Introduction to vehicle dynamics control

Olivier Sename et Soheib Fergani

Gipsa-lab Grenoble and LAAS-CNRS Toulouse, France

July 2-7, 2017

Outline

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Road safety: an international stake

- Worldwide, 1.24 million people of road traffic deaths per year (+ 50 million of injuries) [a]. For people aged 5-29 years, road traffic injuries is the leading cause of death.
- Various causes: speed, alcohol, drugs, non safe driving,...
- Recognized importance of smart and safe cars: passive safety (airbags, belt..) and active (ABS, ESP....)

[a] World health organization 2013

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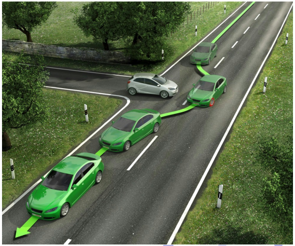
Among the 5 pillars towards road safety

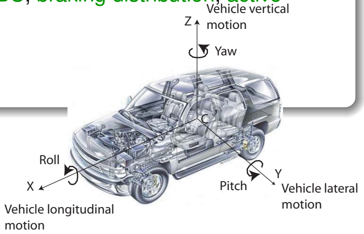
Safer vehicles: Electronic Stability Control is part of the minimum standards for vehicle construction (ex European and Latin New Car Assessment Programs - NCAP)

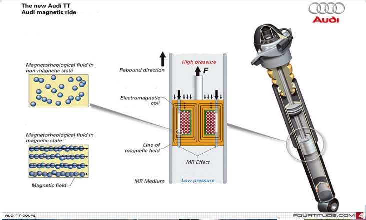


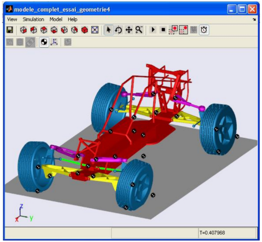
Challenges in chassis control

Today's vehicles. . .

- Growth of controlled organs: suspensions, ABS, ESC, ABC, braking distribution, active steering, tire pressure, TCS
- Increasing number of sensors & actuators
- Heavy networking







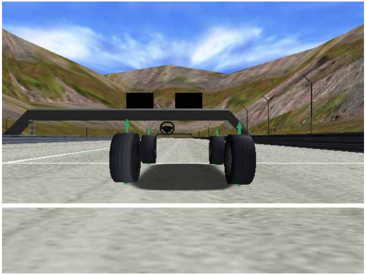
Challenges in chassis control

Complexity to synchronize all the controllers to improve

- Driving comfort (and pleasure)
- Active safety

Need for fault tolerance in case of actuator/sensor malfunctions

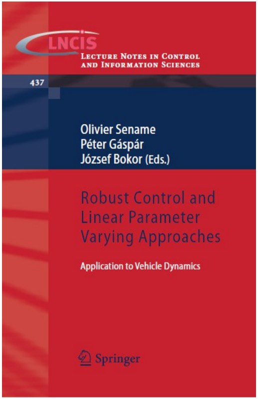
Ferrari VDC



Introduction

This course has been mainly written thanks to:

- the Post-doctoral work of [Moustapha Doumiati (2010)]
- the PhD dissertations of [Damien Sammier (2002), Alessandro Zin (2005), Poussot-Vassal(2008), Sébastien Aubouet (2010), Anh Lam DO (2011), Soheib Fergani (2014)].
- the authors' works since 1995
- interesting books cited below



Collaborations & associated studies

ANR INOVE 2010-2014

Modelling and control of a hydraulic semi-active damper- PhD thesis of Sébastien Aubouet 2010

Global chassis control using LPV/H_∞ control - PhD thesis Soheib Fergani 2014, Alessandro Zin 2005, Charles Poussot 2008

Magneto-rheological dampers - PhD thesis of Charles Poussot 2008, Sébastien Aubouet 2010

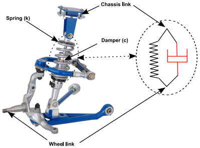
Modelling and control of semi-active suspensions - Post Doc Charles Poussot 09, PhD thesis of Ahn-Lam Do 2011

Skyhook and H_∞ control of semi-active suspensions - PhD thesis of Damien Sammier 2002

Suspension system

Objective

- Link between unsprung and sprung masses
- Involves vertical (z_s, z_{us}) dynamics



Suspension system

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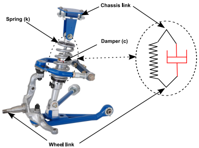
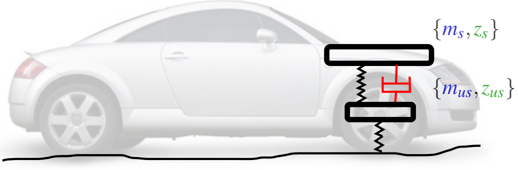
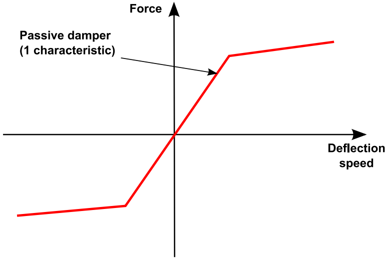


Passive suspension system





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Objective

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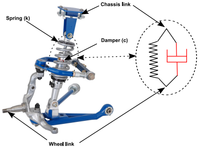
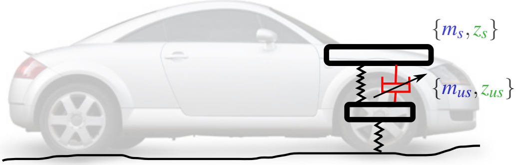
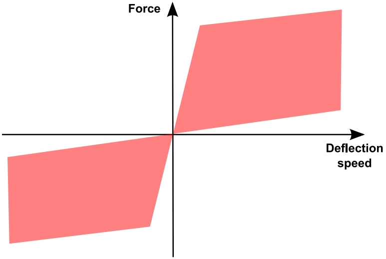


Semi-active suspension system $\longrightarrow$ dissipates energy through an adjustable damping coefficient





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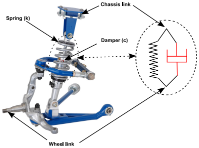
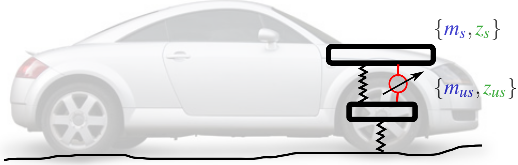
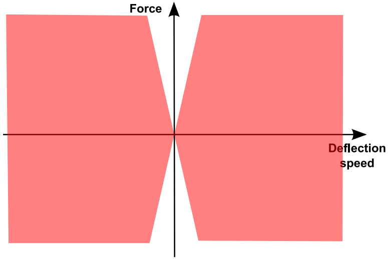


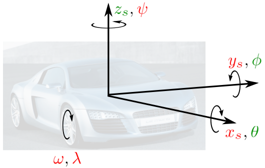
Active suspension system $\longrightarrow$ dissipate and generate energy working as an active actuator





Vehicle model - dynamical equations

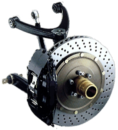
Full vertical model



- Mainly influenced by the vehicle suspension systems.
- Describes the comfort and the roadholding performances.

$$
\begin{aligned}
\ddot{z}_s &= -\left(F_{sz_{fl}} + F_{sz_{fr}} + F_{sz_{rl}} + F_{sz_{rr}}\right)/m_s \\
\ddot{z}_{us_{ij}} &= \left(F_{sz_{ij}} - F_{tz_{ij}}\right)/m_{us_{ij}} \\
\ddot{\theta} &= \left((F_{sz_{rl}} - F_{sz_{rr}})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f\right)/I_x \\
\ddot{\phi} &= \left((F_{sz_{rr}} + F_{sz_{rl}})l_r - (F_{sz_{fr}} + F_{sz_{fl}})l_f\right)/I_y
\end{aligned}
$$

Wheel & Braking system



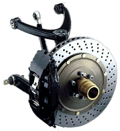
Objective

- Link between wheel and road
- Influences safety performances
- Involves longitudinal (v) rotational (ω) and slipping ($\lambda = \frac{v - R\omega}{\max(v, R\omega)}$) dynamics

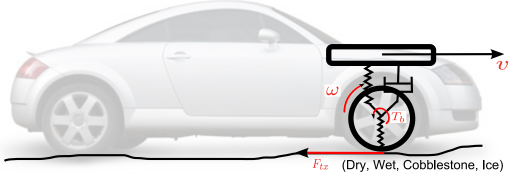
Wheel & Braking system

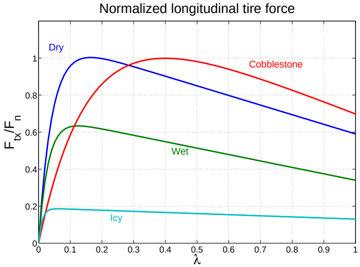
Objective

- Link between wheel and road (z_r, μ)
- Influences safety performances
- Involves longitudinal (v) rotational (ω) and slipping ($\lambda = \frac{v - R\omega}{\max(v, R\omega)}$) dynamics



Extended quarter vehicle model

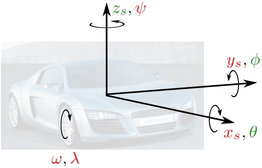




Vehicle model - dynamical equations

Full vertical model

- Mainly influenced by the vehicle suspension systems .
- Describes the comfort and the roadholding performances .

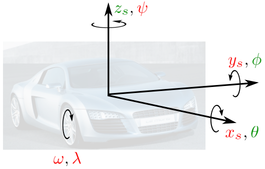


$$
\begin{aligned}
\ddot{z}_s &= -\left(F_{sz_{fl}} + F_{sz_{fr}} + F_{sz_{rl}} + F_{sz_{rr}}\right)/m_s \\
\ddot{z}_{us_{ij}} &= \left(F_{sz_{ij}} - F_{tz_{ij}}\right)/m_{us_{ij}} \\
\ddot{\theta} &= \left((F_{sz_{rl}} - F_{sz_{rr}})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f\right)/I_x \\
\ddot{\phi} &= \left((F_{sz_{rr}} + F_{sz_{rl}})l_r - (F_{sz_{fr}} + F_{sz_{fl}})l_f\right)/I_y
\end{aligned}
$$

Vehicle model - dynamical equations

Full vertical and longitudinal model

- Mainly influenced by the vehicle suspension systems and the braking system.
- Describes the comfort and the roadholding performances and the the stability and security issues.



$$
\begin{aligned}
\ddot{x}_s &= \left((F_{tx_{fr}} + F_{tx_{fl}}) + (F_{txrr} + F_{tx_{rl}})\right)/m \\
\ddot{z}_s &= -\left(F_{sz_{fl}} + F_{sz_{fr}} + F_{sz_{rl}} + F_{szrr}\right)/m_s \\
\ddot{z}_{us_{ij}} &= \left(F_{sz_{ij}} - F_{tz_{ij}}\right)/m_{us_{ij}} \\
\ddot{\theta} &= \left((F_{sz_{rl}} - F_{szrr})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f\right)/I_x \\
\ddot{\phi} &= \left((F_{szrr} + F_{sz_{rl}})l_r - (F_{sz_{fr}} + F_{sz_{fl}})l_f + mh\ddot{x}_s\right)/I_y \\
\lambda_{ij} &= \frac{v_{ij} - R_{ij}\omega_{ij}}{\max(v_{ij}, R_{ij}\omega_{ij})} \\
\dot{\omega}_{ij} &= \left(-RF_{tx_{ij}}(\mu, \lambda, F_n) + T_{b_{ij}}\right)/I_w
\end{aligned}
$$

Wheel & Steering system

Objective

- Wheel / road contact
- Influences safety performances
- Involves lateral (y_s), side slip angle (β) and yaw (ψ) dynamics

Wheel & Steering system

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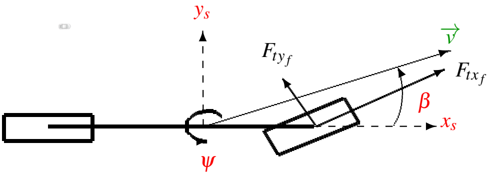
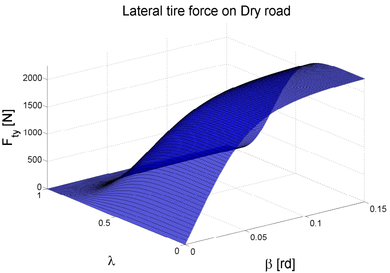
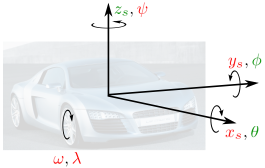
Bicycle model





Vehicle model - dynamical equations

Full vertical and longitudinal model



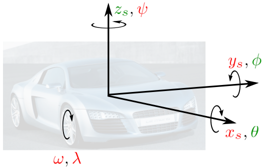
$$
\begin{aligned}
\ddot{x}_s &= \big((F_{tx_{fr}} + F_{tx_{fl}}) + (F_{txrr} + F_{tx_{rl}})\big)/m \\
\ddot{z}_s &= -\big(F_{sz_{fl}} + F_{sz_{fr}} + F_{sz_{rl}} + F_{szrr}\big)/m_s \\
\ddot{z}_{us_{ij}} &= \big(F_{sz_{ij}} - F_{tz_{ij}}\big)/m_{us_{ij}} \\
\ddot{\theta} &= \big((F_{sz_{rl}} - F_{szrr})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f\big)/I_x \\
\ddot{\phi} &= \big((F_{szrr} + F_{sz_{rl}})l_r - (F_{sz_{fr}} + F_{sz_{fl}})l_f + mh\ddot{x}_s\big)/I_y
\end{aligned}
$$

$$
\begin{aligned}
\lambda_{ij} &= \frac{v_{ij} - R_{ij}\omega_{ij}}{\max(v_{ij}, R_{ij}\omega_{ij})} \\
\dot{\omega}_{ij} &= (-RF_{tx_{ij}}(\mu, \lambda, F_n) + T_{b_{ij}})/I_w
\end{aligned}
$$

Vehicle model - dynamical equations

Full model

- A very complex model with dynamical correlations.
- Subject to several external disturbances.



$$
\begin{aligned}
\ddot{x}_s &= \big((F_{tx_{fr}} + F_{tx_{fl}})\cos(\delta) + (F_{txrr} + F_{tx_{rl}}) - (F_{ty_{fr}} + F_{ty_{fl}})\sin(\delta) + m\dot{\psi}\dot{y}_s\big)/m \\
\ddot{y}_s &= \big((F_{ty_{fr}} + F_{ty_{fl}})\cos(\delta) + (F_{tyrr} + F_{ty_{rl}}) + (F_{tx_{fr}} + F_{tx_{fl}})\sin(\delta) - m\dot{\psi}\dot{x}_s\big)/m \\
\ddot{z}_s &= -\big(F_{sz_{fl}} + F_{sz_{fr}} + F_{sz_{rl}} + F_{szrr}\big)/m_s \\
\ddot{z}_{us_{ij}} &= \big(F_{sz_{ij}} - F_{tz_{ij}}\big)/m_{us_{ij}} \\
\ddot{\theta} &= \big((F_{sz_{rl}} - F_{szrr})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f - mh\ddot{y}_s + (I_y - I_z)\dot{\psi}\dot{\phi}\big)/I_x \\
\ddot{\phi} &= \big((F_{szrr} + F_{sz_{rl}})l_r - (F_{sz_{fr}} + F_{sz_{fl}})l_f + mh\ddot{x}_s + (I_z - I_x)\dot{\psi}\dot{\theta}\big)/I_y \\
\ddot{\psi} &= \big((F_{ty_{fr}} + F_{ty_{fl}})l_f\cos(\delta) - (F_{tyrr} + F_{ty_{rl}})l_r + (F_{tx_{fr}} + F_{tx_{fl}})l_f\sin(\delta) \\
&\quad + (F_{txrr} - F_{tx_{rl}})t_r + (F_{tx_{fr}} - F_{tx_{fl}})t_f\cos(\delta) - (F_{tx_{fr}} - F_{tx_{fl}})t_f\sin(\delta) \\
&\quad + (I_x - I_y)\dot{\theta}\dot{\phi}\big)/I_z \\
\lambda_{ij} &= \frac{v_{ij} - R_{ij}\omega_{ij}\cos\beta_{ij}}{\max(v_{ij}, R_{ij}\omega_{ij}\cos\beta_{ij})} \\
\dot{\omega}_{ij} &= (-RF_{tx_{ij}}(\mu, \lambda, F_n) + T_{b_{ij}})/I_w \\
\beta_{ij} &= \arctan\left(\frac{\dot{x}_{ij}}{\dot{y}_{ij}}\right)
\end{aligned}
$$

Vehicle model - dynamical equations

Full model

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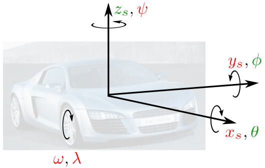
- Subject to several external disturbances.



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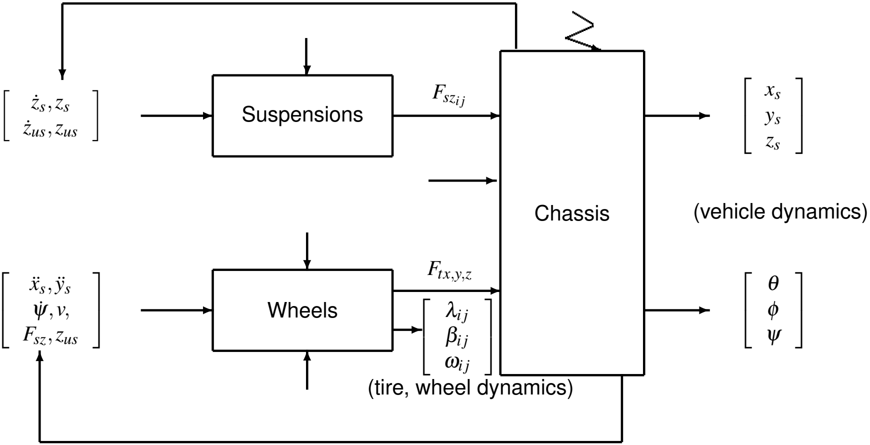
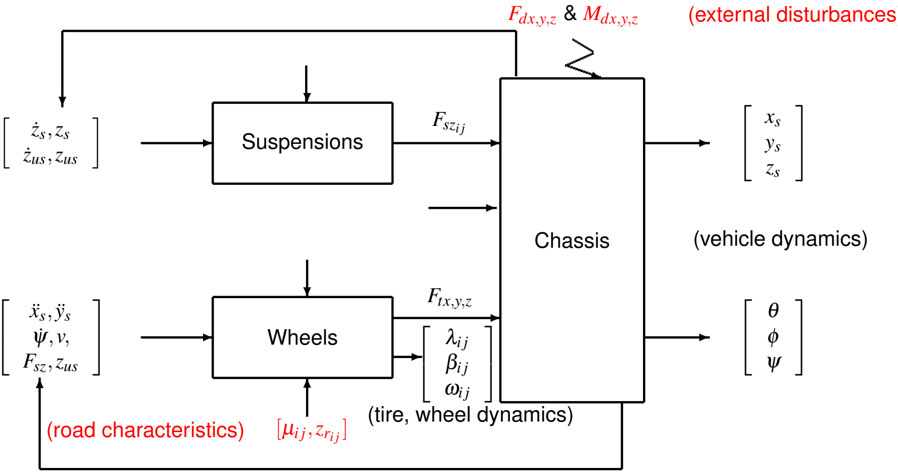
\end{aligned}
$$

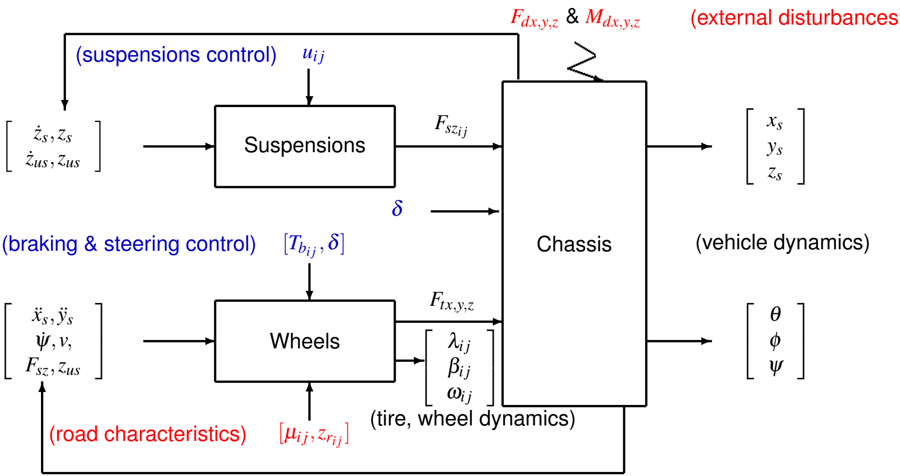
Vehicle model - synopsis



Vehicle model - synopsis



Vehicle model - synopsis



Towards global chassis control approaches (GCC)

Some facts

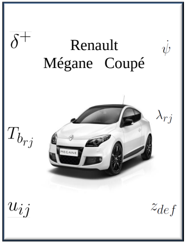
- In most vehicle control design approaches, the vehicle-dynamics control sub-systems (suspension control, steering control, stability control, traction control and, more recently, kinetic-energy management) are traditionally designed and implemented as independent (or weakly interleaved) systems.
- The global communication and collaboration between these systems are done with empirical rules and may lead to unappropriate or conflicting control objectives.
- So, it is important to develop new methodologies (centralized control strategies) that force the sub-systems to cooperate in some appropriate "optimal" way.

Global chassis control

- This approach combines several (at least 2) vehicle sub-systems in order to improve the general behavior of the vehicle ; in particular, the GCC methodology is developed to improve comfort and safety properties, according to the vehicle situation, taking into account the actuators constraints and the knowledge (if any) of the environment of the vehicle.
- The objective is then to make the sub-systems collaborate towards the same goals, according to the vehicle situation (constraints, environment, ...) in order to fully exploit the potential benefits coming from their interconnection.

Towards global chassis control approaches (GCC)

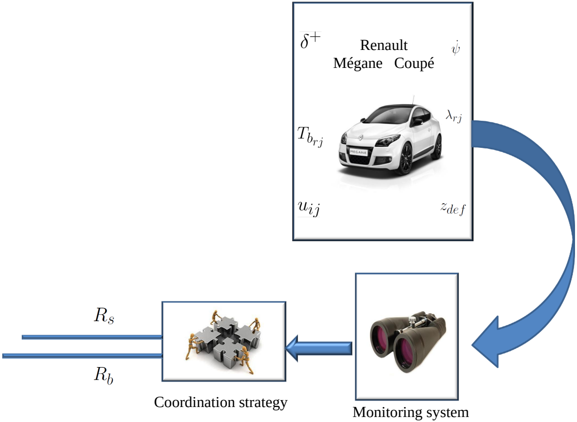
The GCC strategies are developed in 2 steps:



Towards global chassis control approaches (GCC)

The GCC strategies are developed in 2 steps:

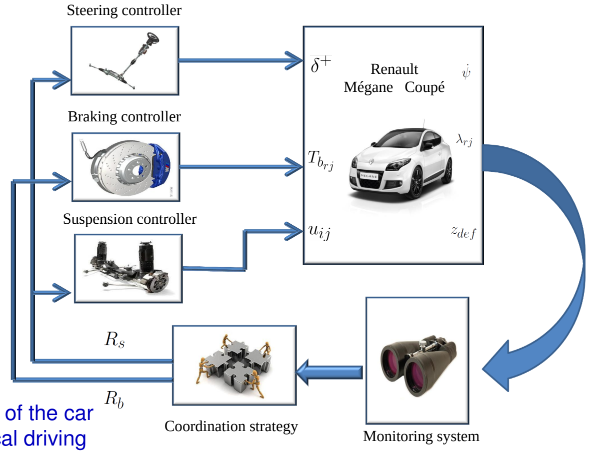
- The monitoring approach $\rightarrow$ collaborative based strategy.



Towards global chassis control approaches (GCC)

The GCC strategies are developed in 2 steps:

- The monitoring approach $\rightarrow$ collaborative based strategy.
- Developping coordinated control strategies $\rightarrow$ achieve close loop performance and actuators coordination.



Main objective:

- Improve the overall dynamics of the car and the vehicle safety in critical driving situations.

Towards Global chassis control approaches

Two main approaches, one considering the vehicle as a MIMO system, the other developing a "*super controller*" for the local actuators. Some references : Lu and DePoyster (2002), Shibahata (2005), Chou and d'Andréa Novel (2005), Andreasson and Bunte (2006), Falcone et al. (2007a), Falcone et al. (2007b), Gáspár et al. (2008), Fergani and Sename (2016)...

Vehicle considered as a MIMO system

- This approach consists in considering the vehicle as a global MIMO system and in designing a controller that solves all the dynamical problems by directly controlling the various actuators with the available measurements. No local controller is considered (no inner loop). See, for instance Lu and DePoyster (2002), Chou and d'Andréa Novel (2005), Andreasson and Bunte (2006), Gáspár et al. (2008), Fergani et al (2016).

High level reference super controller

- The second approach consists in designing a controller which aims at providing somehow, the reference signals to local controllers, which have been previously designed to solve a local subsystem problem (e.g. ABS). Thus, this controller, more than a controller, "monitors" the local controllers. Therefore, such a controller solves the global vehicle dynamical problems, playing the role of "*super controller*". See also Falcone et al. (2007a).

A MIMO case: Suspension and braking

Characteristic of the solution

Build a multivariable global chassis controller Shibahata (2004), (Poussot et al. 2011) :

- Improve comfort in normal cruise situations
- Improve safety in emergency situations (safety prevent comfort)
- Supervise actuators and resources
- The proposed design relies in the introduction of two parameters to handle the performance compromise, actuator efficiency and well-coordinated action.
- The suspension performance moves from comfort to road holding characteristics when the braking monitor identifies a normal or critical longitudinal slip ratio.
- Robust control theory approach (LPV/$\mathscr{H}_\infty$)
  $\Rightarrow$ MIMO internal stability & no switching

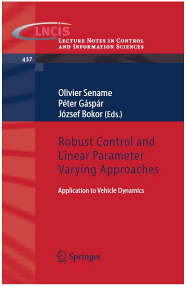
Towards Global chassis control approaches

Some examples

- braking/suspension : non linear approach (Chou & d'Andréa Novel), LPV for heavy vehicles (Gaspar, Szabo & Bokor), for cars (Poussot et al.)
- braking / steering : optimal control [Yang et al.], predictive [Di Cairano & Tseng, control allocation [Tjonnas & Johansen], or LPV [Doumiati et al, 2013]
- braking /suspension/ steering : [Fergani, Sename, Dugard]

LPV interest: on-line Adaption of the vehicle performances

- to various road conditions/types (measured, estimated)
- to the driver actions
- to the dangers identified thanks to some measurements of the vehicle dynamical behavior
- to actuators/sensors malfunctions or failures



Towards Global chassis control approaches

In this presentation, 3 examples are provided for the topic:

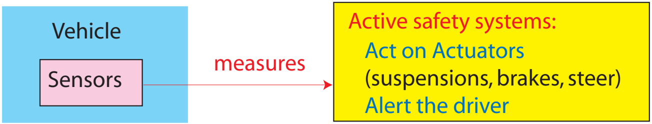
- ⋆ Active safety using coordinated steering/braking control.
- ⋆ Road profile estimation and road adaptive vehicle dynamics control.
- ⋆ LPV FTC for Vehicle Dynamics Control.

Outline

Vehicle safety systems:

- Prevent unintended behavior
- Help drivers maintaining the vehicle control
- Current production systems include:
  - Anti-lock Braking Systems (ABS): Prevent wheel lock during braking
  - Electronic Stability Control (ESC): Enhances lateral vehicle stability
    - Braking based technique
  - 4 Wheel steering (4WS): Enhances steerability
    - Adding additional steering angle

General structure:



Any vehicle control system needs accurate information about the vehicle dynamics, and the more accurate information it gets, the more it can perform

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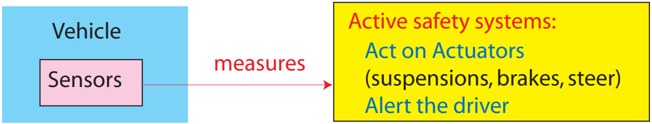

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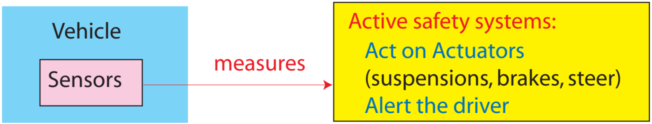

Any vehicle control system needs accurate information about the vehicle dynamics, and the more accurate information it gets, the more it can perform

The presentation of today focuses on:

- Yaw stability by active control
  - Prevents vehicle from skidding and spinning out
  - Improves of the turning (yaw) rate response
  - Improves lateral vehicle dynamics
  - Involves Braking and Steering actuators

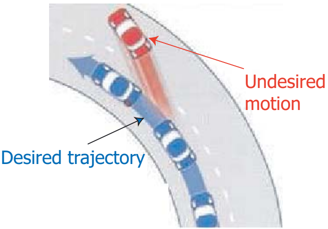

Figure: The objective is to restore the yaw rate as much as possible to the nominal motion expected by the driver

Problematic

Problem tackled: vehicle critical situations

- Lateral and yaw stability of ground vehicles & braking actuator limitations
- Widely treated in literature [Ackermann, Falcone, Villagra, Bunte, Chou, Canale] (mainly steering or braking, but a few use both)

Contributions

- Use Rear braking & Steering actuators to enhance vehicle stability properties
- Extension of [Poussot-Vassal et al., CDC2008 & ECC2009] results
- Propose a simple $\mathscr{H}_\infty$ tuning using [Bünte et al., IEEE TCST, 2004] results
- LPV Controller structure exploiting system properties to handle braking constraints
- Nonlinear frequency validations

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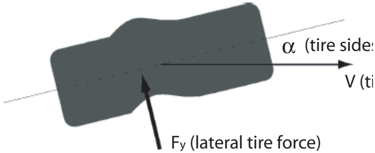
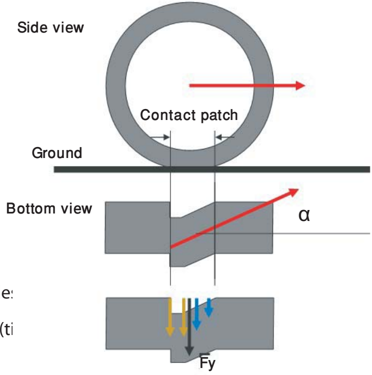
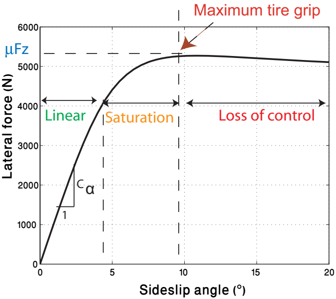
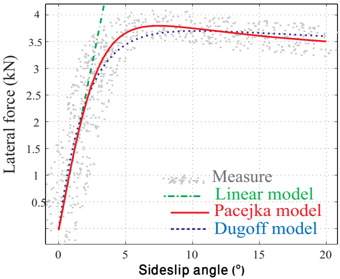
Lateral motion of a vehicle

- Motion of a vehicle is governed by tire forces
- Tire forces result from deformation in contact patch
- Lateral tire force, F_y, is function of:
  1. Tire slip (α)
  2. Vertical load applied on the tire (F_z)
  3. Friction coefficient (μ)



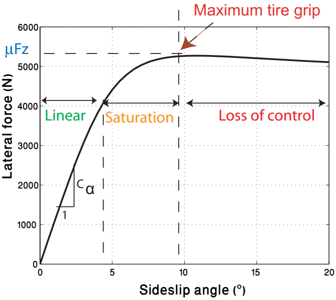


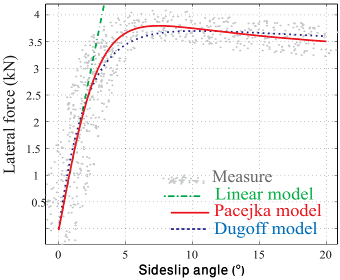




Vehicle response

- Normally, we operate in LINEAR region
  - Predictable vehicle response
- During slick road conditions, emergency maneuvers, or aggressive driving
  - Enter NONLINEAR tire region
  - Response unanticipated by driver





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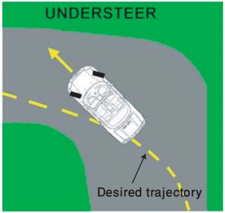
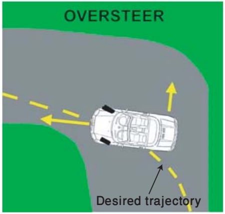
Why we lose the vehicle control?

Imagine making an aggressive turn...

- If front tires lose grip first, plow out of turn (limit understeer)
  - May go into oscillatory response
  - Driver loses ability to influence vehicle motion
- If rear tires saturate, rear end kicks out (limit oversteer)
  - May go into a unstable spin
  - Driver loses control
- Both can result in loss of control





Unstable motion due to nonlinear tire characteristics

Planar bicycle model (Dugoff et al.(1970))

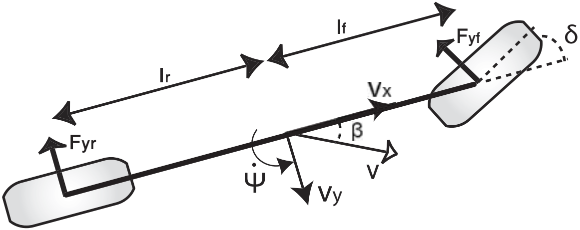
Main dynamics under interest, toward control scheme ...

- Equation of lateral motion:

$$mv\left(\dot{\beta} - \dot{\psi}\right) = Fy_f + Fy_r \tag{1}$$

- Equation of yaw motion:

$$I_z\ddot{\psi} = l_f Fy_f - l_r Fy_r, \tag{2}$$



Linear Synthesis model

The 2-DOF linear bicycle model described in Section 2 is used for the control synthesis. Although the bicycle model is relatively simple, it captures the important features of the lateral vehicle dynamics. Taking into account the controller structure and objectives, this model is extended to include:

- the direct yaw moment input M_z^*,
- a lateral disturbance force F_{dy} and a disturbance moment M_{dz}. F_{dy} affects directly the sideslip motion, while M_{dz} influences directly the yaw motion.

$$\begin{bmatrix} \ddot{\psi} \\ \dot{\beta} \end{bmatrix} = \begin{bmatrix} -\frac{l_f^2 C_f + l_r^2 C_r}{I_z v} & \frac{l_r C_r - l_f C_f}{I_z} \\ 1 + \frac{l_r C_r - l_f C_f}{mv2} & -\frac{C_f + C_r}{mv} \end{bmatrix} \begin{bmatrix} \dot{\psi} \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{l_f C_f}{I_z} \\ \frac{C_f}{mv} \end{bmatrix} \delta^* + \begin{bmatrix} \frac{1}{I_z} \\ 0 \end{bmatrix} M_z^* + \begin{bmatrix} \frac{1}{I_z} \\ \frac{1}{mv} \end{bmatrix} \begin{bmatrix} M_{dz} \\ F_{dy} \end{bmatrix} \quad (3)$$

Steering vs. Braking

Steering control: (Rajamani(2006), Guven et al.(2007))

- Adds steering angle to improve the lateral vehicle dynamics
- Regulates tire slip angles and thus, the lateral tire force
- Drawback:
  - Becomes less effective near saturation

DYC (Direct Yaw Control) - Braking control: (Park(2001), Boada et al.(2005))

- Regulates the tire longitudinal forces
- Maintains the vehicle stability in all driving situations
- Drawbacks:
  - Wears out the tires
  - Causes the vehicle speed to slow down against the driver demand

The idea is to design a controller that:

- Improves vehicle steerability and stability
  - Makes the yaw rate tacking the desired value (response of a bicycle model with linear tires)
  - Makes the slip angle small
- Coordinates Steering/braking control
  - Minimizes the influence of brake intervention on the longitudinal vehicle dynamics
- Rejects yaw moment disturbances

Methodology:

H_∞ synthesis extended to LPV system:

- H_∞ synthesis: frequency based performance criteria
- LPV: One type of a gain scheduled controller

See paper Doumiati et al (2013)

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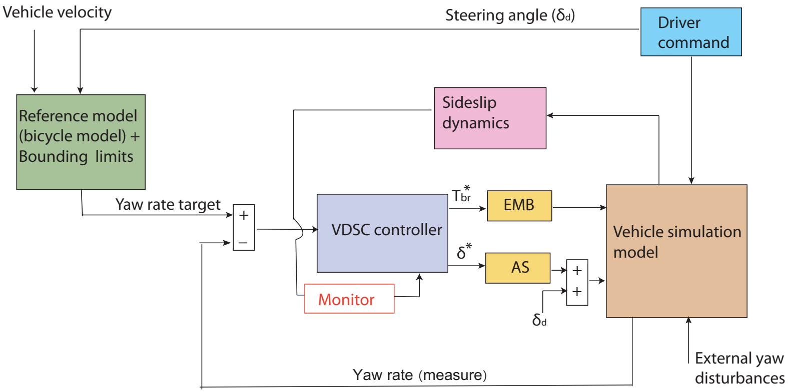
Overall control scheme diagram



AS: Steer-by-wire system
EMB: Brake-by-wire Electro Mechanical system

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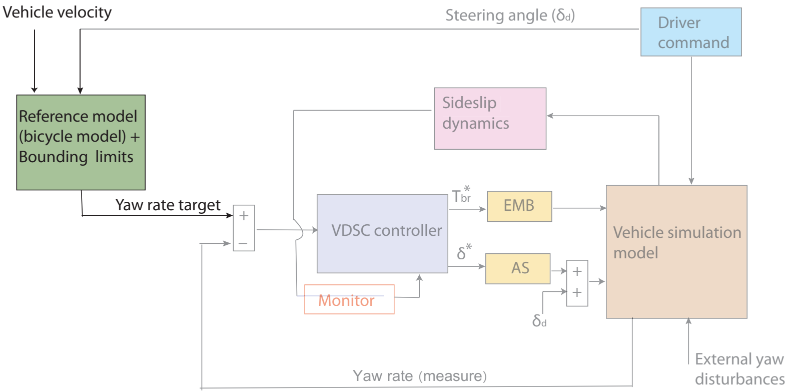

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Reference model

The basic idea is to assist the vehicle handling to be close to a linear vehicle handling characteristic that is familiar to the driver

- Bicycle linear model, $F_y = C_\alpha \alpha$ (low sideslip angle)
- $\dot{\psi} \leq \mu \times g / V_x$
  - Ensures small slip dynamics $(\beta, \dot{\beta})$
  - Attenuates the lateral acceleration

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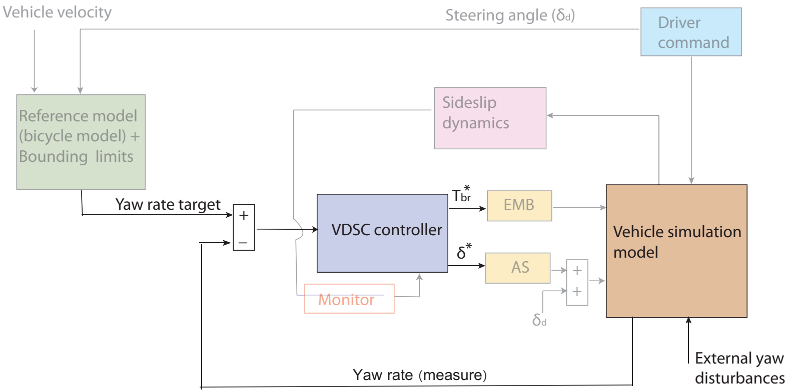
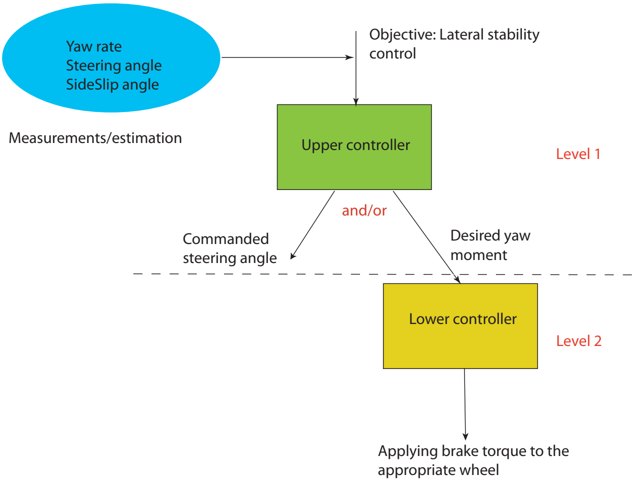
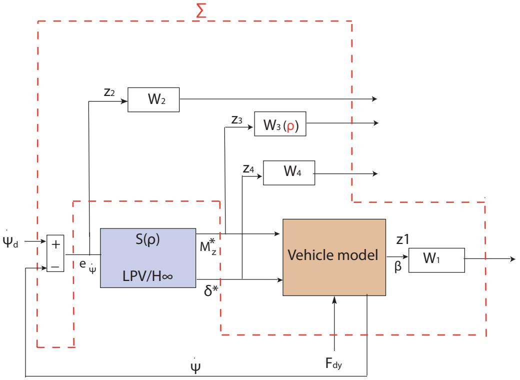


VDSC Controller architecture



VDSC-upper controller: LPV/H_∞ control



Generalized plant (tracking problem)

Vehicle model is LTI:

- Linear bicycle model
- Synthesized considering a dry road

ρ scheduling parameter:

- $\rho(t)$ is time dependent and known function
- ρ bounded: $\rho \in [\underline{\rho}, \overline{\rho}]$

$w(t)$	=	$[\dot{\psi}_d, F_{dy}]$	exogenous input
$u(t)$	=	$[\delta^*, M_z^*]$	control input
$y(t)$	=	$[e_{\dot{\psi}}]$	measurement
$z(t)$	=	$[W_1 z_1, W_2 z_2, W_3 z_3, W_4 z_4]$	controlled output

VDSC design (cont'd)

- z_1 : sideslip angle signal, β: $W_1 = 2$. $\rightsquigarrow$ to reduce the body sideslip angle
- z_2 yaw rate error signal: $W_2 = \frac{s/M+w_0}{s+w_0 A}$, where $M = 2$ for a good robustness margin, $A = 0.1$ so that the tracking error is less than 10%, and the required bandwidth $w_0 = 70\ rad/s$.
- z_3 braking control signal, M_z^* ,according to a scheduling parameter ρ:

$$W_3 = \rho \frac{s/(2\pi f_2)+1}{s/(\alpha 2\pi f_2)+1}, \quad (4)$$

where $f_2 = 10\ Hz$ is the braking actuator cut-off frequency and $\alpha = 100$. $\rho \in \left\{\underline{\rho} \leq \rho \leq \overline{\rho}\right\}$ (with $\underline{\rho} = 10^{-4}$ and $\overline{\rho} = 10^{-2}$).

- z_4, the steering control signal attenuation ($f_3 = 1Hz$, $f_4 = 10Hz$):

$$\begin{aligned} W_\delta &= G_\delta 0 \frac{(s/2\pi f_3+1)(s/2\pi f_4+1)}{(s/\alpha 2\pi f_4+1)2} \\ G_\delta 0 &= \frac{(\Delta_f/\alpha 2\pi f_4+1)2}{(\Delta_f/2\pi f_3+1)(\Delta_f/2\pi f_4+1)} \text{ and } \Delta_f = 2\pi(f_4+f_3)/2 \end{aligned} \quad (5)$$

This filter is designed is order to allow the steering system to act only in $[f_3, f_4]$Hz. At $\Delta_f/2$, the filter gain is unitary [Bunte et al. 2004, TCST].

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VDSC-upper controller: LPV/H_∞ control

Controller solution: LPV/H_∞

- Mixed-Sensitivity problem
- Minimizes the H_∞ norm from w to z
- $\gamma_\infty = 0.89$ (Yalmip/Sedumi solver)

$W_3(\rho)$:

- $\rho = 0.1 \rightarrow$ braking is ON
- $\rho = 10 \rightarrow$ braking is OFF

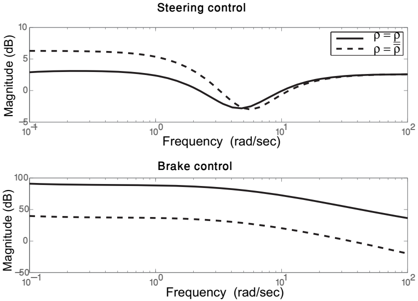

Figure: Bode diagrams of the controller outputs δ^* and M_z^*

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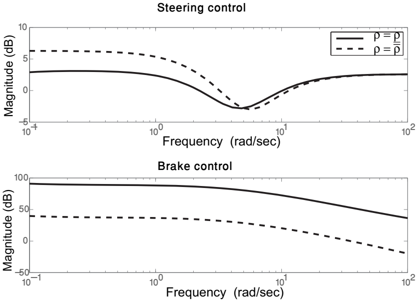

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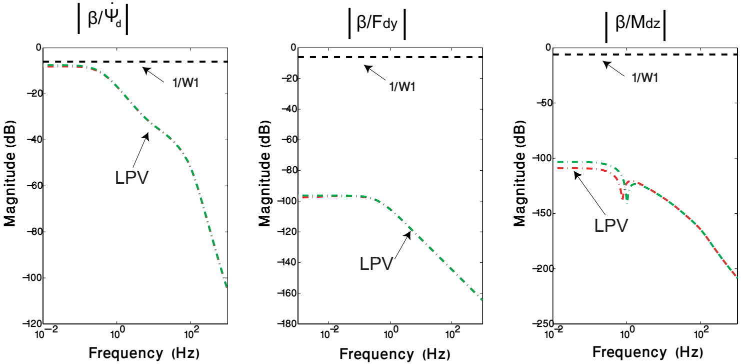


Figure: Closed loop transfer functions between β and exogenous inputs

- Attenuation of the side slip angle
- Rejection of the yaw disturbance

VDSC-upper controller: LPV/H_∞ control

Sensitivity functions:

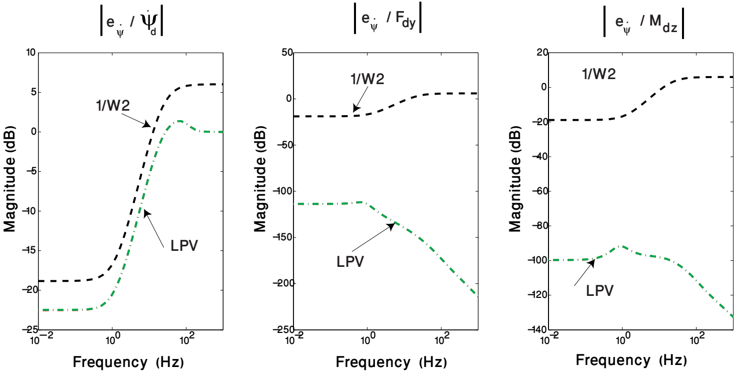


Figure: Closed loop transfer functions between $e_{\dot\psi}$ and exogenous inputs

- Attenuation of the yaw rate error
- Rejection of the yaw disturbance

VDSC-upper controller: LPV/H_∞ control design

Sensitivity functions:

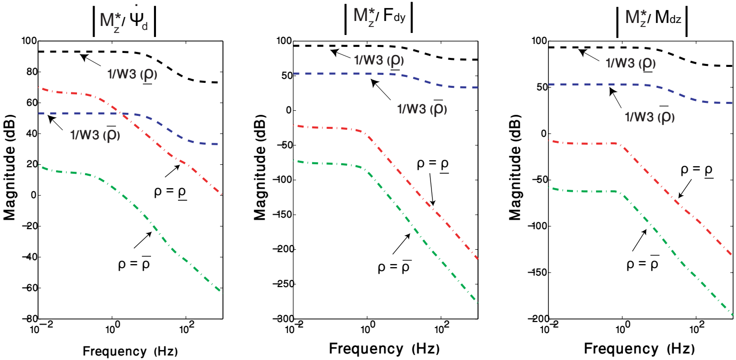


Figure: Closed loop transfer functions between M^* and exogenous inputs

$\rho = 0.1 \rightarrow$ braking is activated, $\rho = 10 \rightarrow$ braking is penalized

VDSC-upper controller: LPV/H_∞ control design

Sensitivity functions:

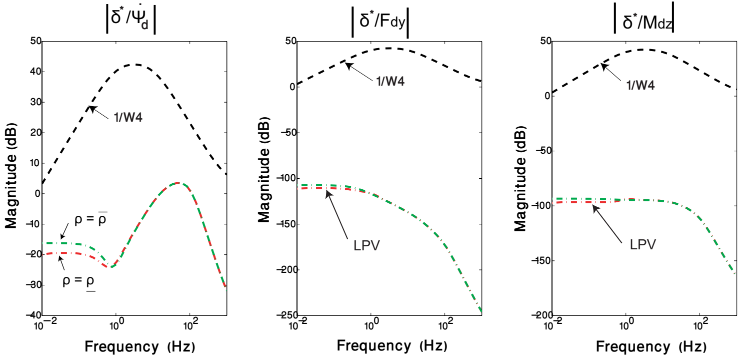
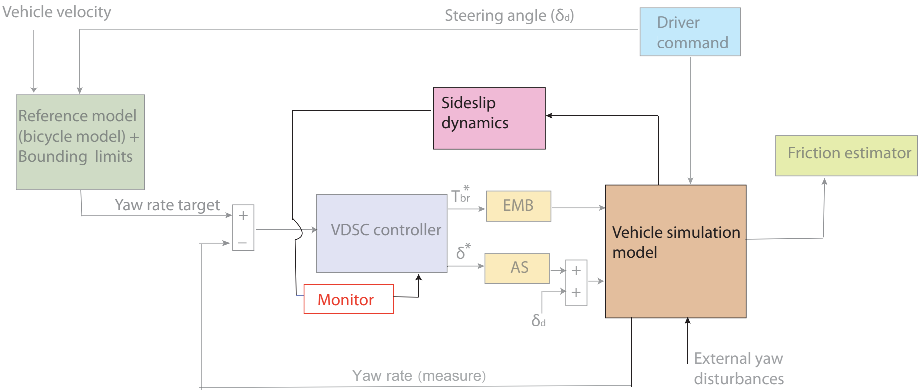


Figure: Closed loop transfer functions between δ^* and exogenous inputs

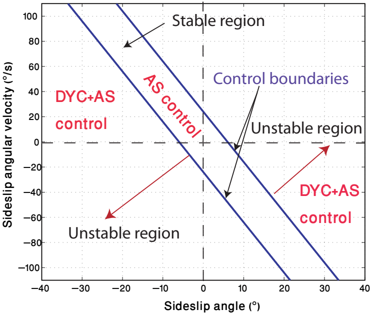
- Steering is activated on a specified range of frequency
- W_4: Activates steering in a frequency domain where the driver cannot act (*Guven et al.*(2007))

Overall control scheme diagram



Coordination between Steering and braking

- $\beta - \dot{\beta}$ phase plane is used as measure of the vehicle operating points
- Stability boundaries for controller design: $\chi = \left| \frac{1}{24}\dot{\beta} + \frac{4}{24}\beta \right| < 1$ (Yang et al.(2009), He et al.(2006))



This criterion, χ, allows accurate diagnosis of the vehicle stability.

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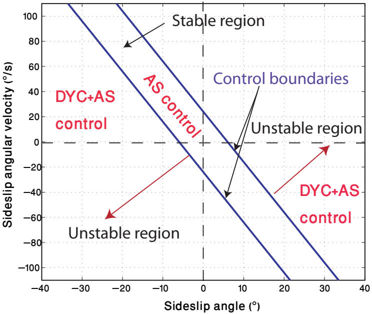
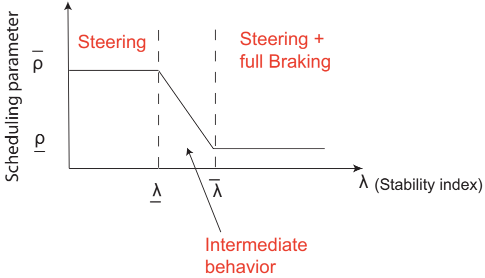
- $\beta - \dot{\beta}$ phase plane is used as measure of the vehicle operating points
- Stability boundaries for controller design: $\chi = \left| \frac{1}{24}\dot{\beta} + \frac{4}{24}\beta \right| < 1$ (Yang et al.(2009), He et al.(2006))



This criterion, χ, allows accurate diagnosis of the vehicle stability.

Monitor



$$\rho(\chi) := \begin{cases} \overline{\rho} & \text{if } \chi \leq \underline{\chi} \text{ (steering control - Steerability control task)} \\ \dfrac{\overline{\chi}-\chi}{\overline{\chi}-\underline{\chi}}\overline{\rho} + \dfrac{\chi-\underline{\chi}}{\overline{\chi}-\underline{\chi}}\underline{\rho} & \text{if } \underline{\chi} < \chi < \overline{\chi} \text{ (steering+braking)} \\ \underline{\rho} & \text{if } \chi \geq \overline{\chi} \text{ (steering+full braking - Stability control task)} \end{cases} \quad (6)$$

where $\underline{\chi} = 0.8$ and $\overline{\chi} = 1$ ($\underline{\chi}$ is user defined)

Sideslip angle estimation

Available measurements (from ESC or reasonable cost sensors):

- Yaw rate, $\dot{\psi}$
- Steering wheel angle, δ
- Wheel speeds, w_{ij}
- Lateral acceleration, a_y
- $\dot{\beta}$ can be evaluated through available sensors:

$$\dot{\beta} = \frac{a_y}{v_x} - \dot{\psi}, \tag{7}$$

β?? $\rightarrow$ Existing Methods:

- Integration of $\dot{\beta}$
- Kinematic equations (eq. a_y, a_x)
- Model-based observer (Vehicle model + Estimation technique)

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Sideslip angle estimation

This study:

- Planar bicycle model (with constant velocity):

$$\begin{cases} \dot{\beta} & = & (Fy_f + Fy_r)/(mv) + \dot{\psi} \\ \ddot{\psi} & = & \left[l_f Fy_f - l_r Fy_r + M_z^* \right] / I_z \end{cases} \qquad (8)$$

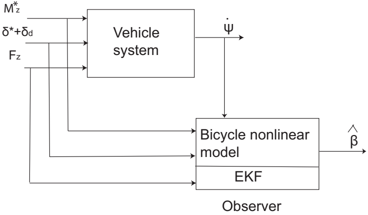
- Dugoff's tire model:

$$F_y = -C_\alpha \times tan(\alpha) \times f(\alpha, F_z, C_\alpha), \text{ where f(.) is nonlinear} \qquad (9)$$

- Nonlinear filtering: Extended Kalman Filter

State-space representation:

- $X = [\beta,\ \dot{\psi}]^T$
- $U = \left[M_z^*,\ \delta,\ F_z\right]^T$.
  - $\delta = \delta^* + \delta_d$.
- $Y = [\dot{\psi}]^T$



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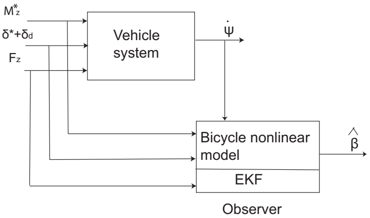
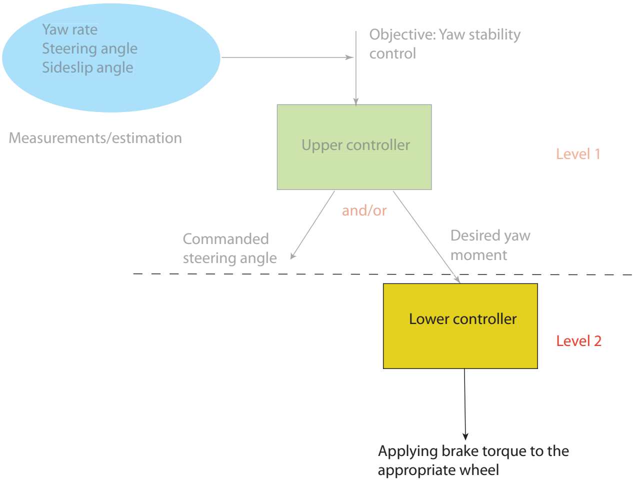


VDSC Controller architecture



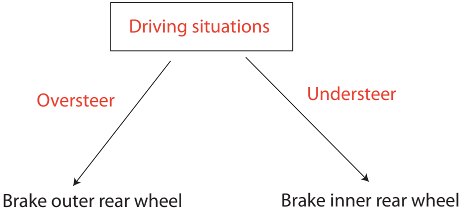
VDSC-lower controller algorithm

The stabilizing moment M_z^* provided by the controller is converted into braking torque and applied to the appropriate wheels

Rules

- Braking 1 wheel: from an optimal point of view, it is recommended to use only one wheel to generate the control moment (Park(2001))
- Only rear wheels are involved to avoid overlapping with the steering control

Decision rule:



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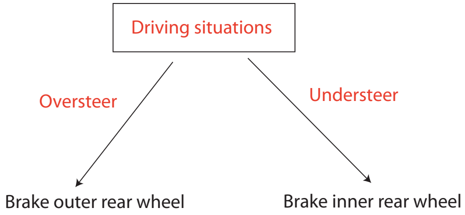

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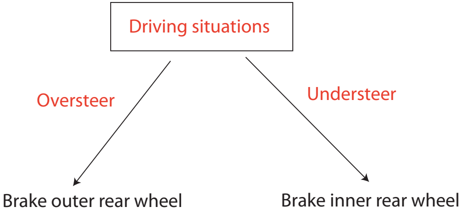

Simulation results and results

- Matlab/Simulink software
- Vehicle Automotive toolbox
  - Full nonlinear vehicle model
  - Validated in a real car "Renault Mégane Coupé"

Two tests:

1. Double-lane-change maneuver at $100\ km/h$ on a dry road ($\mu = 0.9$)
2. Steering maneuver at $80\ km/h$ on a slippery wet road ($\mu = 0.5$)

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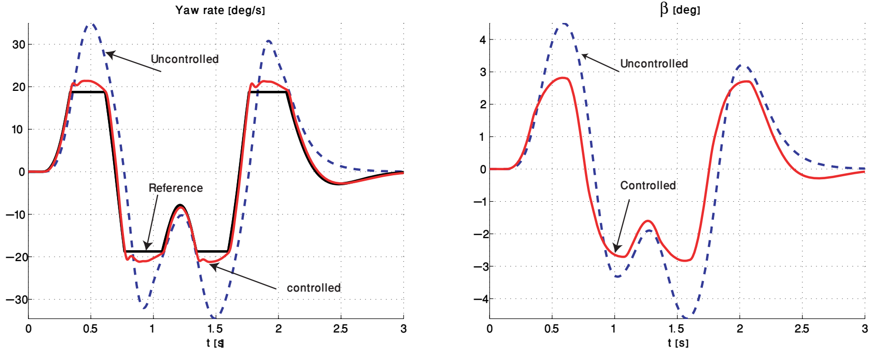
Test 1: Results [dry road $\mu = 0.9$, $V = 100\ km/h$]



Vehicle dynamic responses with and without controller

Test 1: Results [dry road $\mu = 0.9$, $V = 100\ km/h$]

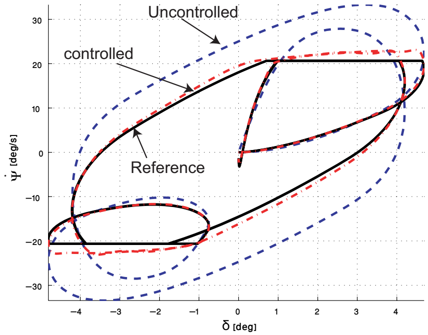


Figure: Response of the yaw rates versus steering wheel angle

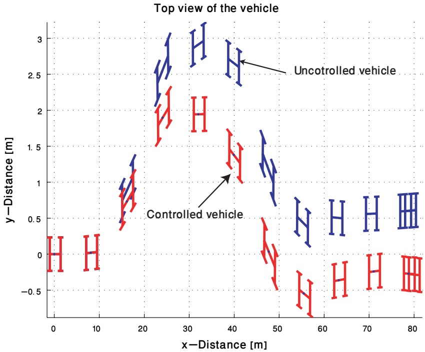


Figure: Trajectories of the controlled and uncontrolled vehicles

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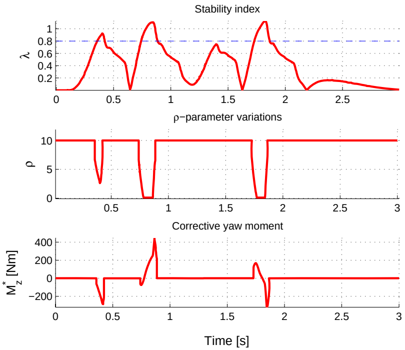


Figure: M_z^* and ρ variations according to χ for the double lane-change maneuver

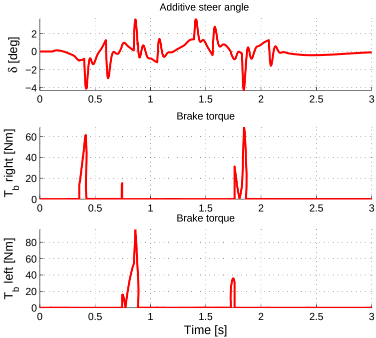
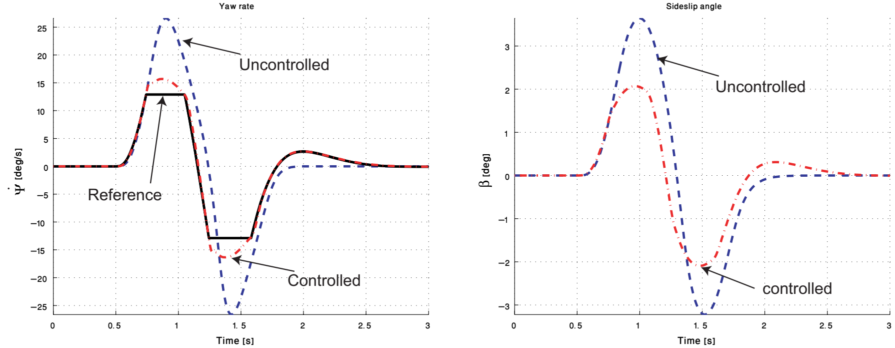


Figure: Control signals generated by the controller

Test 2: Results [wet road $\mu = 0.5$, $V = 80\ km/h$]



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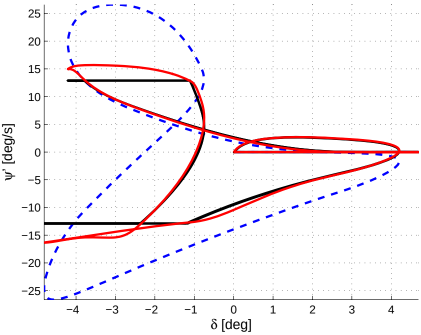

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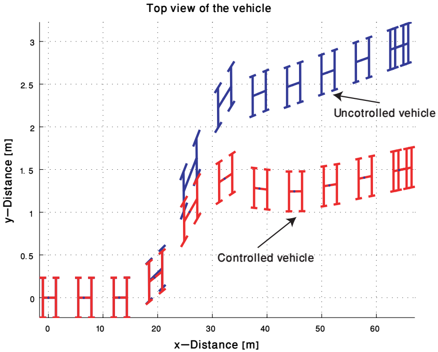

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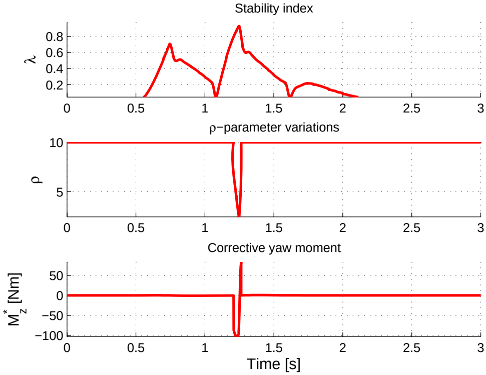


Figure: M_z^* and ρ variations according to χ for the steering maneuver

Outline

Road profile vehicle control adaptation

Road Profile estimation strategies

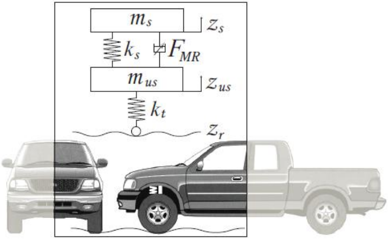
- The $\mathscr{H}_\infty$ observer for road profile estimation.
- The Algebraic flat observer for road profile estimation.
- The Parametric Adaptive Observation for road profile estimation.
- Guaranteed estimation based on interval analysis techniques.
- Vehicle-cloud-vehicle, data clustering and and identification.

LPV/$\mathscr{H}_\infty$ Road profile Adaptation control

- Road Adaptive Semi-Active Suspension for $1/4$ vehicle using an LPV/$\mathscr{H}_\infty$ Controller.
- A new LPV/$\mathscr{H}_\infty$ semi-active suspension control strategy for the full car with performance adaptation to roll behavior based on a non linear algebraic road profile estimation

Road profile vehicle control adaptation

Road Adaptive Semi-Active Suspension for $1/4$ vehicle using an LPV/$\mathscr{H}_\infty$ Controller

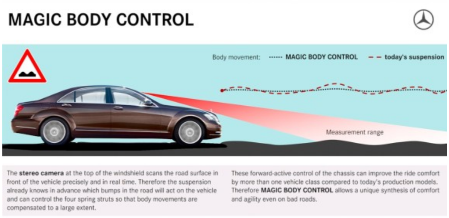


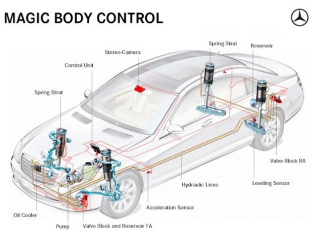
Road Adaptive control

One of the important investigation towards road safety

- On-line performance objectives adaptation (comfort vs roadholding).
- Less expensive and very efficient.

Suspension control and adaptation: Camera based road monitoring selective control, very recently (2013) by Mercedes Benz.





Road Adaptive control

- Road profile roughness estimation to identify the type of the road.
- LPV/H_∞ semi-active suspension control adaptation to the type of the road profile.

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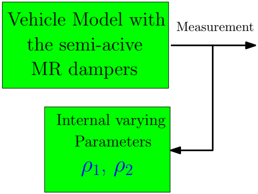

(dissipativity, saturation)

A shock absorber with magneto-rheological fluid that changes damping characteristics in the suspension system depending on electric current.

$$F_{MR} = I f_c \tanh\left(a_1 \dot{z}_{def} + a_2 z_{def}\right) + b_1 \dot{z}_{def} + b_2 z_{def}$$

$$\begin{aligned} \rho_1 &= \quad f(tanh(z_{def}, \dot{z}_{def}), I) \\ \rho_2 &= \quad f(sat(tanh(z_{def}, \dot{z}_{def}), I)) \end{aligned}$$

$$\begin{cases} \dot{x}_{lpv} = A_{lpv}(\rho_1, \rho_2) x_{lpv} + B_1 u_c + B_2 w \\ y_{lpv} = C_1 x_{lpv} \end{cases}$$

$\rho_1 \in [-1, 1] \longrightarrow$ Nonlinearities.
$\rho_2 \in [0, 1] \longrightarrow$ Saturation.

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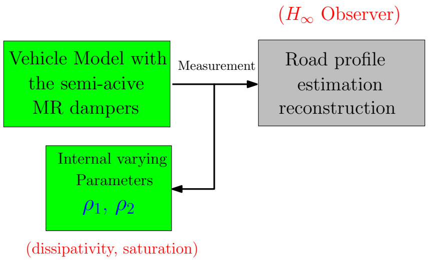

$$\hat{z}_r = [m_{us}\ddot{z}_{us} - k_s(\hat{z}_s - \hat{z}_{us}) + k_t\hat{z}_{us} - F_{MR}] \cdot k_t^{-1}$$

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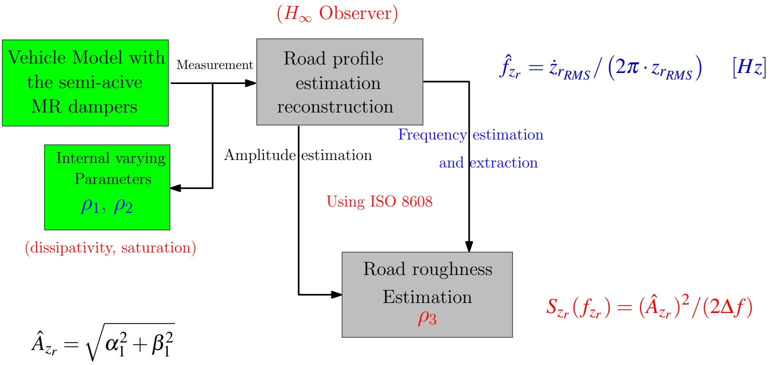

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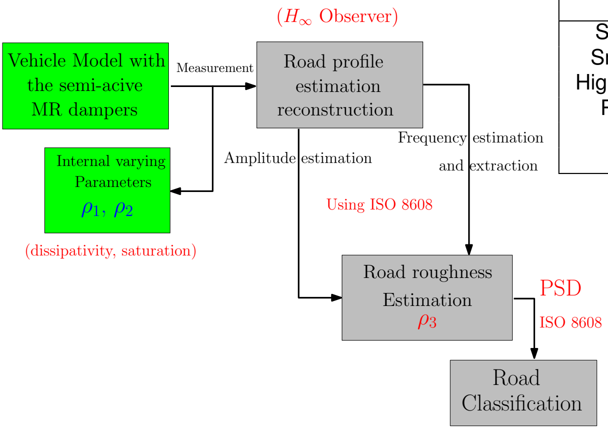

Type of Road	Class
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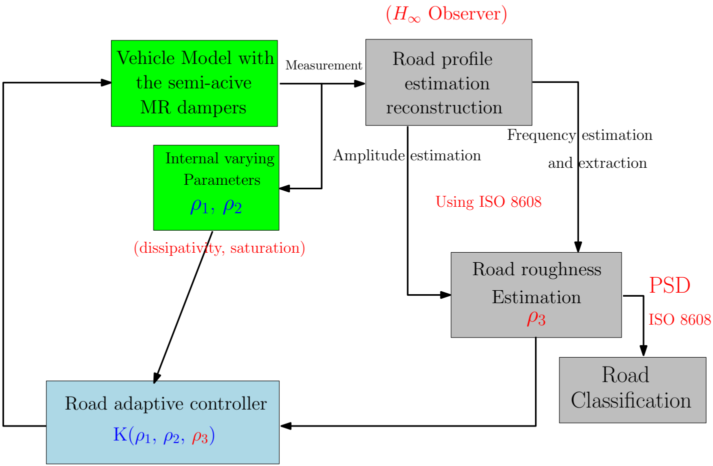

LPV/$\mathscr{H}_\infty$ control synthesis

Two scheduling parameters in the model:

$$\rho_1 = \quad f(tanh(z_{def}, \dot{z}_{def}), I)$$
$$\rho_2 = \quad f(sat(tanh(z_{def}, \dot{z}_{def}), I))$$

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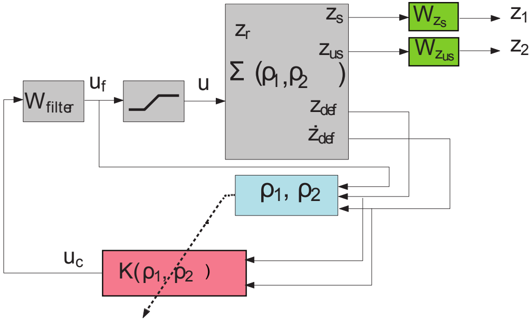


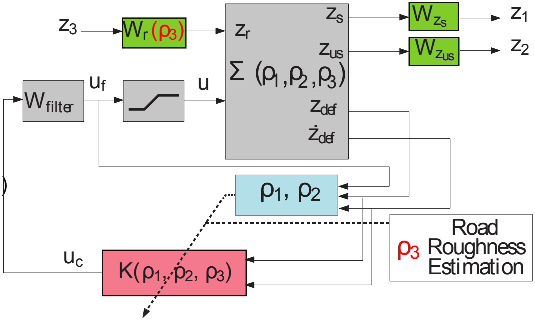
LPV/$\mathscr{H}_\infty$ control synthesis

One scheduling parameter ρ_3 for online suspension adaptation to the road profile:

$$\rho_3 = K_{\rho_3} \cdot S_{z_r}(f_{z_r}) \quad \in \quad [0,1] \tag{10}$$

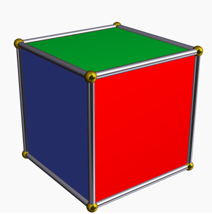
where K_{ρ_3} is used to bound ρ_3, such that

$$I(\rho_3) := \begin{cases} I = I_{max} \\ if\, \rho_3 \geq \overline{\rho_3} \\ I_{min} < I < I_{max} \\ if\, \underline{\rho_3} < \rho_3 < \overline{\rho_3} \\ I = I_{min} \\ if\, \rho_3 \leq \underline{\rho_3} \end{cases} \tag{11}$$



LPV/$\mathscr{H}_\infty$ control synthesis

The general LPV/$\mathscr{H}_\infty$ is obtained thanks to the polytopic appraoch, by solving the considered set of LMIs on each one of the $2^3 = 8$ vertices. The general LPV/$\mathscr{H}_\infty$ is a convex combination of the 8 local controllers.



$$S(\rho) = \sum_{k=1}^{2^3} \alpha_k(\rho) \begin{bmatrix} A_{c_k} & B_{c_k} \\ C_{c_k} & D_{c_k} \end{bmatrix}$$

where,

$$\alpha_k(\rho) = \frac{\prod_{j=1}^{2^3} |\rho(j) - \mathscr{C}^c(\Omega_k)_j|}{\prod_{j=1}^{i} (\overline{\rho}(j) - \underline{\rho}(j))} \ ,$$

$$\sum_{k=1}^{2^3} \alpha_k(\rho) = 1 \ , \ \alpha_k(\rho) > 0$$

Road adaptive control validation & implementation

The test bench is composed of:

- The process: $1/5$ scaled real vehicle equipped with 4 Electro-Rheological semi-active dampers and 4 DC motors to generate the desired road profiles.
- Matlab/Simulink environment $+$ Xpc target environment for real time data acquisition and control.

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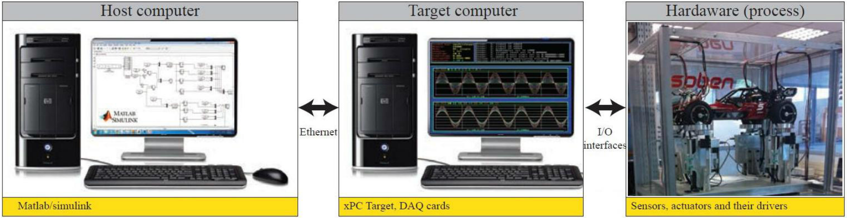

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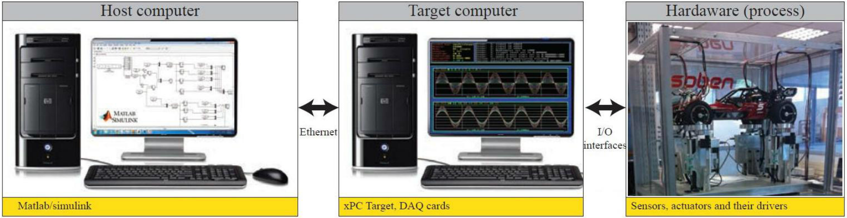
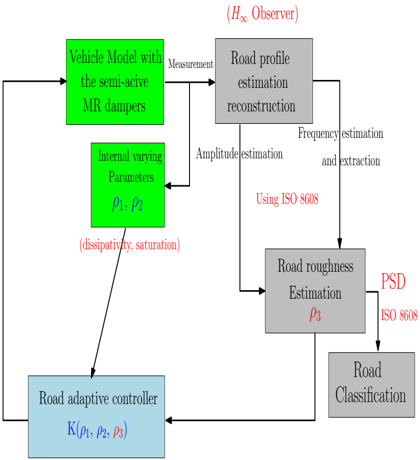
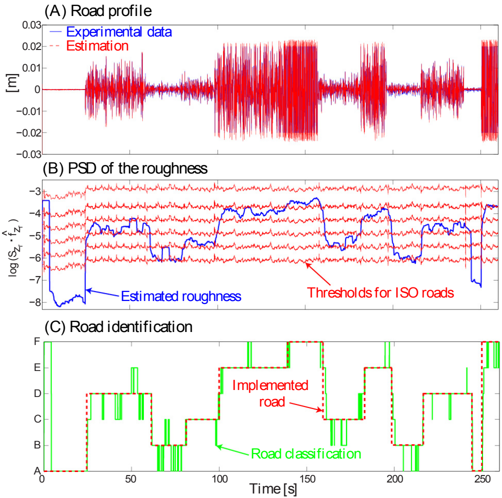


Road classification implementation results





Road classification implementation results

Table: Road profiles Classification (ISO 8608).

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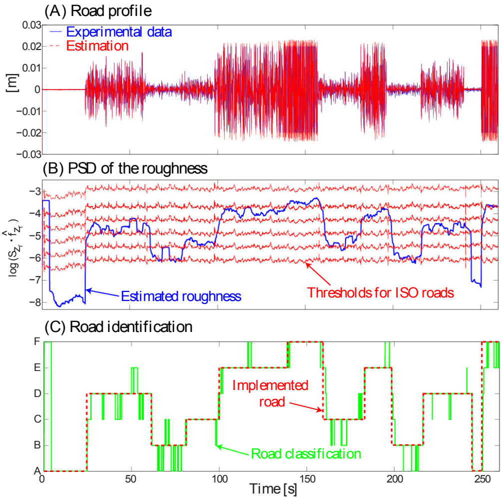
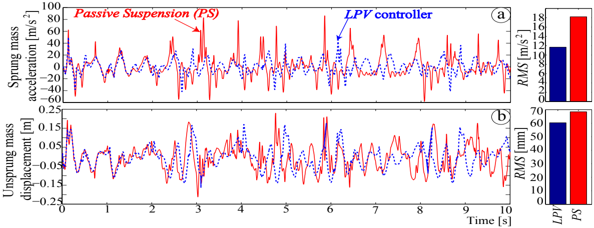


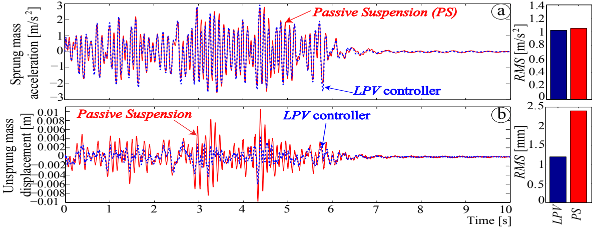
Road adaptive control implementation results

(a) plowed field (hard road F) at low velocity (comfort objective)



(b) Raod type F, $v_x = 30\ Km/h$

(c) highway (smooth road ISO A) at low velocity (road haolding objective)



(d) Raod type A, $v_x = 100\ Km/h$

Outline

Towards global chassis control approaches (GCC)

Some facts

- Vehicle-dynamics sub-systems control (suspension, steering, stability, traction ....) are traditionally designed and implemented as independent (or weakly interleaved) systems.
- Global collaboration between these systems is done through empirical rules and may lead to inappropriate or conflicting control objectives.

What is GCC ?

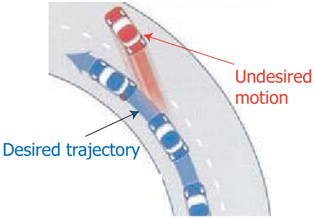
- combine several (at least 2) subsystems in order to improve the vehicle global behavior Shibahata (2004)
- tends to make collaborate the different subsystems in view of the same objectives, according to the situation (constraints, environment, ...)
- is develop to improve comfort and safety, according to the driving situation, accounting for actuator constraints and to the eventual knowledge of the vehicle environment

Active safety using LPV FTC VDC coordinated control

Key points

Yaw is one of the most complex dynamics to handle on a ground vehicle. FTC LPV control:

- Prevents vehicle from skidding and spinning out
- Improves lateral vehicle dynamics face to critical situations
- Handle Braking and suspension actuator malfunctions and Steering activation



The LPV FTC strategy

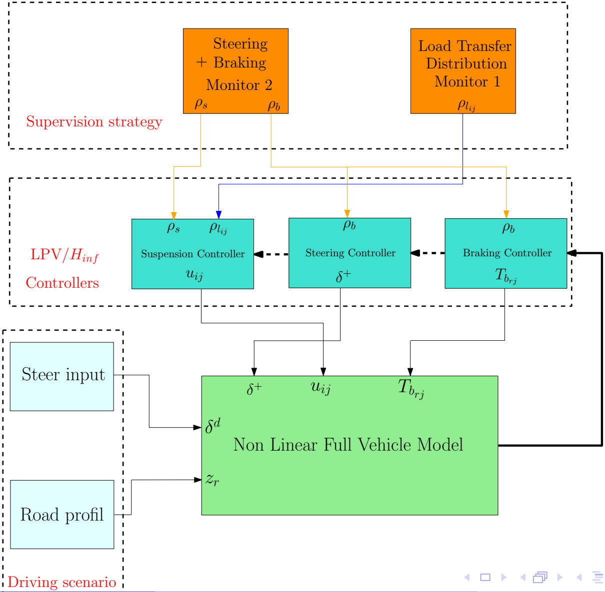
Monitoring Parameters

- Braking efficiency : torque transmission
- Steering activation during emergency situation (low slip)
- LTR: roll induced load transfer by damper malfunctions

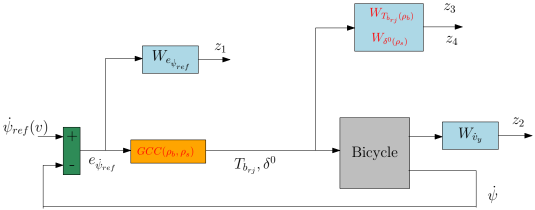
Control Issues

- Lateral coordinated steering/braking control: parameter dependent weighting functions for braking torque limitation and activation of the steering action
- Full car vertical suspension control: fixed control structure for suspension force distribution, parameter dependent weighting functions for roll attenuation in critical situations and comfort improvement in normal ones.

Global chassis control implementation scheme



Coordinated steering/braking control



Vehicle model : Single track model (dry road).

Inputs/Ouputs:

$$
\begin{aligned}
w(t) &= [\dot\psi_{ref}(v)(t), M_{dz}(t)] \\
u(t) &= [\delta^+(t), T^+_{b_{rl}}(t), T^+_{b_{rr}}(t)] \\
y(t) &= e_{\dot\psi}(t) \\
z(t) &= [z_1(t), z_2(t), z_3(t)]
\end{aligned}
$$

Weighting functions for performance requirements

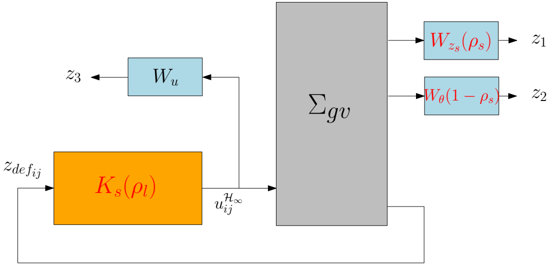
$W_{e_{\dot\psi}}$ and $W_{\dot v_y}$ are 1st order systems.

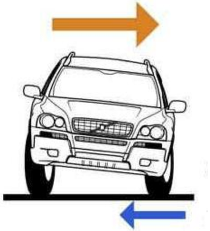
Weighting functions for actuator coordination

- $W_\delta(\rho_s)$ = $(1-\rho_s)\times$ 4th order $\rightarrow$ braking (and steering) penalized if $\rho = \overline{\rho}$
- $W_{T_{b_{rj}}}(\rho_b)$ = $(1-\rho_b)\times$ 1st order $\rightarrow$ braking (and steering) allowed if $\rho = \underline{\rho}$

When a high slip ratio is detected (critical situation) , the tire may lock, so $\rho_b \rightarrow 0$ and the gain of the weighting function is set to be high.
This allows to release the braking action leading to a natural stabilisation of the slip dynamic.

The suspension control configuration





A new partly fixed control structure: manage the suspension control distribution in case of damper malfunction

$$K_s(\rho_s,\rho_l) := \begin{cases} \dot{x}_c(t) = A_c(\rho_s,\rho_l)x_c(t) + B_c(\rho_s,\rho_l)y(t) \\ \begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \begin{pmatrix} 1-\rho_l & 0 & 0 & 0 \\ 0 & \rho_l & 0 & 0 \\ 0 & 0 & 1-\rho_l & 0 \\ 0 & 0 & 0 & \rho_l \end{pmatrix} C_c^0(\rho_s)x_c(t) \end{cases}$$

ρ_l allows to generate the adequate suspension forces in the 4 corners of the vehicle depending on the load transfer (left $\leftrightarrows$ right) caused by the performed driving scenario.

Simulations on a full NL vehicle model

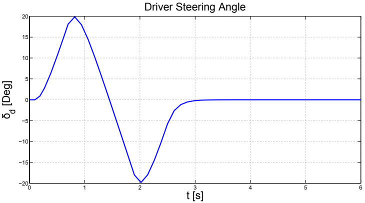
Simulation results

- Vehicle Automotive 'GIPSA-lab' toolbox
  - Full nonlinear vehicle model
  - Validated in a real car "Renault Mégane Coupé" coll. MIAM lab [Basset, Pouly and Lamy] see C. Poussot-Vassal PhD. thesis

The stabilizing torques T_b^* provided by the controller is then handled by a local ABS strategy Tanelli et al. (2008)

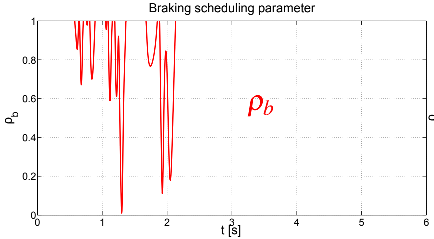
Simulation scenario

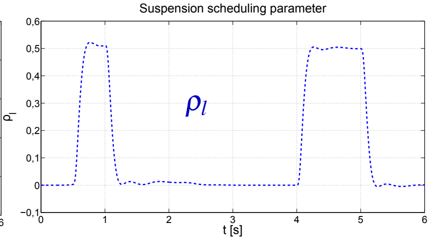
Double lane-change maneuver at $100\ km/h$ on a WET road (from $t = 2$s to $t = 6$s)

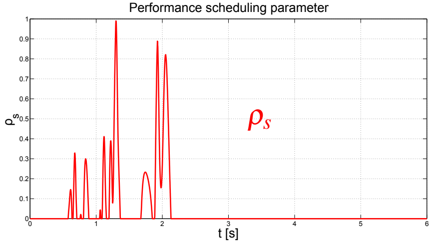


- Faulty left rear braking actuator: saturation = $75N$
- 5cm Road bump from $t = 0.5$s to $t = 1.5$s and from $t = 4$s to $t = 5$s)
- Faulty front left damper: force limitation of 70%
- Lateral wind occurs at vehicle's front generating an undesirable yaw moment (from $t = 2.5$s to $t = 3$s).

Monitoring parameters

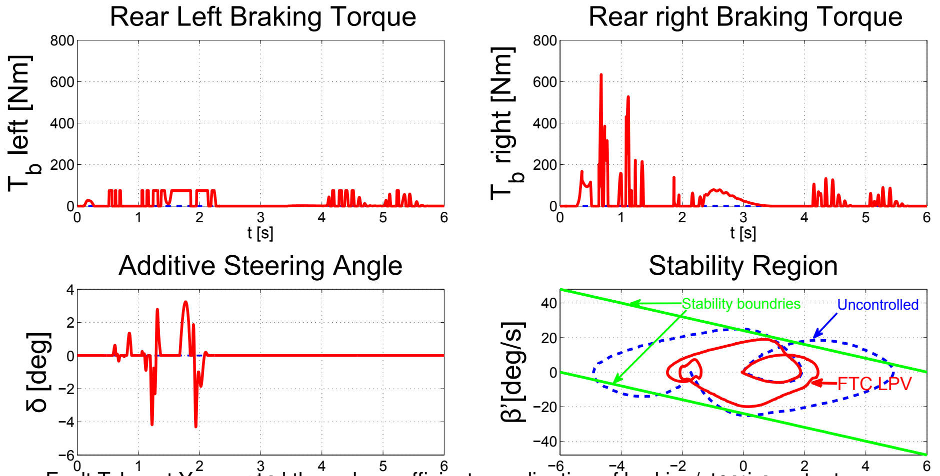






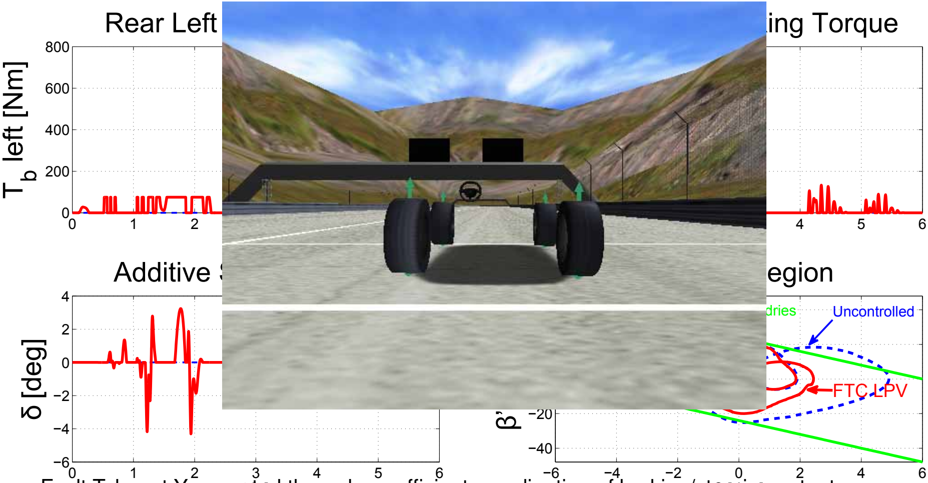
- ρ_b handles the braking efficiency
- ρ_s: activation of the steering actuator
- ρ_l (LTR): Coordination of suspension control and on-line modification of the suspension performances

Braking/Steering actuators - stability analysis



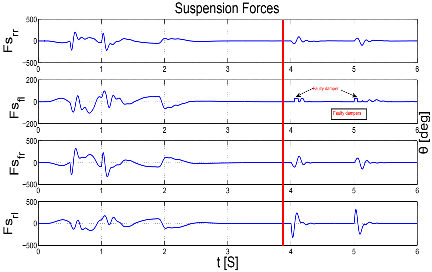
Fault Tolerant Yaw control through an efficient coordination of braking/steering actuators.

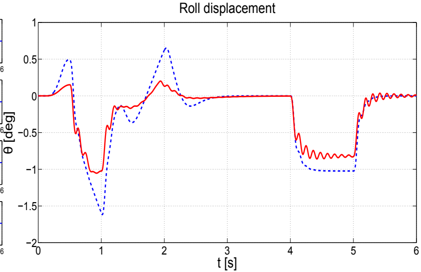
Braking/Steering actuators - stability analysis

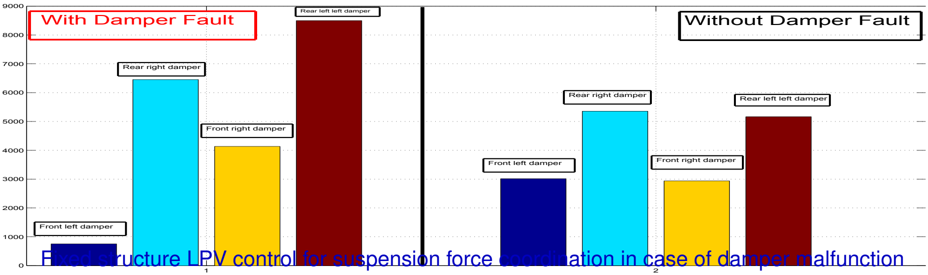


Fault Tolerant Yaw control through an efficient coordination of braking/steering actuators.

Suspension control distribution







Fixed structure LPV control for suspension force coordination in case of damper malfunction

Conclusions

About today's presentation:

An approach to the vehicle yaw stabilizing problem. . .

- Objective: Enhance vehicle steerability and stability
  - Steerability is enhanced in normal driving condition.
  - Braking is involved only when the vehicle tends to instability.
- Flexible design: Integration of different scheduled sub-controllers
- Scheduling parameters: Estimation of the sideslip angle
- Real-time implementation: General structure does not involve online optimization

Future work

- Implementation of the controller in a real car
- Integration of the suspension system in the control scheme
- Design of an LPV vehicle system
    - Variation of the cornering stiffness with respect to road conditions (dry, wet, icy,. . .)

Thank you for your attention

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