A centralized multilayer LPV/$\mathcal{H}_\infty$ control architecture for vehicle’s global chassis control, and comparison with a decentralized architecture

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Abstract: This paper deals with the development of Global Chassis Controller where the Active Front steering, Direct Yaw Control and Active Suspensions, are coordinated together in the aim to improve the overall vehicle performance i.e. maneuverability, lateral stability and rollover avoidance. The main contribution of this work is the integration of the Active suspension system (AS) in a centralized multilayer control architecture to control the roll angle. A polytopic approach is used to find the LPV/$\mathcal{H}_\infty$ controller where an offline Linear Matrix Inequality (LMI) optimal solver is used to realize the optimality of this controller. The different layers of this architecture are detailed. The proposed LPV/$\mathcal{H}_\infty$ controller is validated by simulation using Matlab/Simulink, and a comparison is done with a decentralized architecture that has been developed in the laboratory, to show the difference in behavior and performance of both strategies of control and the effectiveness of centralized one on the rollover avoidance.

Keywords: Centralized Multilayer Control; LPV/$\mathcal{H}_\infty$; Global Chassis Control; Active Suspensions; Direct Yaw Control; Active Front Steering; decentralized control;

1. INTRODUCTION

Active safety is an important issue considered in intelligent vehicles. According to the “National Highway Traffic Safety Administration (NHTSA)” statistics, human’s faults cause almost 90% of road accidents as explained in Rajamani (2012). Advanced Driving Assistance Systems (ADAS) have been developed for several years in order to enhance the stability of road vehicles and to help the driver in maintaining the control of the vehicle under dangerous situations. ADAS systems are formed by several single-actuator approaches that have been proposed and marketed, such as: Electronic Stability Program (ESP) or Direct Yaw Control (DYC) to enhance the vehicle lateral stability; Active Front Steering (AFS) to mainly improve the vehicle maneuverability or lane keeping; and (Semi-) Active Suspensions (AS) to improve comfort, road holding and rollover avoidance [Chokor et al. (2019)].

Many advanced studies are developed in literature to improve the global performance of the vehicle in different driving situations. These studies suggest coordination between several ADAS systems known as Global Chassis Control (GCC). The GCC system deals with the complexity of control problems for Multi-Input-Multi-Output (MIMO) systems. The main objective of the GCC is the coordination between several actuators to improve the vehicle global behavior in terms of maneuverability and lateral stability depending on the driving situation. Many advanced control approaches have been proposed for this issue. The authors in He et al. (2006) applied a decentralized approach where they developed an AFS controller for maneuverability purpose and a DYC controller for lateral stability, based on sliding mode technique, and then a monitor coordinates between the two controllers according to the driving situations. However, the overall stability of the system cannot be guaranteed in the decentralized approach, especially when the two controllers can be actuated concurrently. In Doumiati et al. (2013), Poussot-Vassal et al. (2009), the authors propose several robust and optimal centralized controllers for the MIMO system based on the LPV/$\mathcal{H}_\infty$ control technique, where the LPV/$\mathcal{H}_\infty$ controller promotes or reduces the steering and braking to enhance maneuverability and lateral stability. With this approach, the overall stability of the system is guaranteed and a polytopic approach is used to actuate the different controllers. However, these controllers were synthesized while disregarding the roll motion; the deduced rollover enhancement was a consequence of the lateral stability control. Authors in Chen et al. (2016), Sename et al. (2013) have presented several centralized LPV/$\mathcal{H}_\infty$ controllers, where AFS, DYC and AS are used to control the lateral and vertical vehicle dynamics. From the other side, authors in Yao et al. (2017), Vu et al. (2017) and Mirzaei and Mirzaeinjed (2017), have used the roll angle and roll rate to control the vehicle load transfer that leads to rollover avoidance. Furthermore, authors deduced lateral stability improvement as a consequence of roll control.

All these interesting research have motivated us to study the control of the vehicle yaw rate, the side slip angle and the roll angle in order to improve the overall vehicle performance. Thus, in our present work, a new centralized multilayer control structure is developed to improve the maneuverability, lateral stability, and rollover avoidance using steering, braking actuators and active suspension system. The global centralized multilayer control architecture is shown in Fig. 1, and is developed later. The paper contributions are as follows:
merged together at the center of the front axle. Similarly, $F_{yr}$ is noted for the rear axle. $F_{xf}$ and $F_{yr}$ are given as:

$$F_{xf} = \mu_C \alpha_f,$$

$$F_{yr} = \mu_C \alpha_r,$$

and the tires slip angles as:

$$\alpha_f = -\beta - \frac{\psi_f}{V} + \delta_f,$$

$$\alpha_r = -\beta + \frac{\psi_r}{V}.$$  

By substituting (3) in (2), and then by substituting 2 in (1), the state space representation of the $Plant P$ can be represented in (4) (given next page), where $X = [\psi, \beta, \theta] \in \mathbb{R}^3$ is the state vector, $U = [\delta_f, M_x, M_y, M_z]^T$ is the vector of control inputs, $D = [M_d \psi, M_d \beta, M_d \theta]^T$ is the vector of exogenous inputs. Noting that the matrix $A \in \mathbb{R}^{3 \times 3}$ and the input matrices $B_0 \in \mathbb{R}^{3 \times 3}$ and $B_1 \in \mathbb{R}^{3 \times 3}$. In real time control, the output controlled variables $\psi$ and $\theta$ are given at the center of gravity (CG) of the vehicle by a gyrometer; $\theta$ is estimated by a simple time integration from $\dot{\theta}$ and could be directly delivered from the Inertial Measurement Unit (IMU) if available. The other states, side-slip angle $\beta$ and its velocity $\dot{\beta}$, could be calculated by an estimation. To do that, many observer approaches that deal with the real time implementation and vehicle dynamics have been presented in the literature, e.g. an observer based on Extended Kalman Filter EKF as proposed in Chen et al. (2016).

It should be noticed that the “bicycle model” used in the control layer of Fig. 1 is presented in Rajamani (2012) and is given in (5):

$$\left( \begin{array}{c} \psi_{ref} \\ \beta_{ref} \end{array} \right) = \left( \begin{array}{c} -\mu \frac{L_r^2 \psi_{ref} + L_r \psi_{ref}}{M_V^2} \\ -1 + \mu \frac{L_r^2 \psi_{ref} - L_r \psi_{ref}}{M_V^2} \end{array} \right) \left( \begin{array}{c} \psi_{ref} \\ \beta_{ref} \end{array} \right) + \left( \begin{array}{c} \mu \frac{L_r^2 \psi_{ref} + L_r \psi_{ref}}{M_V^2} \\ \mu \frac{L_r^2 \psi_{ref} - L_r \psi_{ref}}{M_V^2} \end{array} \right) \delta_d,$$

where $\delta_d$ is the driver steer angle on the front wheels, $\psi_{ref}$ is the desired reference yaw rate, $\beta_{ref}$ is the corresponding side slip angle, and $V_c$ is the vehicle longitudinal speed, considered as a varying parameter. For security reasons, the authors in Rajamani (2012) propose to saturate $\beta_{ref}$ and $\psi_{ref}$ below a threshold, as described in (6):

$$| \psi_{ref} | \leq 0.85 \mu_{g},$$

$$\beta_{ref} = \arctan(0.02 \mu_{g}).$$

Table 1. Parameters Values for Simulation

| Symbols | Description | Parameters values |
|---------|-------------|--------------------|
| $\psi$  | Vehicle yaw rate | $[rad/s]$ |
| $\beta$ | Vehicle side slip angle at CG | $[rad]$ |
| $\theta$ | Sprung mass roll angle | $[rad]$ |
| $F_0$  | Lateral forces at the i axle | $[N]$ |
| $\alpha_0$ | Driver steering angle | $[rad]$ |
| $V$    | Vehicle speed | $[m/s]$ |
| $I_1$  | Roll moment of inertia | $[kg m^2]$ |
| $I_2$  | Vehicle yaw moment of inertia | $[kg m^2]$ |
| $I_3$  | Vehicle yaw-roll product of inertia | $[kg m^2]$ |
| $t_f$  | Half front track | $[m]$ |
| $t_r$  | Half rear track | $[m]$ |
| $l_f$  | Wheelbase to the front | $[m]$ |
| $l_r$  | Wheelbase to the rear | $[m]$ |
| $h_t$  | Sprung mass roll arm | $[m]$ |
| $h_s$  | Sprung mass | $[kg]$ |
| $C_f$  | Front, rear tire cornering stiffness | $[N/\mu_{g}]$ |
| $K_0$  | Roll suspension angular stiffness | $[N \cdot m/\mu_{g}]$ |
| $C_p$  | Roll suspension angular damper | $[\mu_{g} \cdot m/s]$ |
| $g$    | Gravity constant | $[\mu_{g}]$ |
| $\mu$  | Road adherence coefficient | $[\mu_{g}]$ |
\[
\dot{X} = \begin{bmatrix}
\psi \\
\beta \\
\theta \\
\dot{\psi} \\
\dot{\beta} \\
\dot{\theta}
\end{bmatrix} = \begin{bmatrix}
\psi \\
\beta \\
\theta \\
\dot{\psi} \\
\dot{\beta} \\
\dot{\theta}
\end{bmatrix} + \begin{bmatrix}
\dot{b}_{u,11} & \dot{b}_{u,12} & \dot{b}_{u,13} & \dot{b}_{u,14} \\
\dot{b}_{u,21} & \dot{b}_{u,22} & \dot{b}_{u,23} & 0 \\
\dot{b}_{u,41} & \dot{b}_{u,42} & \dot{b}_{u,43} & 0
\end{bmatrix} \begin{bmatrix}
\delta_d + \delta_l \\
M_z \\
M_\theta \\
0 \\
0 \\
0
\end{bmatrix} + \begin{bmatrix}
\dot{b}_{d,11} & \dot{b}_{d,12} & \dot{b}_{d,13} & \dot{b}_{d,14} \\
\dot{b}_{d,22} & 0 & 0 & 0 \\
0 & 0 & 0 & 0
\end{bmatrix} \begin{bmatrix}
M_d, \psi \\
F_d, \psi \\
M_d, \theta
\end{bmatrix};
\]

(4)

\[
y = X.
\]

3. GLOBAL CHASSIS CONTROL

3.1 Centralized Control Architecture:

In this subsection, we present a detailed synthesis of the centralized multi-layer global chassis control architecture of Fig. 1, based on the LPV/H∞ theory. In the control layer, the output variables i.e the vehicle yaw rate \(\psi\), the side-slip angle \(\beta\), and the suspended mass roll \(\theta\) are controlled and optimized together through an optimal MIMO LPV/H∞ centralized controller, in order to enhance the vehicle maneuverability, the lateral stability and the rollover avoidance. \(\theta\) is an exogenous output. The desired states \(\psi_{ref}\) and \(\beta_{ref}\) are given in (5). In addition, \(\rho_1\) and \(\rho_2\) are two time-varying scheduling gains/patterns to organize the objectives of the MIMO LPV/H∞ controller. A decision layer is developed to control the situation of the vehicle; it sends the values of scheduling parameters, based on two criteria: lateral stability (SI) and load transfer ratio (LTR), discussed later. The actuator layer contains AFS, DYC and AS; it generates the physical inputs to vehicle system.

i) Control layer:

The architecture of the control layer is presented in Fig. 2. It contains the controller \(K_{LPV/H∞}(\rho_1, \rho_2)\) to be developed, the generalized plant \(\Sigma^G\), where \(\rho_1\) and \(\rho_2\) are two endogenous weighted parameters determined by the decision-making monitor according to the driving situations and the LTI model (Plant P) used in the synthesis of this controller. The inputs of \(K_{LPV/H∞}(\rho_1, \rho_2)\) controller are the errors between the desired states and the actual ones of the yaw rate \(\psi\), the side-slip angle \(\beta\), and the roll angle \(\theta\). Noting that the actual yaw rate \(\dot{\psi}\), side slip angle \(\dot{\beta}\), and roll angle \(\dot{\theta}\) are the outputs variables to be controlled and they are delivered from the LTI model (Plant P). The inputs of the Plant P are the AFS angle \(\delta_c\), the moment \(M_z\) around z axis (generated by the DYC) and the moment \(M_\theta\) around the roll axis (generated by the AS); \(M_{d,\psi}, F_{d,\psi}\), and \(M_{d,\theta}\) are the disturbances related to the road status and weather conditions like wind (exogenous inputs). \(W_\psi(\rho_1), W_\beta(\rho_1), W_\theta(\rho_2), W_\delta(\rho_1), W_M(\rho_1)\) and \(W_M(\rho_2)\) are the weighting functions determined in order to define the performance objectives \(Z_1, Z_2\), and \(Z_3\) and the actuator limitations \(Z_4, Z_5\) and \(Z_6\). The general form of these functions (see [Doumiati et al. (2014)]) depends on the simulated vehicle and integrated actuators:

- \(W_\psi(\rho_1)\) weights the yaw rate signal:

\[
W_\psi(\rho_1) = \frac{\rho_1 s}{M_1 + 2\pi f_1}
\]

where \(M_1\) is the margin of robustness, and \(T_1\) is the tolerated tracking error on \(\psi\). \(W_\beta(\rho_1)\) is designed similarly to \(W_\psi(\rho_1)\). \(W_\beta(\rho_1)\) is inversely dependent on the varying parameter \(\rho_1\). This is because the lateral stability is more penalized than maneuverability in critical situations. This issue is detailed later in the decision layer.

- \(W_\theta(\rho_2)\) weights the roll angle according to a scheduling parameter \(\rho_2\):

\[
W_\theta(\rho_2) = \frac{s}{M_2 + 2\pi f_2}
\]

\(M_2, T_2\) and \(f_2\) have similar definitions as \(M_1, T_1\) and \(f_1\). \(W_\theta(\rho_2)\) is linearly parametrized by the varying parameter \(\rho_2\), where \(\rho_2 \in \{\rho_2 \leq \rho \leq \rho_2^T\}\) (with \(\rho_2\) and \(\rho_2^T\) are constants representing the lower and higher values of \(\rho_2\)). When \(\rho_2 = \rho_2^T\), the performance objective \(e_\theta\) is penalized and the rollover avoidance becomes a priority objective to be realized.

- \(W_\delta(\rho_1)\) weights the steering control signal, \(\delta_c\):

\[
W_\delta(\rho_1) = \frac{1}{\rho_1} G_\delta^0 \left( \frac{s}{\pi f_4 + 1} \right) \left( \frac{\pi f_5 + 1}{\alpha^2 \pi^2 f_5 + 1} \right)
\]

\[
G_\delta^0 = \left( \frac{\Delta_\psi}{\alpha^2 \pi f_5 + 1} \right)^2
\]

\[
\Delta_\psi = 2\pi \left( f_4 + f_5 \right),
\]

where \(f_4\) and \(f_5\) are the lower and upper limits of the filter bandwidth \([f_4, f_5]\). This filter is proposed to force the active steering system to act in this range \([f_4, f_5]\). Otherwise, this filter ensures the activation of steering system below the cut of frequency \(f_5\) and above the driver ones \(f_4\) (see [Doumiati et al. (2014)])]. Note that steering system is promoted to enhance the maneuverability when \(\rho_1 = \rho_2^T\) and vice versa.

- \(W_M(\rho_1)\) weights the braking, \(M_z\):
$W_{M}(ρ_1) = ρ_1 \cdot 10^{-6} \frac{s}{(2π f_6)} + 1 \frac{s}{(κ2π f_6)} + 1$, \hspace{1cm} (11)

where $f_6$ is the braking actuator cut-off frequency and $κ$ to treat the braking actuator limitations (see [Doumiati et al. (2013)]). When $ρ_1 = \overline{ρ}_1$, the braking input is penalized, on the contrary, the braking control signal is relaxed when $ρ_1 = ρ_1$. This design depends on the vehicle lateral stability. \ - $W_{M}(ρ_2)$ weights the suspensions $M_δ$:

$W_{M}(ρ_2) = \left(1 \frac{1}{ρ_2} \right) \cdot 10^{-4} \frac{s}{(2π f_j)} + 1 \frac{s}{(κ2π f_6)} + 1$, \hspace{1cm} (12)

where $f_j$ is the cut-off frequency and $κ$ determines the limitation of suspension system. When $ρ_2 = \overline{ρ}_2$, the suspension system is penalized, however, the suspension system is relaxed when $ρ_2 = \overline{ρ}_2$ (case of rollover risk is detected). Note that we didn’t consider the actuators’ constraints in this control layer. However, the control inputs are filtered by using the weights above and the actuators’ models in the actuator layer. In addition, the tuning of the gains $ρ_1$ and $ρ_2$ respects the time response of the actuators.

After determining the subsystems of Fig. 2, $H_{∞}$ control technique is applied in order to minimize the controlled outputs $Z_1, Z_2, Z_3, Z_4, Z_5$ and $Z_6$ in presence of disturbances $M_{d,f}, F_{d,y}, M_δ$ and exogenous inputs $ψ_{d,les}, θ_{des}$. More information about the optimal $LPV/H_{∞}$ theory is presented in Sename et al. (2013) and Gu et al. (2005).

A “systic” Matlab function (Robust Control Toolbox) is used to make the interconnection between $Σ_{k}$ subsystems. The generalized plant $Σ_k$ is $LPV$ [Apkarian et al. (1995)], given as following as:

$Σ_k(ρ): [\begin{array}{c} \dot{x} \\ z \\ y \\ x \\ w \\ u \end{array}] = [\begin{array}{cccccc} A(ρ) & B_1(ρ) & B_2(ρ) & 0 \\ C_1(ρ) & D_{11}(ρ) & D_{12}(ρ) & 0 \\ C_2 & C_2 & D_{21} & 0 \\ 0 & 0 & D_{21} & 0 \end{array}] [\begin{array}{c} x \\ w \\ u \end{array}]$, \hspace{1cm} (13)

where $ρ = \{ρ_1, ρ_2\}$, $x$ is the vector of states variables of Plant $P$ and of the weighting functions, $u = [δ, M_δ, M_θ]^T$ represents the control inputs, $w = [ψ_{d,les}, θ_{des}, M_{d,f}, F_{d,y}, M_δ]^T$ is the exogenous input vector, $y = [w, β, θ]^T$ is the measurement vector fed-back to the controller, $ψ_{d,les} = [θ]^T$ is the exogenous output, and $z = [Z_1, Z_2, Z_3, Z_4, Z_5, Z_6]^T$ is the controlled output vector. Note that the matrices $B_2$ and $D_{21}$ depend on $ρ$, which is not consistent with the requirements of $H_{∞}$ synthesis for polytopic systems. Some filters on the control input have been used to solve this problem [Apkarian and Gahinet (1995)].

**Problem resolution: $LMIs$ based $LPV/H_{∞}$**

The problem of $LPV/H_{∞}$ is to find the controller $K_{LPV/H_{∞}}(ρ_1, ρ_2)$, scheduled by the parameters $ρ_1$ and $ρ_2$, such that:

$K_{LPV/H_{∞}}(ρ) : [\begin{array}{c} \dot{x} \\ u \end{array}] = [\begin{array}{c} A(ρ) & B_1(ρ) \\ C_1(ρ) & 0 \end{array}] [\begin{array}{c} x \\ y \end{array}]$, \hspace{1cm} (14)

This controller aims to minimize the $H_{∞}$ norm of the closed-loop $LPV$ system established by the equations (13) and (14).

Several approaches exist in the literature to solve this problem such as: polytopic, gridding and Linear Fractional Transformation LFT [Zin (2005)]. In our work, a polytopic approach [see Scherer et al. (1997)] has been used for controller synthesis. Applying the Bounded Real Lemma (BRL) extended to $LPV$ systems and after a change of basis presented in [Scherer et al. (1997)], a non conservative $LMIs$ is formulated in (15) and a Semi-Definite Program (SDP) has been applied to solve these inequalities equations (see [Doumiati et al. (2013)]), while minimizing $γ$ for $ρ = \{ρ_1, ρ_2\}X(ρ_2, ρ_2)$. The aim of polytopic approach is to find the $A(ρ), B(ρ)$ and $C(ρ)$ by using a common Lyapunov function i.e $X(ρ) > 0$ and $Y(ρ) > 0$ at each vertex of the polytope function of $ρ ∈ Ω$. Noting that the number of vertex is $4 (2^n)$ where $n$ is the number of exogenous parameters. Thus, the solution is given by the resolution of system (16) at each vertex of the convex hull $Ω$:

$\begin{cases}
C_i(ρ) = \tilde{C}(ρ)M_i(ρ)^{-T}
B_i(ρ) = N(ρ)^{-1}B(ρ)
A_i(ρ) = N(ρ)^{-1}(A(ρ) - Y(ρ)A(ρ)X(ρ) - N(ρ)B_i(ρ))
\end{cases}$, \hspace{1cm} (16)

where $M(ρ)N(ρ)^T = I - X(ρ)Y(ρ)$ with $M(ρ)$ and $N(ρ)$ are given by the user. More details about the computation solution have been presented in [Scherer et al. (1997)]. Therefore, referring to the polytopic approach, the final controller $K_{LPV/H_{∞}}(ρ_1, ρ_2)$ is the summation of each convex controller calculated on each vertex of polytope [Apkarian et al. (1995)]:

$K_{LPV/H_{∞}}(ρ_1, ρ_2) = α_1K_{H_{∞}}(α_1) + α_2K_{H_{∞}}(α_2)$

$α_1K_{H_{∞}}(α_1) + α_2K_{H_{∞}}(α_2)$

ii) Decision Layer:
The decision layer is dedicated to adjust the controller achievements according to the driver situations. This layer delivers the two endogenous parameters $ρ_1$ and $ρ_2$ based on two criteria, lateral stability ($SI$) and load transfer ratio ($LTR$). Before the determination of these parameters, let us introduce the definition of these important criteria in the decision of the driver situation.

-“Lateral Stability Index” $SI$:
The lateral stability index $SI$ determines the orientation of the vehicle depending on the speed vector at the CG, and its rate of change. $SI$ is given as (see [Chen et al. (2016)]):

$SI = c_1β + c_2β$, \hspace{1cm} (19)

where $c_1$ and $c_2$ are estimated w.r.t the vehicle parameters and the shape of the road. $SI$ varies between 0 and 1. An analysis of the driver situation is done depending on $SI$. For example, when $SI < SI$ the vehicle is in normal driving situations. Therefore, the $\overline{FB}$ is activated to enhance the maneuverability and the lateral stability in the moderate range of $SI$, and the DYC is penalized. In the contrary, when $SI$ increases progressively and becomes closer to $SI$ limit ($SI ≥ SI$), a lateral stability enhancement is needed and the DYC is activated. Referring to this analysis, the scheduled gain $ρ_1$ is designed to provide the $LPV/H_{∞}$ controller, the necessary information about the weights to be pushed or attenuated. The relation between $ρ_1$ and $SI$ is given through a “$sigmoid$” function (20) (see Fig. 4.a) that guarantees a continuous and smooth variation of $ρ_1$.

$ρ_1 = \overline{ρ}_1 \overline{ρ}_1 - \frac{P_I - P_I}{1 + e^{-\frac{SI - \overline{SI}}{\overline{SI} - \overline{SI}}}}$, \hspace{1cm} (20)
\[
\begin{align*}
A(\rho)X + XA(\rho)^T + B_2 \tilde{C}(\rho) + \tilde{C}(\rho)^T B_2^T \\
A(\rho)Y + A(\rho)^TY + B_2 \tilde{C}(\rho) + \tilde{C}(\rho)^T B_2^T \\
C_1(\rho)X + D_{12} \tilde{C}(\rho)
\end{align*}
\]

where \( X_{\text{des}} \) is the desired state of \( X \) with \( X_{\text{des}} = [x_{\text{des}}, \dot{x}_{\text{des}}]^T \in \mathbb{R}^2 \). The error vector is given by

\[
\begin{align*}
\begin{bmatrix}
\dot{x} \\
x
\end{bmatrix}
\end{align*}
\]

In the same way, the DYC moment \( M_c \) can be realized by applying a braking torque on the rear wheels of radius \( r \) [Doumiati et al. (2014)]. The applied braking torque is given as follows:

\[
\begin{align*}
\begin{bmatrix}
T_{bfr} \\
T_{btr}
\end{bmatrix}
\end{align*}
= \begin{cases} 
-2M_{sr}r, & \text{if } M_c \leq 0, \\
0, & \text{if } M_c > 0,
\end{cases}
\]

where \( T_{bfr} \) and \( T_{btr} \) are the left and right differential braking torque respectively. A simple model for the electro mechanical braking (EMB) actuator is used. The EMB actuator is modeled as:

\[
T_{b,r,j}^a = 2\pi f_b (T_{b,r,j} - T_{b,r,j}^a),
\]

where \( T_{b,r,j}^a \) tracks \( T_{b,r,j} \). \( f_b \) is the actuator cut-off frequency. This actuator control is bounded between \( [0, T_{a,\text{max}}^u] \), where \( T_{a,\text{max}}^u \) is the saturation of the EMB actuator.

Finally, the \( M_b \) moment is achievable by applying the active suspensions force-actuators on each wheel. These forces are given in the following form (see [Chokor et al. (2017)]):

\[
\begin{align*}
U_{j,l} &= \frac{0.5L_j}{L_j + T_{a,\text{max}}^u} M_j, \\
U_{j,r} &= -\frac{0.5L_j}{L_j + T_{a,\text{max}}^u} M_j, \\
U_{l} &= \frac{0.5L_j}{L_j + T_{a,\text{max}}^u} M_j, \\
U_{r} &= -\frac{0.5L_j}{L_j + T_{a,\text{max}}^u} M_j.
\end{align*}
\]

3.2 Decentralized Control Architecture:

The global decentralized multilayer control architecture of Fig. 5 is presented briefly in this subsection. This architecture has been developed in the frame of global chassis control in the laboratory and is used to compare the results between the centralized and decentralized architecture [Chokor et al. (2019)]. This decentralized sliding mode approach is chosen for the comparison purpose, because it represents the industrial state of the art and it is robust in case of system’s uncertainty, disturbance and possible sensor faults. The main difference between the two architectures is in the control layer. Thus, the output variables i.e. vehicle yaw rate \( \psi \), the side-slip angle \( \beta \), and the suspended mass roll \( \theta \) are controlled independently by using the single-input, single-output controller based on the Super-Twisting Sliding Mode (STSM) technique. Let us introduce an overview of the theory of Super-Twisting Sliding Mode. The STSM is a robust control technique that forces the states of the system to reach a sliding surface during a finite time (convergence phase) and to stay on this surface (sliding phase) in presence of perturbations.

Consider the second order system given as:

\[
\dot{x} = f(X,t) + g(X,t)u(t)
\]

where \( X = [x, \dot{x}]^T \in \mathbb{R}^2 \) is the state vector, \( u \) is the control input, and \( f, g \) are continuous functions. \( X_{\text{des}} \) is the desired state of \( X \) with \( X_{\text{des}} = [x_{\text{des}}, \dot{x}_{\text{des}}]^T \in \mathbb{R}^2 \). The error vector is given by

\[
\begin{align*}
\begin{bmatrix}
\dot{x} \\
x
\end{bmatrix}
\end{align*}
\]
where the sliding variables \( s \) are given by:

\[
\dot{s}(s,t) = \Phi(s,t) + \xi(s,t) u(t)
\]

where \( \Phi(s,t) \) and \( \xi(s,t) \) are the unknown bounded signals. The goal of the Super-Twisting algorithm is to enforce the sliding variable \( s \) to converge to zero (\( s = 0 \)) in finite time. Assume that there exist positive constants \( \delta_0, b_{\text{min}}, b_{\text{max}}, C_0, \) and \( U_{\text{max}} \) verifying for all \( x \in \mathbb{R}^n \) and \( |s(x,t)| < \delta_0 \):

\[
\begin{cases}
  |u(t)| \leq U_{\text{max}} \\
  |\Phi(s,t)| < C_0 \\
  0 < b_{\text{min}} \leq |\xi(s,t)| \leq b_{\text{max}}
\end{cases}
\]

Thus, the control input based on the Super-Twisting Sliding Mode algorithm, is given as:

\[
u(t) = u_1 + u_2 \begin{cases}
  u_1 = -\alpha_1 |s| \text{sign}(s), & \tau \in [0, 0.5] \\
  u_2 = -\alpha_2 \text{sign}(s)
\end{cases}
\]

\[\alpha_1 \text{ and } \alpha_2 \text{ are positive gains. The following conditions guarantee the finite time convergence:}
\]

\[
\begin{cases}
  \alpha_1 \geq \frac{3C_0(b_{\text{max}}\alpha_2+C_0)}{b_{\text{min}}(b_{\text{max}}\alpha_2-C_0)} \\
  \alpha_2 \geq \frac{C_0}{b_{\text{min}}}
\end{cases}
\]

The analysis of convergence is presented in Utkin (2013). An approximation function \( \frac{s}{\sqrt{s^2+\varepsilon}} \) is used to smooth the \( \text{sign}(s) \) function, where \( \varepsilon > 0 \).

Let us define the three sliding variables for the three decentralized controllers as follows:

\[
\begin{align*}
  s_y &= e_y = \psi - \psi_{\text{ref}}, \\
  s_b &= e_\beta = \beta - \beta_{\text{ref}}, \\
  s_\theta &= e_\theta = \theta - \theta_{\text{ref}} + k_0(\theta - \theta_{\text{ref}}),
\end{align*}
\]

The sliding variables \( s_y, s_b \) and \( s_\theta \) have a relative degree equal to one w.r.t \( \delta, M_c, \) and \( M_\beta \) respectively. Thus, in order to converge these variables to zero and the controlled states follow the desired ones, and based on the above discussion, the control inputs of AFS, DYC and AS applied to the system are given by:

\[
\begin{align*}
  \delta &= -\alpha_{s_1} |s_y|^2 \text{sign}(s_y) - \alpha_{s_2} \int_0^\tau \text{sign}(s_y) \, d\tau, \\
  M_c &= -\alpha_{M_c} |s_b|^2 \text{sign}(s_b) - \alpha_{M_c} \int_0^\tau \text{sign}(s_b) \, d\tau, \\
  M_\beta &= -\alpha_{M_\beta} |s_\theta|^2 \text{sign}(s_\theta) - \alpha_{M_\beta} \int_0^\tau \text{sign}(s_\theta) \, d\tau,
\end{align*}
\]

where \( \alpha_{s_i}, \alpha_{M_{c_i}}, \) and \( \alpha_{M_{\beta_i}} \), with \( i = [1, 2] \), are positive constants satisfying the conditions in (32). \( \tau_0, \tau_{M_c}, \) and \( \tau_{M_\beta} \) are constants between \([0, 0.5]\).

The decision layer is the same as before to monitor the driver situation based on S1 and LTR criteria, then it delivers the different gains \( \lambda \) in order to activate or deactivate the different actuators. These gains are given as follows (for more details see [Chokor et al. (2019)]):

\[
\lambda_S = \frac{1}{1+e^{-sT_{\text{LTR}}}}(\lambda_0 - \lambda_S),
\]

Concerning the actuator layer, it is the same as the one was developed in the centralized architecture.

4. SIMULATION RESULTS

In this section, the developed controller will be validated with a double lane change test at 110 km/h as initial speed. All simulations are done using Matlab/Simulink with a complete nonlinear model of the vehicle [Chokor et al. (2016)], validated on “SCANeR Studio” (OKtal) [Chokor et al. (2017)]. Then, a

\[\text{“SCANeR Studio” is a simulator dedicated to vehicle dynamics simulations.}\]
for a lateral stability improvement through the active suspensions system AS aims to avoid rollover by diminishing the roll angle, when the side-slip angle should be reduced as shown in Fig. 7. Both controllers have almost the same performance compared with the uncontrolled vehicle. Thus, the maneuverability objective is achieved. In order to improve the lateral stability and to prevent an undesirable driver situation, the side-slip angle should be reduced as shown in Fig. 7. Both controllers have similar influence on this angle. On the other hand, the convergence of roll angle to zero allows the avoidance of rollover risk, by reducing the load transfer ratio $L_T R$. The Fig. 8 shows that the $L P V/ \mathcal{H}_{\infty}$ controller is capable to diminish more the roll angle to zero compared to the STSM controller that is less performant w.r.t to $L P V/ \mathcal{H}_{\infty}$ controller. Noting that the choice and the tuning of parameters $\rho_1$ and $\rho_2$ (Fig. 11) is not obvious since $L P V/ \mathcal{H}_{\infty}$ controller aims to compromise between the different control objectives in order to give good results of optimality. For this reason, some oscillations appear in the Lateral stability index (Fig. 9) and the load transfer ratio (Fig. 10). Therefore, $\rho_1$ is chosen as $\rho_1 = \bar{\rho}_1$ when $S I \leq \bar{S} I$ in order to promote the maneuverability objective, while $\rho_2 = \rho_1$ for a lateral stability improvement through the activation of differential braking actuators. Similarly, $\rho_2$ is chosen as $\rho_2 = \bar{\rho}_2$ almost all the time, except 1 s and 3 s, where $L_T R$ is less than $\bar{L_T R} = 0.6$ and there is no risk of rollover. Around 1 s and 3 s, $\rho_2 = \bar{\rho}_2$ in order to increase the use of the active suspensions that diminish the roll angle, when the $L_T R$ becomes higher then the maximal threshold (see Fig.10).

Fig. 12 shows the driver steering angle $\delta_d$, the AFS steering angle of both controllers $\delta_c$, and the total steering angle applied to the vehicle $\delta_t$. The oscillations appear in $\delta_c$ with the $L P V/ \mathcal{H}_{\infty}$ because the controller forced the maneuverability objective. Fig. 13 shows the differential braking torque of rear wheels. The decentralized controller activates more the braking to ameliorate the lateral stability, while the $L P V/ \mathcal{H}_{\infty}$ saves energy. Fig. 14 and 15 show the AS control inputs of each controller. The vehicle speed is less dropped in the centralized approach since less braking is applied as can be seen from Fig. 16.

### Table 2. Controller Parameters for Simulation

| Parameters | Values |
|------------|--------|
| $f_1 = f_2$ | 11.15 Hz; 1 Hz; 10 Hz |
| $c_1, c_2$ | 9.55; 2.49; 2.5; 0.5:0.1 |
| $S I, S T, L_T R, L_T R\infty$ | 0.6:0.7; 0.6:0.7 |
| $\sigma_{\min}, \sigma_{\max}$ | $-5\div1200 N.m$ |

Fig. 9. Lateral stability comparison

Fig. 10. Load Transfer Ratio comparison

Fig. 11. Gains of $L P V/ \mathcal{H}_{\infty}$ controller

Fig. 12. Steering angle comparison

Fig. 13. Braking comparison

Fig. 14. AS control inputs STSM controller
To conclude, in this paper a centralized multilayer LPV/$H_\infty$ control architecture has been developed to improve the overall vehicle performance. A coordination of the Active Front Steering, Direct Yaw Control and Active Suspensions in one centralized controller has been proposed, to enhance the global behavior of the system. The proposed controller is validated in Matlab/Simulink and a comparison is done with another decentralized approach based on the Super-Twisting Sliding Mode (STSM) technique. Results confirm the importance of active suspensions in the centralized approach to prevent the rollover risk. In the Future, the LPV/$H_\infty$ controller will be extended, in order to realize more objectives concerning the vertical displacement, control of pitch-angle...with the introduction of artificial intelligence (AI)-based techniques to improve the decision layer and the tuning of gains, and to make the controller more robust and optimal. We will consider also the variation of the road adherence, and the generalization of centralized approach especially for tuning gains.

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