the chassis. It also models the vertical and rotational motions of the wheels ($z_{us_{ij}}$ and ω_{ij} respectively), the slip ratios ($\lambda_{ij} = \frac{v_{ij} - R_{ij}\omega_{ij}\cos\beta_{ij}}{\max(v_{ij}, R_{ij}\omega_{ij}\cos\beta_{ij})}$) and the center of gravity side slip angle (β_{cog}) dynamics as a function of the tires and suspensions forces. The main dynamical equations are given in equation (1), where $F_{tx_i} = F_{tx_{il}} + F_{tx_{ir}}$, $F_{ty_i} = F_{ty_{il}} + F_{ty_{ir}}$, $F_{tz_i} = F_{tz_{il}} + F_{tz_{ir}}$ are the tire forces (based on Pacejka tire non linear model) and $F_{sz_i} = F_{sz_{il}} + F_{sz_{ir}}$, $(i = \{f, r\})$.
Suspensions are modeled by a spring and a damping element, as:

$$F_{sz_{ij}} = k_{ij}(z_{s_{ij}} - z_{us_{ij}}) + c_{ij}(\dot{z}_{s_{ij}} - \dot{z}_{us_{ij}}) + u_{ij}^{\mathcal{H}_\infty} \quad (2)$$

where k_{ij} are the stiffness coefficients, c_{ij} are the damping coefficients and $u_{ij}^{\mathcal{H}_\infty}$ are the suspension control.
Note that the characteristics of the stiffness and the damping coefficients are non linear for simulation [12], and linear for control design.

III. Design of the Global Chassis Control Strategy

In this paper, a two step procedure is proposed for a hierarchical controllers synthesis. The first one is the braking and steering control synthesis, introduced previously by the authors in [6], that improves the stability of the vehicle

$$
\begin{cases}
\dot{v}_x &= -\big(F_{tx_f}\cos(\delta) + F_{tx_r} + F_{ty_f}\sin(\delta)\big)/m - \dot{\psi}v_y \\
\dot{v}_y &= \big(-F_{tx_f}\sin(\delta) + F_{ty_r} + F_{ty_f}\cos(\delta)\big)/m + \dot{\psi}v_x \\
\ddot{z}_s &= -\big(F_{sz_f} + F_{sz_r} + F_{dz}\big)/m_s \\
\ddot{z}_{us_{ij}} &= \big(F_{sz_{ij}} - F_{tz_{ij}}\big)/m_{us_{ij}} \\
\ddot{\theta} &= \big((F_{sz_{rl}} - F_{sz_{rr}})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f + mh\dot{v}_y\big)/I_x \\
\ddot{\phi} &= \big(F_{sz_f}l_f - F_{sz_r}l_r - mh\dot{v}_x\big)/I_y \\
\ddot{\psi} &= \big(l_f(-F_{tx_f}\sin(\delta) + F_{ty_f}\cos(\delta)) - l_rF_{ty_r} + (F_{tx_{fr}} - F_{tx_{fl}})t_f\cos(\delta) - (F_{tx_{rr}} - F_{tx_{rl}})t_r + M_{dz}\big)/I_z \\
\dot{\omega}_{ij} &= (R_{ij}F_{tx_{ij}} - T^f_{b_{ij}})/I_w \\
\dot{\beta}_{cog} &= (F_{ty_f} + F_{ty_r})/(mv_x) + \dot{\psi}
\end{cases} \tag{1}
$$

through the supervision of the stability index evolution. The second step is concerned with the suspension control at the four corners of the vehicle, based on the proposed new load transfer distribution.

A. Monitoring systems and Scheduling parameters generation

In this section the monitoring parameter dedicated to the suspension load transfer is new, while the LPV control strategy of the braking/steering system can be found in [6], [8].

Load transfer distribution monitoring system (ρ_1): The load transfer monitor is based on the evaluation of the roll load transfer when the vehicle is running. The main idea is to compare the right and left vertical forces in the four corners of the vehicle. The difference between the right and left vertical forces is used to schedule the suspensions actuators. Based on the ρ_1 value, the suspensions on the left and right sides of the vehicle are tuned either to "soft" or "hard" (resp. comfort or road holding performance objectives). This suspension monitor is characterized by the following equations:

$$
\begin{cases}
F_{z_l} &= m_s \times g/2 + m_s \times h \times a_y/l_f \\
F_{z_r} &= m_s \times g/2 - m_s \times h \times a_y/l_r \\
\rho_1 &= |(F_{z_l} - F_{z_r})/(F_{z_l} + F_{z_r})|;
\end{cases} \tag{3}
$$

where F_{z_l} and F_{z_r} are the vertical forces, a_y the lateral acceleration, ρ_1 the scheduling parameter. Note that $\rho_1 \in [0 \quad 1]$. The load transfer distribution is handled as follows: when $\rho_1 \rightarrow 1$, the suspensions actuators are tuned to enhance road holding and vehicle manoeuvrability by attenuating the vehicle roll motion; conversely, when $\rho_1 \rightarrow 0$ right suspensions are tuned to improve passengers comfort (see [11]).

Braking and Steering monitoring system [6] (ρ_2): This supervision strategy was introduced by the authors in [6]. Since the vehicle stability is directly related to the sideslip motion of the vehicle, judging the vehicle stability region is derived from the phase-plane ($\beta - \dot{\beta}$) as follows:

$$\chi < 1, \tag{4}$$

where $\chi = \left|2.49\dot{\beta} + 9.55\beta\right|$ is the "Stability Index". Therefore, when the vehicle states move beyond the control boundaries and enter the "unstable region", braking actuators will be involved to generate an additive corrective yaw moment, pulling the vehicle back into the stable region. As mentioned in [13], one of the significant benefits of this stability index is that the reference region defined in (4) is largely independent of the road surface conditions and hence, the accurate estimation of the road surface coefficient of friction is not required.

Hence, a scheduling parameter $\rho_2(\chi)$ is used to activate, hierarchically, either the steering or braking, to ensure the vehicle stability and prevent dangerous driving situations (for more details, see [6]) .

B. Yaw control design of the braking/steering actuators

The LPV braking/steering control developed in [6], [8] is briefly presented below.

The generalized plant described in Fig 2 is used for the synthesis of the gain scheduled controller $K(\rho_2)$. This synthesis is done, based on a bicycle model (a lateral linear model of the vehicle [5]).

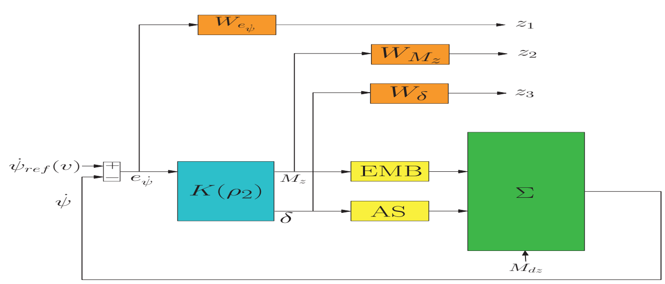

Fig. 2. Generalized plant model.

- Σ, EMB and AS stand for the extended bicycle, electro-mechanical braking and active steering actuators models, respectively.
- $W_{e_{\dot{\psi}}}$, $W_{M_z}(\rho_2)$, W_δ are weighting functions that allow to represent the desired tracking performances and the actuators limitations.

$$
\begin{cases}
\ddot{z}_{us_{ij}} &= (F_{sz_{ij}} - F_{tz_{ij}})/m_{us_{ij}} \\
\ddot{\theta} &= \big((F_{sz_{rl}} - F_{sz_{rr}})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f + mh\dot{v}_y\big)/I_x \\
\ddot{\phi} &= \big(F_{sz_f} l_f - F_{sz_r} l_r - mh\dot{v}_x\big)/I_y \\
\ddot{\psi} &= \big(l_f(-F_{tx_f}\sin(\delta) + F_{ty_f}\cos(\delta)) - l_r F_{ty_r} + (F_{tx_{fr}} - F_{tx_{fl}})t_f\cos(\delta) - (F_{tx_{rr}} - F_{tx_{rl}})t_r + M_{dz}\big)/I_z \\
\dot{\omega}_{ij} &= (R_{ij}F_{tx_{ij}} - T^f_{b_{ij}})/I_w
\end{cases}
\tag{5}
$$

IV. A NEW LPV SUSPENSION CONTROL METHOD FOR PERFORMANCE ADAPTATION TO THE ROLL INDEX

A. Control-structure model

In this study, a 7 DOF vehicle model is considered, see (5). For the control design purposes, linear models are assumed for the stiffness k_{ij} and damping c_{ij} in the suspension force computation.

In this section, the suspension control with performance adaptation (see [14]), to be integrated in the global VDC strategy (Vehicle Dynamic Control), is presented. The following H_∞ control scheme is considered, including parameter varying weighting functions.

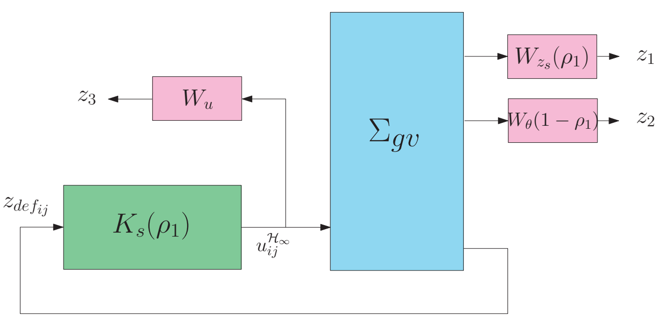

Fig. 3. Suspension system generalized plant.

where $W_{z_s} = \rho_1 \frac{s^2+2\xi_{11}\Omega_{11}s+\Omega_{11}{}^2}{s^2+2\xi_{12}\Omega_{12}s+\Omega_{12}{}^2}$ is shaped in order to reduce the bounce amplification of the suspended mass (z_s) between $[0, 12]$Hz.
$W_\theta = (1-\rho_1)\frac{s^2+2\xi_{21}\Omega_{21}s+\Omega_{21}{}^2}{s^2+2\xi_{22}\Omega_{22}s+\Omega_{22}{}^2}$ attenuates the roll bounce amplification in low frequencies.
$W_u = 3.10^{-2}$ shapes the control signal.

Remark 1: The parameters of these weighting functions are obtained using genetic algorithm optimization as in [15].

According to Fig. 3, the following parameter dependent suspension generalized plant ($\Sigma_{gv}(\rho_1)$) is obtained:

$$
\Sigma_{gv}(\rho_1) := \begin{cases} \dot{\xi} = A(\rho_1)\xi + B_1\tilde{w} + B_2 u \\ \tilde{z} = C_1(\rho_1)\xi + D_{11}\tilde{w} + D_{12}u \\ y = C_2\xi + D_{21}\tilde{w} + D_{22}u \end{cases} \tag{6}
$$

where $\xi = [\chi_{vert} \quad \chi_w]^T$; $\tilde{z} = [z_1 \quad z_2 \quad z_3]^T$; $\tilde{w} = [z_{rij} \quad F_{dx,y,z} \quad M_{dx,y}]^T$; $y = z_{def_{ij}}$; $u = u_{ij}^{\mathcal{H}_\infty}$; and χ_w are the vertical weighting functions states.

One of the main interesting contributions is the use of the parameter ρ_1 to schedule the distribution of the left & right suspensions on the four corners of the vehicle and tune the suspension dampers smoothly, thanks to the LPV frame work, from "soft" to "hard" to improve the car performances according to the driving situation. This distribution is handled using a specific structure of the suspension controller, given as follows :

$$
K_s(\rho_1) := \begin{cases} \dot{x}_c(t) = A_c(\rho_1)x_c(t) + B_c(\rho_1)y(t) \\[2ex] \begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \underbrace{U(\rho_1)C_c^0(\rho_1)}_{C_c(\rho_1)} x_c(t) \end{cases} \tag{7}
$$

where $x_c(t)$ is the controller state, $A_c(\rho_1)$, $B_c(\rho_1)$ and $C_c(\rho_1)$ controller scheduled by ρ_1. $u^{\mathcal{H}_\infty}(t) = [u_{fl}^{\mathcal{H}_\infty}(t)u_{fr}^{\mathcal{H}_\infty}(t)u_{rl}^{\mathcal{H}_\infty}(t)u_{rr}^{\mathcal{H}_\infty}(t)]$ the input control of the suspension actuators and $y(t) = z_{def}(t)$.

In this synthesis, the authors wish to stress that an interesting innovation is the use of a partly fixed structure controller, combined with a parameter dependency on the control output matrix introduced to allow a smooth load transfer distribution, depending on the situation. Then, the LPV framework is obtained, thanks to the matrix $U(\rho_1)$,

$$
U(\rho_1) = \begin{pmatrix} 1-\rho_1 & 0 & 0 & 0 \\ 0 & \rho_1 & 0 & 0 \\ 0 & 0 & 1-\rho_1 & 0 \\ 0 & 0 & 0 & \rho_1 \end{pmatrix} \tag{8}
$$

The parameter $\rho_1 \in [0; 1]$ is used to distribute the load at the four corners of the vehicle, ensuring a good tuning of the suspensions. When a load transfer is performed from the right to the left side, $\rho_1 \to 1$, and the suspensions actuators are set to be "hard" and tuned to provide more force to handle the big load transfer (left $\leftrightarrows$ right). The suspensions control at each corner aims at handling the overweight, by providing the accurate suspension force to ensure better stability and handling for the vehicle. Conversely, when the load transfer is carried out on the right side, $\rho_1 \to 0$, this control allows to considerably reduce the roll motion of the vehicle when running, the suspensions actuators are then tuned to "soft", and aim at enhancing the passengers comfort.

The LPV system (6) includes a single scheduling parameter and can be described as a polytopic system, i.e, a convex combination of the systems defined at each vertex of a polytope defined by the bounds of the varying parameter. The synthesis of the controller is made under the framework of the $\mathcal{H}_\infty$ control of polytopic suspensions, (for more details, see [16]).

Remark 2: All controllers presented along the paper are synthesized in the LPV/H_∞ framework. This design is achieved, thanks to the LMI-based H_∞ resolution.

V. Simulation Results

In this section, the model used for simulations is a full vehicle non linear model, see Section II-A. This model was validated by experimental tests on a real car (Renault Mégane Coupé). The following results are those obtained by the controllers previously described in Section III-B.

To test the efficiency of the proposed LPV/H_∞ GCC, the following scenario is used:

1) the vehicle runs at 100km/h in straight line on wet road ($\mu = 0.5$, where μ is a coefficient representing the adherence to the road).
2) a 5cm bump on the left wheels (from $t = 0.5$s to $t = 1$s),
3) a line change manoeuvre is performed by the driver,
4) lateral wind occurs at vehicle's front, generating an undesirable yaw moment (from $t = 2$s to $t = 2.5$s),
5) a 5cm bump on the left wheels ($t = 2$s), another on the left wheels,($t = 2.5$s),

For sake of completeness, the new LPV/H_∞ GCC strategy is compared to the uncontrolled vehicle, to show the vehicle stability enhancement and the improvements of the different dynamics.

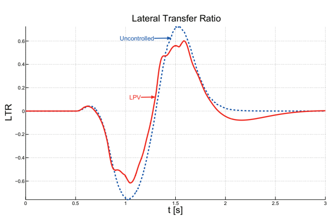

Fig. 4. ρ_1: load transfer index.

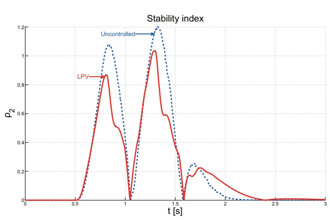

Fig. 5. ρ_2: stability index.

Fig 4 and 5 show the scheduling parameters ρ_1 and ρ_2. The proposed LPV strategy enhances very well the car stability since the load transfer bounce is reduced and stability index is attenuated.

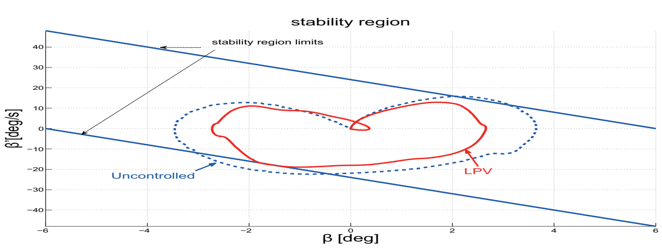

Fig. 6. Evolution of the vehicle in the β-$\dot{\psi}$ plane.

Fig. 6 shows one of the main results of this paper, namely, the vehicle stability enhancement. The evolution of the vehicle in the β-$\dot{\beta}$ plane, clearly demonstrates that the LPV strategy prevents the car from going beyond the stability region limits. This proves the efficiency of the control designed to reach the performance objectives.

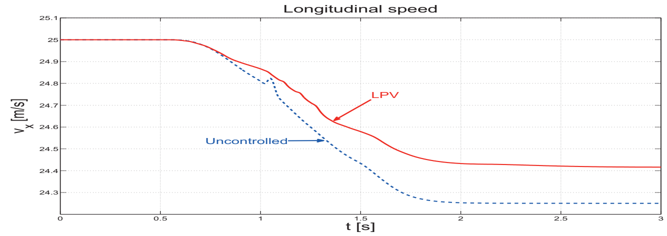

Fig. 7. Evolution of the vehicle longitudinal speed v_x.

In Fig. 7, the longitudinal speed of the car decreases more smoothly with the proposed LPV control when performing the driving scenario. This property gives more stability for the vehicle while running, compared to the uncontrolled case (in blue).

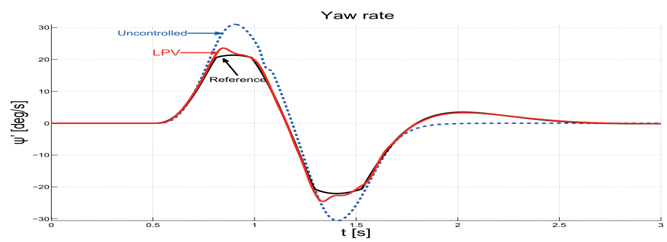

Fig. 8. Yaw rate.

Fig. 8 shows the yaw rate tracking, one of the main lateral car dynamics. It is clear that the designed LPV/H_∞ control strategy (in red) significantly enhances the vehicle stability since the results given by this approach fit well with the reference car performances (in black). This comparison allows to emphasize the good results in terms of vehicle lateral stability and dynamics improvement brought by the proposed approach.

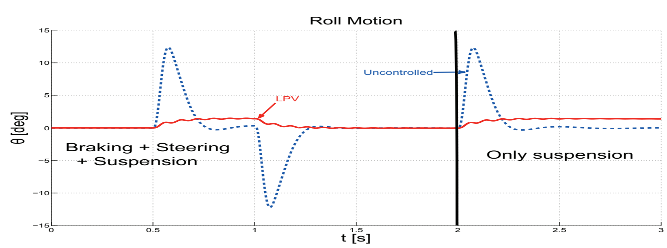

Fig. 9. Roll motion.

Fig 9 represents one of the main results of the paper since it shows the improvement brought by the new load transfer distribution control strategy. It is clear that the use of the new scheduling parameter ρ_1 reduces the roll motion of this vehicle. This leads to a better vehicle stability and enhances driving safety when performing critical scenarios.

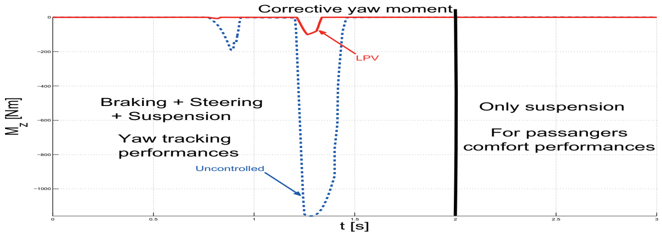

Fig. 10. Corrective yaw moment.

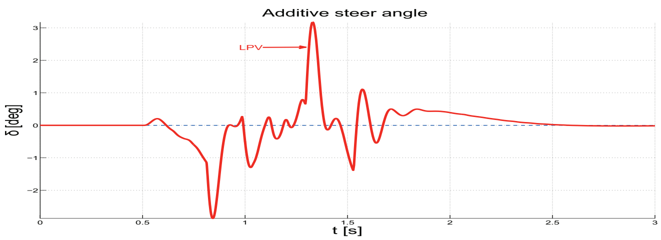

Fig. 11. Additive steer angle δ^+.

Fig 10 and 11 show the braking moment and additive steering angle respectively.
One of the important results shown in the previous figures is that the proposed suspension system control can take part in the coordination strategy between braking, steering and suspension actuators when necessary (depending on the driving situations) to preserve vehicle stability, or work as an independent control system using load transfer distribution (thanks to the parameter ρ_1) when only the vertical dynamics are concerned without any braking and steering actions.

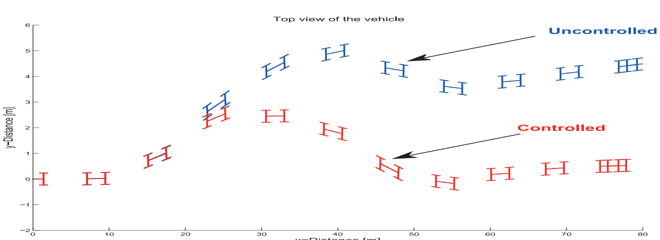

Fig. 12. Top view of the controlled and uncontrolled vehicle.

Fig. 12 summarizes the vehicle stability and performance improvement. It is clear, through the trajectory tracking, that the vehicle controlled by the proposed LPV/H_∞ (in red) behaves better, in terms of car dynamics and stability, than the uncontrolled one does.

VI. Conclusion

This paper has presented a new LPV/H_∞ global chassis control strategy involving several actuators, namely, suspensions, electro-mechanical rear braking and active steering ones. It introduced an innovative solution to the problem of vehicle dynamical stability and performances improvement, based on roll vehicle motion attenuation and load transfer distribution control and monitoring.
Simulations of a consistent representative driving situation, performed on a complex nonlinear model, have shown the efficiency of the proposed approach. The results based on those simulations show also the improvements given by this LPV GCC design method. Then, the stability of the vehicle has been considerably enhanced using this new control strategy while improving the different car dynamics.

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