Robust and LPV control of suspension systems

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Outline

Outline

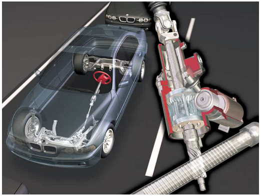
Challenges in chassis control

Today's vehicles. . .

- Growth of controlled organs: suspensions, ABS, ESC, ABC, brake repartition, active steering, tire pressure, TCS
- Increasing number of sensors & actuators
- Heavy networking + need to to synchronize many subsystems

Need for advanced control to improve

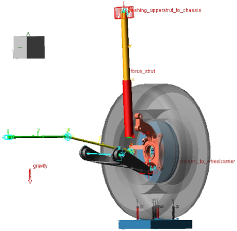
- Driving comfort (and pleasure)
- Active safety

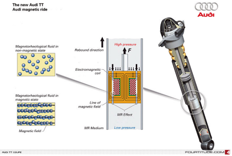


French ANR INOVE project 2010-2015

INtegrated approach of Observation and control and VEhicle dynamics

Objective: improve comfort and road holding of car vehicle (active or semi-active suspensions, steering, braking)





French ANR INOVE project 2010-2015

INtegrated approach of Observation and control and VEhicle dynamics

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Long term International supported collaboration projects

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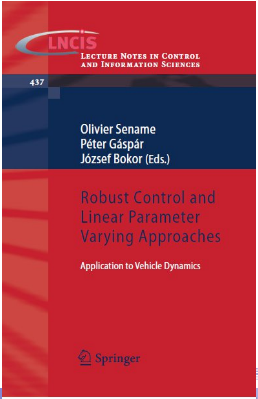
Introduction

This course has been mainly written thanks to:

- the PhD dissertations of [Poussot-Vassal(2008), Alessandro Zin (2005), Damien Sammier (2002), Sébastien Aubouet (2010), Anh Lam DO (2011), Soheib Fergani (2014), Manh Quan Nguyen (2016)].
- the Post-doctoral work of [Moustapha Doumiati (2010)]
- the authors' works since 1995
- interesting books cited below



Outline

Framework & Objectives

Semi-active suspension control research

- LQ clipped: complex, involve state measurement Tseng et al. [VSD, 1994]
- $\mathscr{H}_\infty$ & skyhook clipped: Sammier et al. [VSD, 2003]
- MPC based: involve optimization, state measurement, robustness? Canale et al. [Trans. CST, 2006], Giorgetti et al. [IJRNLC, 2006], Guia et al. [VSD, 2004]
- ADD, Mixed SH-ADD: simple structure, comfort oriented Savaresi et al.[ASME, 2005, 2007]

Objectives

- Enhance passenger comfort & road-holding
- Ensure semi-active constraint
- Simplify controller structure

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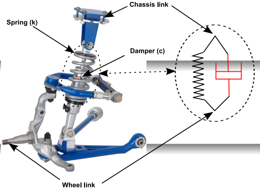
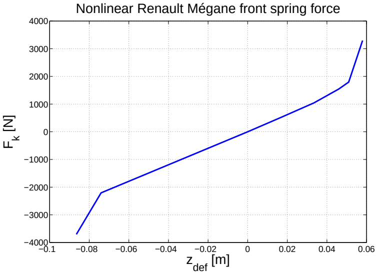
Suspension system

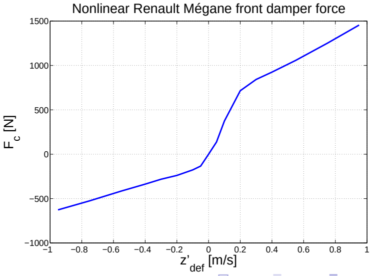
Objective

- Link between unsprung and sprung masses
- Influences comfort / road-holding performances
- Involves vertical (z_s, z_{us}) dynamics

Passive quarter vehicle model







Suspension system

Objective

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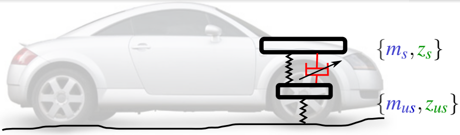
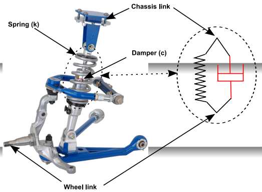
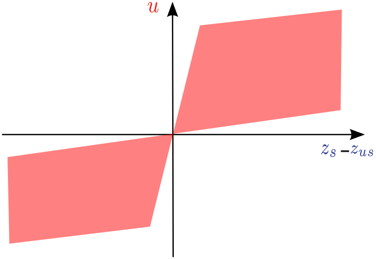
Semi-active quarter vehicle model





The controlled actuator can only dissipate energy (i.e. modify the damping factor in real time): good performances, fast dynamics, weight comparable to passive ones, economically viable.



Suspension system

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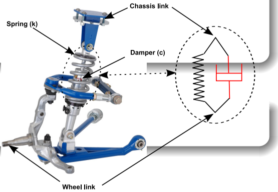
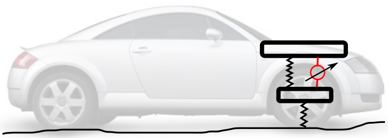
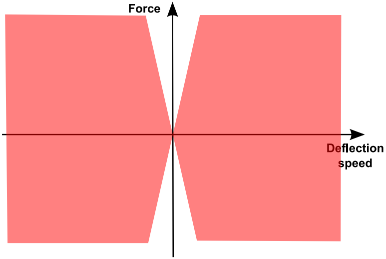


Active quarter vehicle model



$\{m_s, z_s\}$

$\{m_{us}, z_{us}\}$



The controlled actuator

Different types of suspensions : a summary

- Passive suspensions
  - fixed (linear or non linear) characteristics that can be optimized (by adjusting the shape of the speed-effort rule (SER)) in order to orientate the vehicle towards comfort or road holding, Oustaloup et al. (1996)
- Active suspensions
  - the passive damper is replaced (or helped) by a controlled actuator, able to provide a force whatever the deflection speed of the damper
    - allows very good performances
    - not economically viable
    - limited dynamics, high weight
    - only for up-market vehicles
- Semi-active suspensions
  - the passive damper is replaced (or helped) by a controlled actuator, able only to dissipate energy (i.e. modify the damping factor, by adjusting the shape of the speed-effort rule (SER)) in real time
    - allows good performances (however, not as good as those of active ones)
    - economically viable
    - fast dynamics, weight comparable to passive ones
    - for mid-range vehicles

Outline

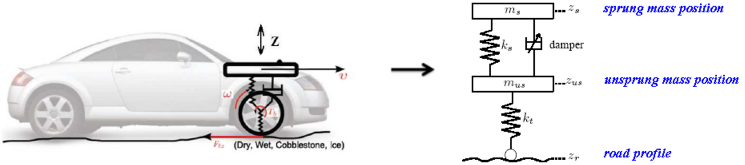


Figure: Simple quarter vehicle model for semi-active suspension control

Quarter vehicle dynamics

$$\left\{ \begin{array}{ll} m_s\ddot{z}_s & = -k_s z_{def} - F_{damper} \\ m_{us}\ddot{z}_{us} & = k_s z_{def} + F_{damper} - k_t\left(z_{us} - z_r\right) \end{array} \right. \tag{1}$$

$z_{def} = z_s - z_{us}$: damper deflection, $\dot{z}_{def} = \dot{z}_s - \dot{z}_{us}$: deflection velocity.

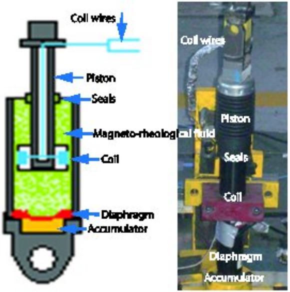
- The damper's characteristics : Force-Deflection-Deflection Velocity relation

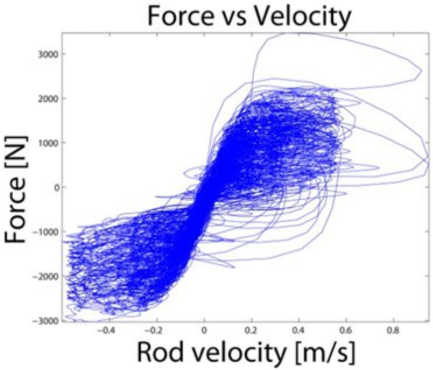
$$F_{damper} = g\left(z_{def}, \dot{z}_{def}\right) \tag{2}$$

where g can be linear or nonlinear.

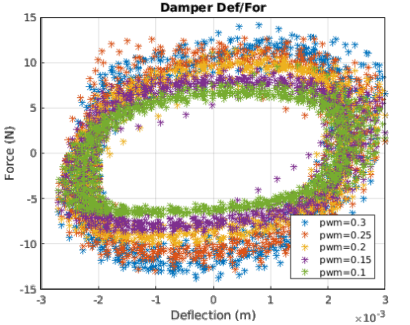
Magneto-Rheological (MR) dampers - ITESM Mexique

- adaptive behavior through the application of a magnetic field
- fast time response and a low battery voltage consumption.
- but highly non linear: bi-viscosity, temperature dependency and hysteresis

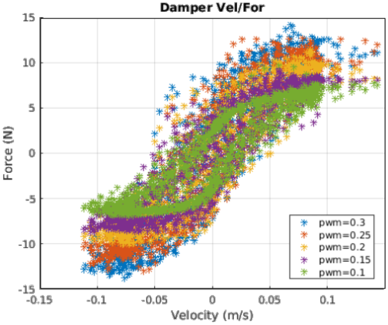




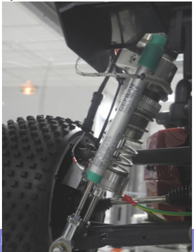
Electro-Rheological (ER) dampers -GIPSA



Force-Displacement map of a semi-active damper



Force-Velocity map of a semi-active damper



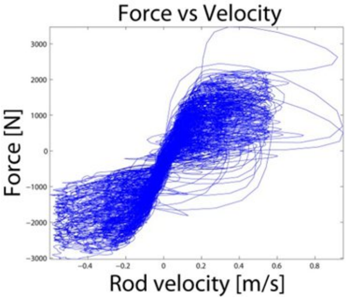
Magneto-Rheological (MR) dampers: a simulation model

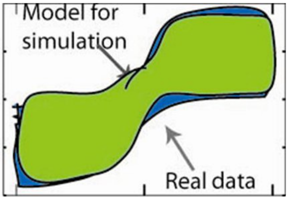
- adaptive behavior through the application of a magnetic field
- fast time response and a low battery voltage consumption.
- but highly non linear: bi-viscosity, temperature dependency and hysteresis

Model developed in Lozoya et al 2009:

$$F_{damper} = C_1 \tanh(C_2\dot{x}_{mr} + C_3 x_{mr}) + C_4\dot{x}_{mr} + C_5 x_{mr} + C_6\ddot{x}_{mr} + C_7 I \tanh(C_8\dot{x}_{mr} + C_9 x_{mr}) \quad (3)$$

identified on the test-rig at *Metalsa* [1] with diffrent values in compression ($\dot{x}_{mr} < 0$) and extension ($\dot{x}_{mr} \geq 0$) modes





[1] www.metalsa.com.mx

Controlled-oriented models

Semi-active LINEAR damper model (Method 1)

$$F_{damper} = c_0\dot{z}_{def} + u, \quad c_0 = (c_{max} + c_{min})/2$$

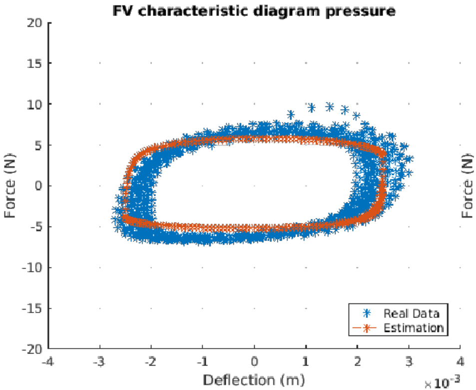
used in most of control approaches (optimal, MPC, H_∞, H_2...), with F_{damper} bounded.

Semi-active NON LINEAR MR/ER damper model (Method 2) [Gu et al., 2006]

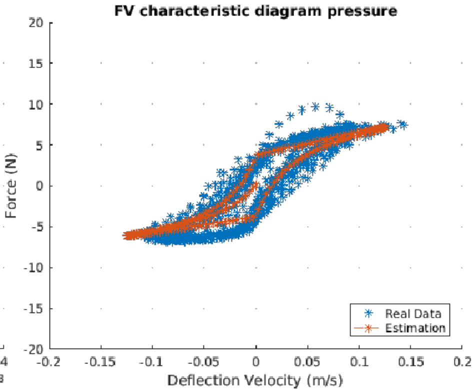
$$F_{damper} = \underbrace{c_0\dot{z}_{def} + k_0 z_{def}}_{passive} + \underbrace{f_c \cdot \tanh\left(c_1\dot{z}_{def} + k_1 z_{def}\right)}_{semi-active} \tag{4}$$

- $\tanh$: allows to model the bi-viscous behavior.
- f_c is a controllable force and depends on input current I (or voltage V). Constraint on f_c: $0 \leq f_{cmin} \leq f_c \leq f_{cmax}$ - passivity constraint. (f_{cmin}=soft damper, f_{cmax}=hard damper).

Model validation



Force-Displacement map validation



Force-Velocity map validation

Outline

Performance objectives

Comfort

- linked to the road vibration isolation of the chassis
- relating to the human sensitivity (between 0.5Hz and 20 Hz): ISO 2631

★ (C1) Comfort at high frequencies: $\ddot{z}_s/z_r$, [4-30]Hz

★ (C2) Comfort at low frequencies: z_s/z_r, [0-5]Hz

Road holding

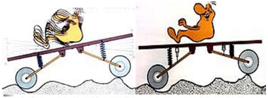
- concerns the wheel rebound (or tire deflection)... keep contact between the wheel and the road

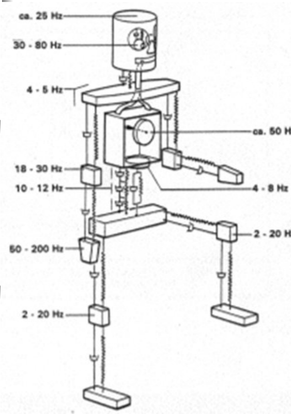
★ (RH1) Road-holding: z_{us}/z_r, [0-20]Hz

Constraints

- End-stop, saturation

★ (RH2) Suspension constraints: z_{def}/z_r, [0-20]Hz



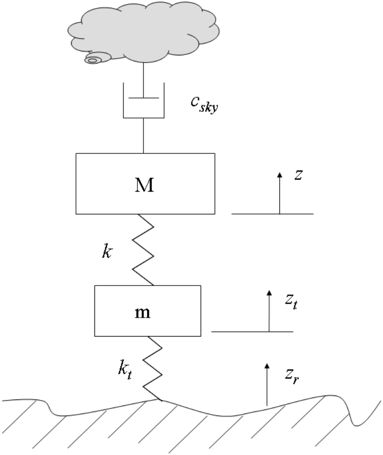


Some suspension control approaches

Skyhook control (SH)

The idea is to clip the body of the car to the sky.

- Advantages : Simple, 2 degree of freedom, improvement of comfort.
- Drawbacks : Cannot improve road holding (comfort oriented only).
- (Karnopp et al. 1974), (Emura et al. 1994), (Sammier et al. 2003), (Poussot et al. 2006), ...



Ideal

Some suspension control approaches

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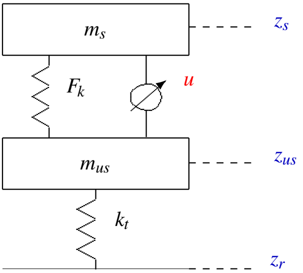
- Drawbacks : Cannot improve road holding (comfort oriented only).
- (Karnopp et al. 1974), (Emura et al. 1994), (Sammier et al. 2003), (Poussot et al. 2006), ...
- It can be approached by a realizable version, where the SH damper force u is represented by the following equation :

$$u = -c_{sky}\dot{z_{def}} - c_{sky}(1-\alpha)\dot{z_{us}} \qquad (5)$$



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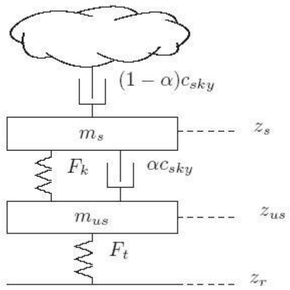
- It can be approached by a realizable version, where the SH damper force u is represented by the following equation :

$$u = -c_{sky}\dot{z_{def}} - c_{sky}(1-\alpha)\dot{z_{us}} \qquad (6)$$

leading to the following scheme :



Approached Skyhook

Some suspension control approaches

Skyhook two-state damper control (SH-2)

The two-state Skyhook control is an on/off strategy that switches between high and low damping coefficients in order to achieve body comfort specifications.

Proposition (SH-2 state control)

This law is defined as :

$$c_{in} = \begin{cases} c_{min} & \text{if } \dot{z}_s \dot{z}_{def} \leq 0 \\ c_{max} & \text{if } \dot{z}_s \dot{z}_{def} > 0 \end{cases} \quad (7)$$

- Basically, it consists in a switching controller which deactivates the controlled damper when the body speed $\dot{z}_s$ and suspension deflection speed $\dot{z}_{def}$ have opposite signs.
- Only needs to have two damping coefficient states.
- Simple strategy but requires two sensors.

Many studies have concerned the Skyhook control strategy since it represents a simple but efficient way to achieve good comfort requirement (see *e.g.* (Simon 2001), (Ahmadian et al. 2004). Some extended versions of the Skyhook control have been also developed, such as the adaptive one in (Song et al. 2007) or the gain-scheduled one in (Hong et al. 2002).

Some suspension control approaches

Skyhook linear approximation damper control (SH-L)

An improved version of Skyhook control has been used to handle variable damping, either with discrete damping coefficients, or with continuously variable damper, as illustrated in (Sohn et al. 2000), (Sammier et al. 2003). The linear approximation of the Skyhook control algorithm, adapted to semi-active suspension actuators, is given as :

Proposition

The SH-L law is defined as:

$$c_{in} = \begin{cases} c_{min} & \text{if } \dot{z}_s \dot{z}_{def} \leq 0 \\ \textbf{sat}\Big(\dfrac{\alpha c_{max} \dot{z}_{def} + (1-\alpha) c_{max} \dot{z}_s}{\dot{z}_{def}}\Big) & \text{if } \dot{z}_s \dot{z}_{def} > 0 \end{cases} \quad (8)$$

*where $\alpha \in [0;1]$ is a tuning parameter that modifies the closed-loop performances and **sat()** denotes that $c_{in} \in [c_{min}; c_{max}]$.*

As the SH-2, the SH-L modifies the damping factor according to $\dot{z}_s$ and $\dot{z}_{def}$, but the innovation comes from the infinite number of possible damping coefficients.

- Equivalent to the Skyhook two-state control when $\alpha = 1$.
- Requires a continuously variable controlled damper (*e.g.* an MR damper).
- Requires only two measurements and is simple to implement, but suffers of $\dot{z}_{def}$ zero crossing

Some suspension control approaches

Acceleration Driven Damper control (ADD)

The ADD control (Savaresi et al. 2005) is a semi-active control which consists in changing the damping factor using the body acceleration $\ddot{z}_s$ knowledge.

Proposition

The ADD law is defined as:

$$c_{in} = \begin{cases} c_{min} & \text{if } \ddot{z}_s \dot{z}_{def} \leq 0 \\ c_{max} & \text{if } \ddot{z}_s \dot{z}_{def} > 0 \end{cases} \tag{9}$$

- Strategy shown to be optimal (it minimizes $\ddot{z}_s$ when no road information is available).
- Very similar to the two-state approximation of the Skyhook algorithm, with the difference that the switching law depends on $\ddot{z}_s$, instead of $\dot{z}_s$ (which is easier to measure in practice).
- Simple from the implementation point of view, since it requires the same number of sensors as the SH-2 and SH-L control laws.
- ADD design well adapted to comfort improvement but not to road-holding.

Some suspension control approaches

Power Driven Damper (PDD)

In (Morselli and Zanasi 2008), the authors propose a semi-active suspension control strategy using the port Hamiltonian techniques, which provide powerful tools for modeling mechatronics systems with dissipative components.

Proposition

The PDD control approach is described by:

$$c_{in} = \begin{cases} c_{min} & \text{if } k z_{def}\dot{z}_{def} + c_{min}\dot{z}_{def}^2 \geq 0 \\ c_{max} & \text{if } k z_{def}\dot{z}_{def} + c_{max}\dot{z}_{def}^2 < 0 \\ \dfrac{c_{min}+c_{max}}{2} & \text{if } z_{def} \neq 0 \text{ and } \dot{z}_{def} = 0 \\ -\dfrac{k z_{def}}{\dot{z}_{def}} & \text{otherwise} \end{cases} \tag{10}$$

where k is the stiffness of the considered suspension.

- The authors show that this strategy provides results comparable to those of the ADD control law, while avoiding the chattering effect of the damping control value.
- The additional cost : the need for the knowledge of the spring stiffness k and a more complex rule.

Some suspension control approaches

Mixed Skyhook-Acceleration Driven Damper (SH-ADD)

The SH-ADD mixes the best behavior of SH and ADD, without increasing the computational effort nor the hardware complexity. The Key idea is the use of a very simple *frequency range selector*, which distinguishes the dynamical behavior of the suspension (SH selected for the low frequency dynamics, otherwise ADD).

Proposition

The SH-ADD approach is described by:

$$c_{in} = \begin{cases} c_{max} & \text{if} \quad \begin{array}{l} \left[(\ddot{z}_s{}^2 - \alpha^2 \dot{z}_s{}^2) \leq 0 \text{ and } \dot{z}_s \dot{z}_{def} > 0\right] \text{or} \\ \left[(\ddot{z}_s{}^2 - \alpha^2 \dot{z}_s{}^2) > 0 \text{ and } \dot{z}_s \ddot{z}_{def} > 0\right] \end{array} \\ c_{min} & \text{otherwise} \end{cases} \tag{11}$$

where $\alpha \in \mathbb{R}^+$ is the only tuning parameter allowing for frequency range selector, i.e., it adjusts the "switch" between the SH and the ADD.

- The amount $(\ddot{z}_s{}^2 - \alpha^2 \dot{z}_s{}^2)$ is the simple "frequency-range selector" where α represents the frequency limit between the low and the high frequency ranges (value set at the cross-over frequency (in rad/s) between SH and ADD).
- Resulting control law very simple and requires the same apparatus as SH.
- A simplified version using one single sensor leads to very satisfactory results, (see (Savaresi

Some suspension control approaches

Ground-hook 2 state control (GH-2)

Very few studies have been devoted to the possible improvement of road-holding, using suspension actuator. Since few years, the studies on Global Chassis Control have shown that the suspension system may also help getting better road-handling.
In a dual way to the Skyhook case, the GH-2 control (Valasek et al. 1998) consists in a switching control law depending now on the sign of the product between the suspension deflection speed $\dot{z}_{def}$ and the speed of the unsprung mass $\dot{z}_{us}$.

Proposition

The GH-2 control approach is given by :

$$c_{in} = \begin{cases} c_{min} & \text{if } -\dot{z}_{us}\dot{z}_{def} \leq 0 \\ c_{max} & \text{if } -\dot{z}_{us}\dot{z}_{def} > 0 \end{cases} \tag{12}$$

- This control has globally the same properties as the SH-2 one, but focuses on the unsprung mass m_{us} instead of the body m_s.

Some suspension control approaches

Ground-hook linear (GH-L)

In this case, the semi-active damper allows to continuously change the damping coefficient.

Proposition

The GH-L control approach is defined by :

$$c_{in} = \begin{cases} c_{min} & \textit{if } -\dot{z}_{us}\dot{z}_{def} \leq 0 \\ \textbf{\textit{sat}}\left(\dfrac{\alpha c_{max}\dot{z}_{def} + (1-\alpha)c_{max}\dot{z}_{us}}{\dot{z}_{def}}\right) & \textit{if } -\dot{z}_{us}\dot{z}_{def} > 0 \end{cases} \quad (13)$$

*where $\alpha \in [0;1]$ is a tuning parameter that modifies the closed-loop performances and **sat( )** denotes that $c_{in} \in [c_{min}; c_{max}]$.*

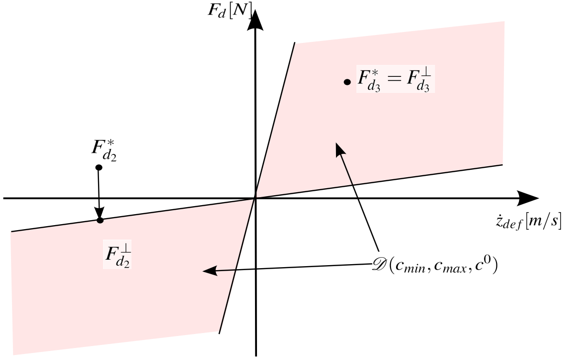
- Equivalent to the Groundhook two-state control (GH-2) when $\alpha = 1$.

Some suspension control approaches

Semi-active suspensions : Clipped control

- The "*clipped control*" approach consists in designing an active control, mainly based on linear (without taking into account passivity constraints), and then to make it semi-active by saturating the control signal.
- So, many works have concerned the application of classical control methods (*e.g.* H_∞, H_2, pole placement, disturbance rejection, optimal, active Skyhook ...). Then, the *dissipative constraint* of the damper is usually handled using a simple projection (*i.e.* saturation, as shown in the figure related to semi-active suspension), as in (Zin et al. 2008), (Karnopp 1983), (Margolis 1983).
- In the control step, the force applied by the semi-active damper is then chosen to be as close as to the force required by the controller for a given suspension deflection speed and for the possible range of forces the damper can deliver. This simple strategy has been then applied in many cases (see, for instance Rossi and Lucente (2004), Du et al. (2005), Sename and Dugard (2003).

The semi-active control paradigm



- Usual "Clipped control" (design without constraint and saturation) : leads to unpredictable behaviors
- Objective 1 : handle the semi-active constraintthrough an LPV model based approach with non-linear damper model Do et al [IFAC WC 2011, Springer 2012]
- Objective 2 : account for loss of damping efficiency using an additional parameter

Some suspension control approaches

Semi-active suspensions

- The question of optimality arises for clipped control which leads to unpredictable behaviors and ensures neither closed-loop internal stability nor performances any longer (it is a *synthesize and try* method). To cope with this last drawback, some modern control techniques have been applied to the specific semi-active suspension problem.

What's about the MPC approach ?

- MPC for quarter car model [Canale TCST06], [Giorgetti IJC06]
- MPC for full car model [Sawodny 2014] (active dampers + road preview)
- MPC full car [Nguyen et al CDC16]
  - Take into account explicitly the input constraints by a MIMO MPC
  - MPC design with considering the road disturbance effects using estimation

  ⇒ Enhance passenger comfort and handling

However, often "not sufficiently fast" for practical implementation

LPV Controller structure philosophy

Principle

The idea is to design a controller

- where the control input is limited when the required force is achievable by the semi-active actuator
- synthesized on the quarter vehicle model

Methodology

The proposed strategy is designed so that it minimizes the $\mathscr{H}_\infty$ performance criteria while guaranteeing the *dissipative constraint*, thanks to a specific parameter dependent structure and a scheduling strategy design.
We use the $\mathscr{H}_\infty$ synthesis, extended to LPV systems Shamma et al. [Automatica, 1991], Scherer et al. [TAC, 1997] and Scherer [IJRNLC, 1996].

- $\mathscr{H}_\infty$ synthesis: frequency based performance criteria, $\frac{||z||_2}{||w||_2}$ (as pole placement, disturbance rejection)
- LPV: Linear Parameter Varying, to handle nonlinearity or derive adaptive controller

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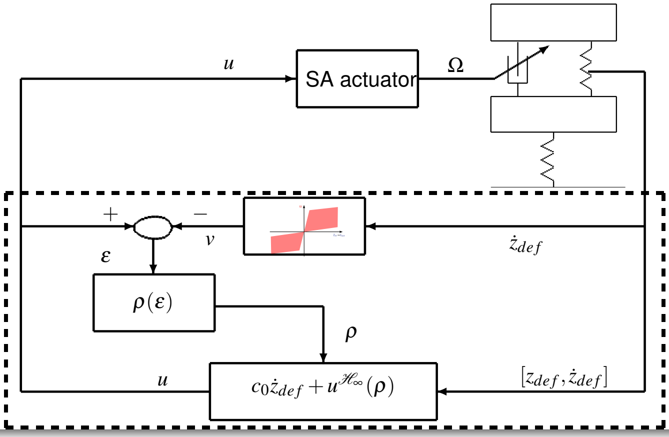
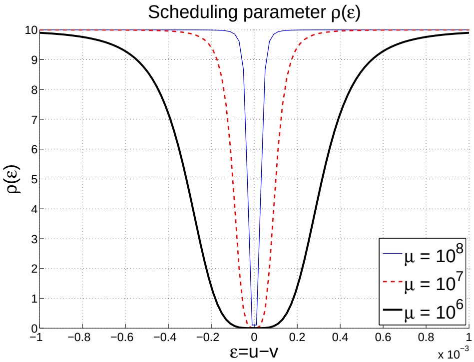
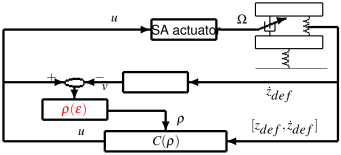
Implementation scheme & principle



Scheduling strategy

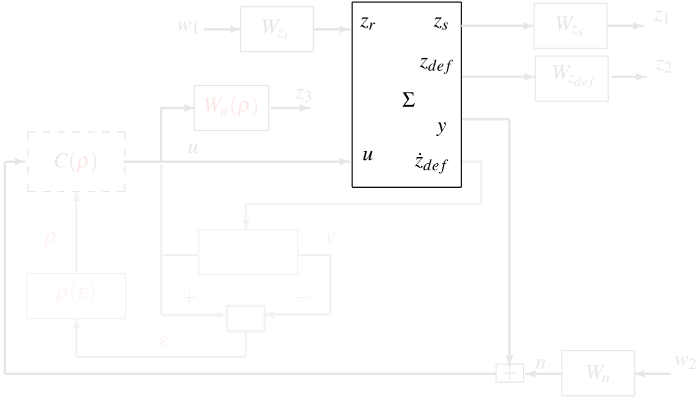
Scheduling strategy



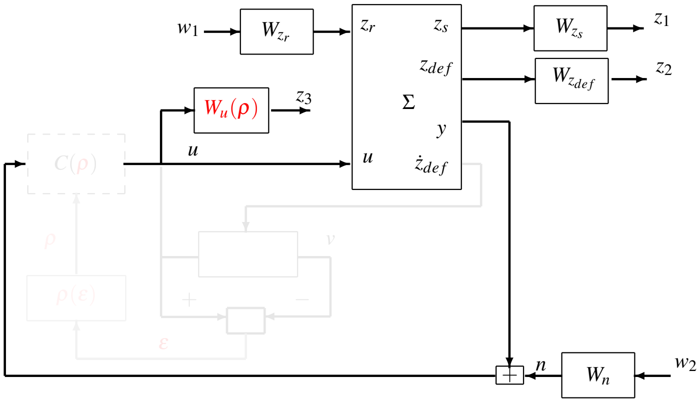


- $\rho(\varepsilon) \in [\ 0.1 \quad 10\ ]$

LPV control design (1)



LPV control design (1)



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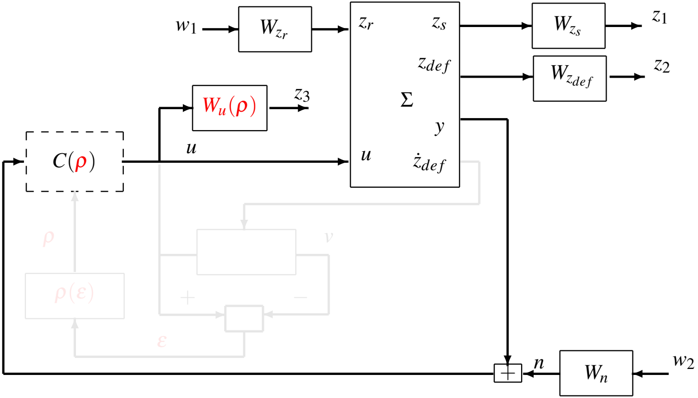

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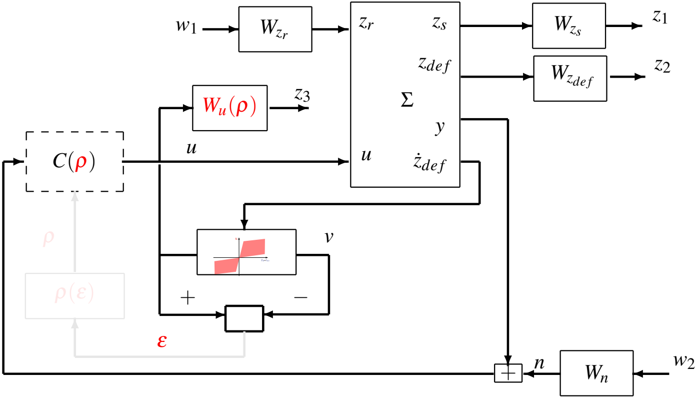

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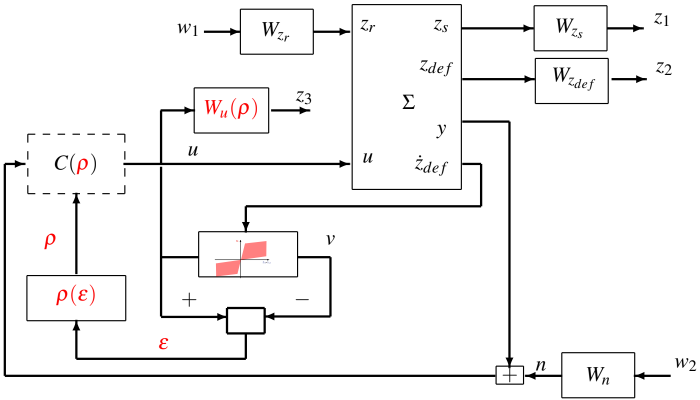
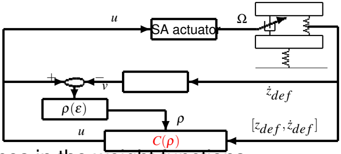


LPV control design (2)

$\Rightarrow W_u(\rho)$ is a ρ-parameter dependent weight



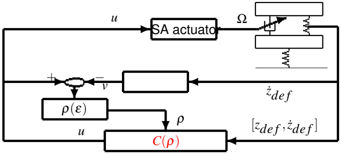
System & Weights

The system is LTI, and the parameter dependency comes in the weight functions. . .

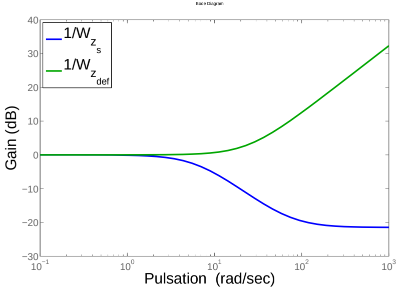
- $W_{z_s} = \frac{\frac{s}{\omega_{11}}+1}{\frac{s}{\omega_{12}}+1}$, chassis performance objective
- $W_{z_{def}} = \frac{1}{\frac{s}{\omega_{21}}+1}$, suspension performance objective
- $W_{z_r} = 7.10^{-2}$, road model
- $W_n = 10^{-4}$, noise model
- $W_u(\rho) = \rho \frac{1}{\frac{s}{1000}+1}$, control attenuation
- $\rho \in \left[\begin{array}{cc} 0.01 & 10 \end{array} \right]$

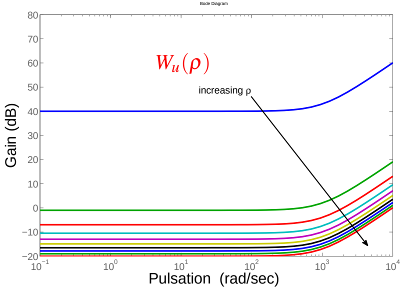
LPV control design (2)

$\Rightarrow W_u(\rho)$ is a ρ-parameter dependent weight



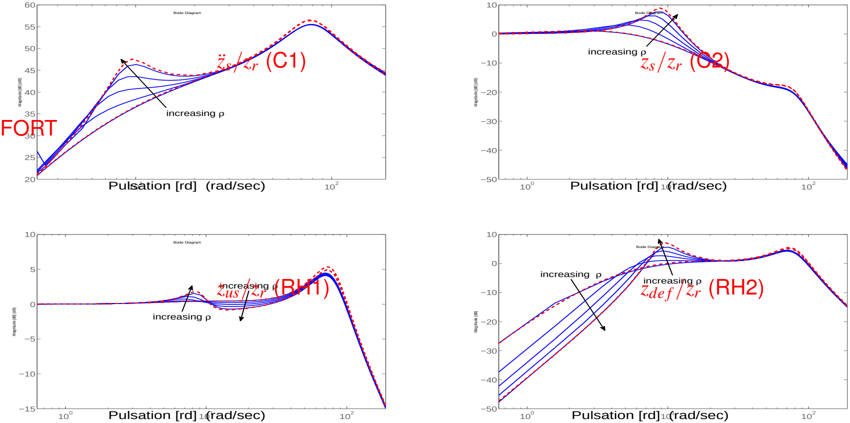
System & Weights





Bode diagram for frozen ρ

COMFORT



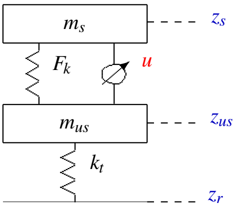
ROAD HOLDING

Performance evaluation on the nonlinear model

Criteria used for evaluation (frequency based)

$$PSD_{\{f_1,a_1\}\to\{f_2,a_2\}}(x) = \sqrt{\int_{f_1}^{f_2}\int_{a_1}^{a_2} x2(f,a)da \cdot df}$$

Performances & PSD metric



- (C1) Comfort at high frequencies: $\ddot{z}_s/z_r$, [4-30]Hz
- (C2) Comfort at low frequencies: z_s/z_r, [0-5]Hz
- (RH1) Road-holding: z_{us}/z_r, [0-20]Hz
- (RH2) Suspension constraints: z_{def}/z_r, [0-20]Hz

Performance evaluation on the nonlinear model

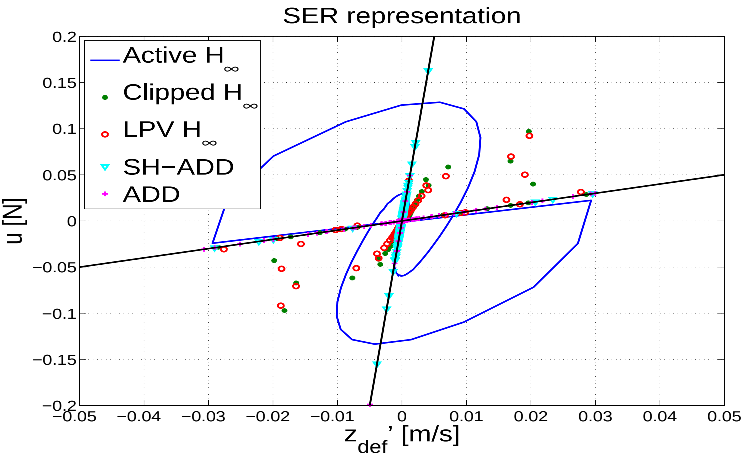
Improvement rate

$$\text{Improvement rate} = \frac{PSD_{passive} - PSD_{controlled}}{PSD_{passive}}$$

Results for nonlinear simulation

Signal	Active $\mathcal{H}_\infty$	Clipped $\mathcal{H}_\infty$	**LPV** $\mathcal{H}_\infty$	ADD	SH-ADD
(C1) $\ddot{z}_s/z_r$ [4-30]Hz	4.8%	3.8%	**-4.4%**	10%	10.8%
(C2) z_s/z_r [0-5]Hz	52.8%	23.5%	**18.9%**	16.9%	36.2%
(RH1) z_{us}/z_r [0-20]Hz	3.2%	4.2%	**9.9%**	-4.9%	-5.8%
(RH2) z_{def}/z_r [0-20]Hz	5.3%	5.7%	**10.4%**	-7.8%	-4.5%

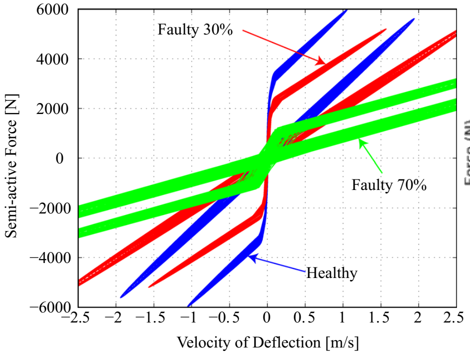
Nonlinear simulation - time

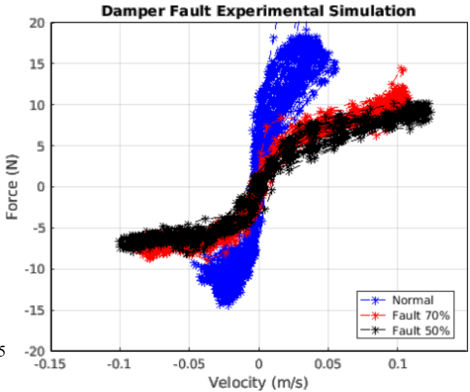


Outline

MR damper and oil leakage effects

Force-Velocity map of a semi-active damper (low and high damping) subject to different leakages.





An oil leakage on a semi-active damper is modelled as:

$$\overline{F}_{damper} = \alpha F_{damper} \tag{14}$$

$\alpha \in [0, 1]$ is the oil leakage degree,

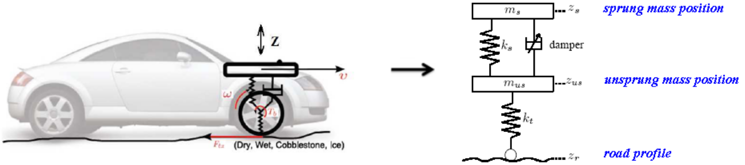


Figure: Simple quarter vehicle model for semi-active suspension control

Quarter vehicle dynamics

- Dynamical equations

$$\begin{cases} m_s \ddot{z}_s &= -k_s z_{def} - F_{damper} \\ m_{us} \ddot{z}_{us} &= k_s z_{def} + F_{damper} - k_t \left(z_{us} - z_r \right) \end{cases} \tag{15}$$

$z_{def} = z_s - z_{us}$: damper deflection, $\dot{z}_{def} = \dot{z}_s - \dot{z}_{us}$: deflection velocity.

- Damper's characteristics : Force / (Deflection-Deflection Velocity) relation

$$F_{damper} = g \left(z_{def}, \dot{z}_{def} \right) \tag{16}$$

where g can be linear or nonlinear.

Estimation of α

Can be tackled using several methods:

Fast Adaptive Fault Estimation

see (Zhang, K., Jiang, B., and Cocquempot, V. (2008)) for additive fault

$$\hat{f}(t) = \Gamma U \left(e_y(t) + \sigma \int_{t_f}^{t} e_y(\tau) d\tau \right) \text{ where } e_y(t) = y(t) - \hat{y}(t). \quad (17)$$

$\rightarrow$ combines a proportional term with an integral one to improve the fault estimation speed.

Use of parity space equation (Sename et al, SYSTOL 2013)

Estimate $\overline{F}_{damper}$ (LTI formulation) and deduce α as: $\alpha \approx \sqrt{\frac{\sum_{i=1}^{N} \widehat{F}_{sa_i}^2}{\sum_{i=1}^{N} F_{sa_i}^2}} \in [0,1]$

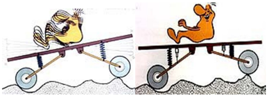
Use of LPV switched observer (Nguyen & Sename & Dugard, IFAC LPVS2015)

- The actuator fault is modeled in a multiplicative way by using a constant coefficient ($\alpha \in [0 \;\; 1]$)
- estimation is based on an LPV extended observer
- The theoretical formulation can be done in several framework for switching (using dwell time or average dwell time characteristics).

Performance objectives

Comfort

linked to the road vibration isolation of the chassis $\rightsquigarrow$ evaluated using the chassis movement: $\ddot{z}_s/z_r$ and z_s/z_r



Road holding

- concerns the wheel rebound (or tire deflection)... keep contact between the wheel and the road
- evaluated using z_{us}/z_r on [0-20]Hz

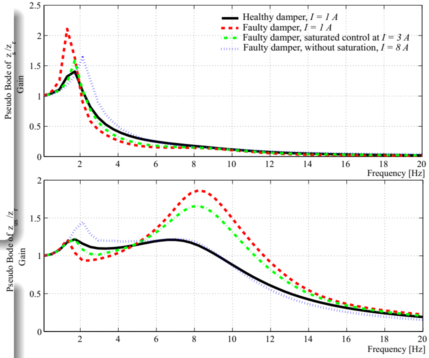
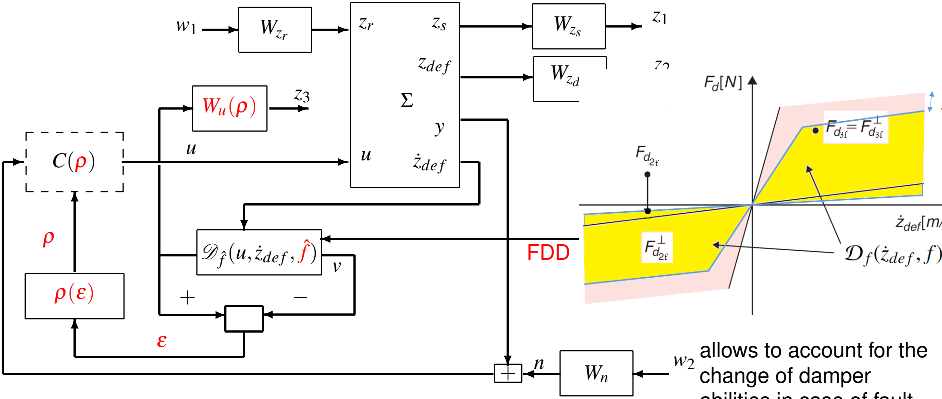


Figure: Semi-active suspension performances at different manipulations, by considering a fault $\alpha = 0.5$.

LPV fault-scheduling suspension control strategy (Method 1)

Idea: design a controller with model 1 ($F_{damper} = c_0\dot{z}_{def} + u$), where the control input $u = u^{\mathscr{H}_\infty}(\rho)$ is limited when the required force is not achievable by the semi-active faulty damper

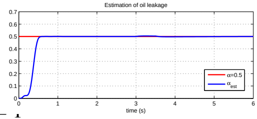


allows to account for the change of damper abilities in case of fault

$W_u(\rho) = \rho \frac{1}{\frac{s}{1000}+1}$ with $\rho(\varepsilon) \in [\ 0.1 \quad 10\ ]$

Time domain analysis (Method 1)

Simulations performed using the quarter-car model with th linear damper simulation model (validated on real data). Scenario: oil leakage, 50% of reduction of the nominal dan force ($\alpha = 0.5$), from $t = 0$. a $3cm$ bump on the wheel from $t = 1$ to $1.5s$.



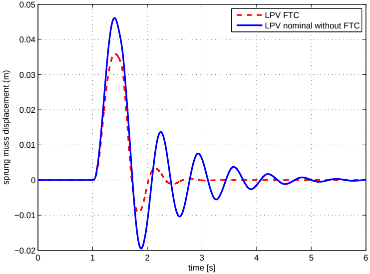


Figure: Comfort performance: Transient response of the sprung mass displacement.

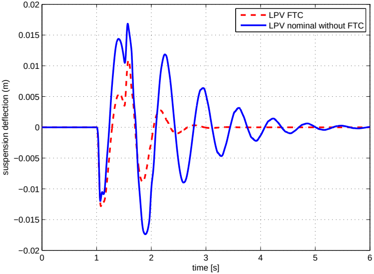


Figure: Road holding performance: Transient response of the relative displacement among the sprung and unsprung mass.

Conclusion about method 1

Interest

- The design model P is a LTI one
- $W_u(\rho)$ is LPV
- The controller is LPV and adapts to the damper capabilities scheduled by the fault-estimation (needed of damper faulty characteristics): a kind of fault tolerant anti-windup
- this adaptation will degrade the CL performances

Drawback

- The closed-loop performances depend on the quality of the characteristic map of the faulty damper (may be conservative).
- Do not handle explicitely the state and input constraints of the suspension system : see Do et al [CDC 2011] or Nguyen et al [CDC 2015] for such a theoretical formulation

LPV fault-scheduling suspension control strategy (Method 2)

Use of a more accurate damper model

- Define two new scheduling parameter $\begin{array}{l} \tanh\left(c_1\dot{z}_{def}+k_1 z_{def}\right)) \longrightarrow \rho_1 \\ I\in[I_{min},I_{max}] \longrightarrow \rho_2 \end{array}$
- The LPV model for the semi-active suspension FTC problem is :

$$\Sigma\begin{cases} \dot{x}=A\left(\rho_1,\rho_2,\alpha\right)x+Bu_c+B_1 w \\ y=Cx \end{cases}$$

where the 3 varying parameters are bounded ($\alpha\in[0,1]$, $\rho_1\in[-1,1]$ and $\rho_2\in[0,1]$)

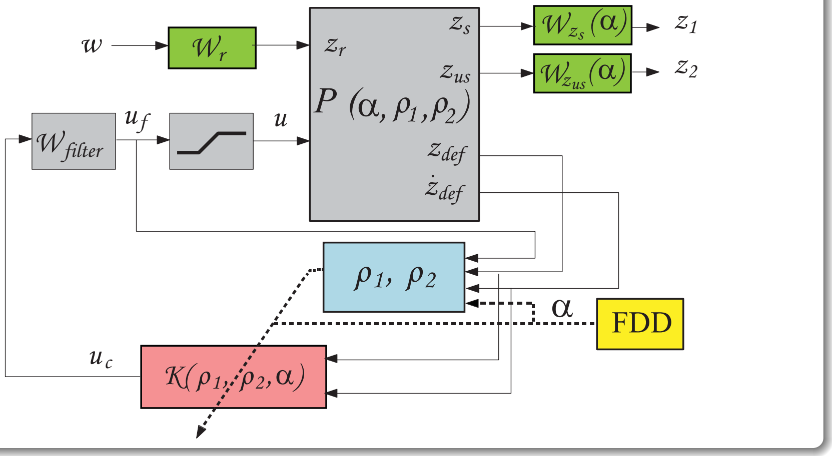
LPV/$\mathscr{H}_\infty$ control design

The generalized plant $P(\theta)$ is the LPV system: $\left[\begin{array}{c} \dot{\xi} \\ \hline z_\infty \\ y \end{array}\right] = \left[\begin{array}{c|cc} \mathscr{A}(\alpha,\rho_1,\rho_2) & \mathscr{B}_1 W_r & \mathscr{B}_2 \\ \hline \mathscr{C}_\infty(\alpha,\rho_1,\rho_2) & 0 & 0 \\ \mathscr{C} & 0 & 0 \end{array}\right]\left[\begin{array}{c} \xi \\ \hline w \\ u_c \end{array}\right]$

where $\xi=[\chi_{vert}\ \chi_w]^T$ with χ_{vert} the states of the LPV *QoV* model and χ_w the weighting functions states, $z_\infty=[z_1\ z_2]^T$, $y=[z_{def}\ \dot{z}_{def}]^T$ and $u_c=u^{\mathscr{H}_\infty}$.

Synthesis: solving the LMI problem for a polytopic set of parameters. The global LPV-FTC is a convex combination of 8 local controllers.

LPV fault-scheduling suspension control strategy (Method 2)

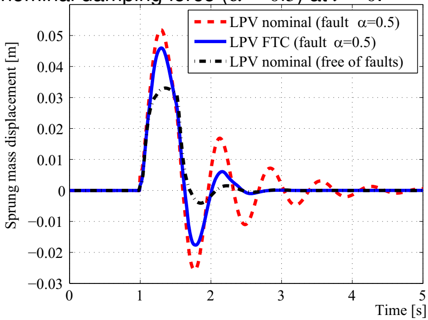


- uses a varying parameter (α) associated to the fault to schedule the suspension actuator work according to new damping characteristics.
- parameter dependent weighting functions allowing to modify on-line the performance specifications according to the state of health of the damper

Time domain analysis (Method 2)

Simulations performed using the quarter-car model with the non linear damper simulation model (validated on real data).

Scenario: 3cm bump on the wheel from $t = 1s$ to $t = 1.5s$. Damper leakage: 50% of reduction of the nominal damping force ($\alpha = 0.5$) at $t = 0$.



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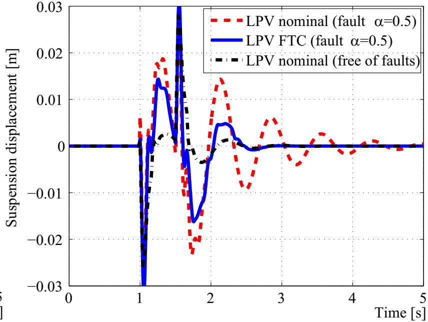


Figure: Road holding performance: Transient response of the relative displacement among the sprung and unsprung mass.

Time domain analysis (Method 2) (cont..)

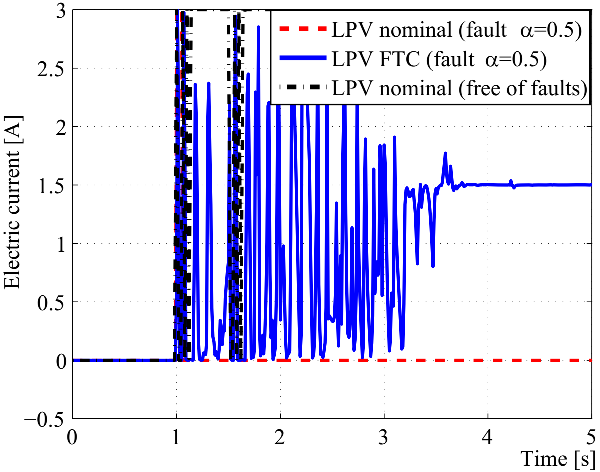


Figure: Controller output in the semi-active suspension.

Conclusion about method 2

Interest

- The design model P is LPV
- The performance weighting functions $W_{z_s}(\alpha)$ and $W_{z_{us}}(\alpha)$ are LPV to adapt the CL performances to the damper capabilities (and accordingly degrade them)

Drawback

- Do not handle explicitely the state and input constraints of the suspension system : see Do et al [CDC 2011] or Nguyen et al [CDC 2015] for such a theoretical formulation
- the controller includes 3 varying parameters which may be a problem for the synthesis and implementation

Outline

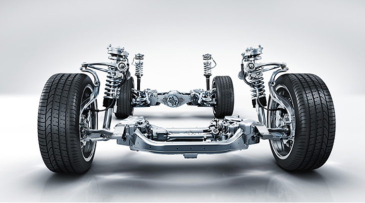
Problem

Suspension system:

Ensure comfort and road holding

Problem: Comfort improvement

Mitigate the body motions (bounce, roll, pitch) induced by road effects using only the suspension actuators.



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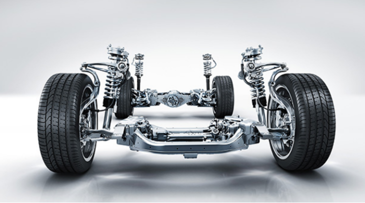

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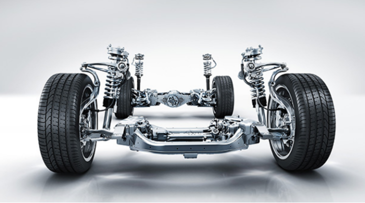
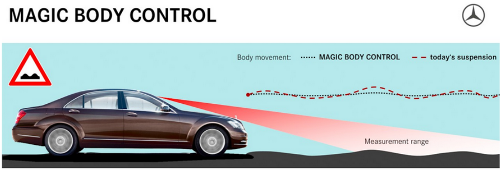


Existing solution:



Problem

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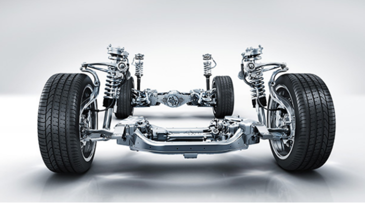
Ensure comfort and road holding



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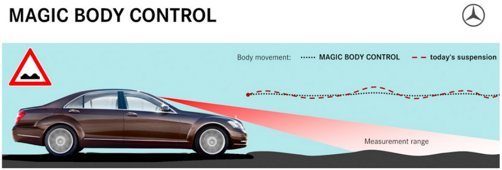
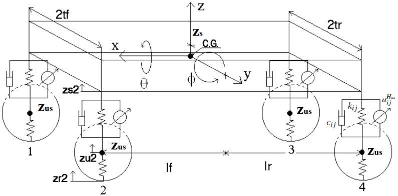


A solution without road preview nor identification: A MIMO controller scheduled according to vehicle motion

Step 1: Motion detection strategy (bounce, roll, pitch)
Step 2: LPV suspension controller design



A 7 dof full vertical vehicle model [$z_s\ \theta\ \phi\ z_{usfl}\ z_{usfr}\ z_{usrl}\ z_{usrr}$]:

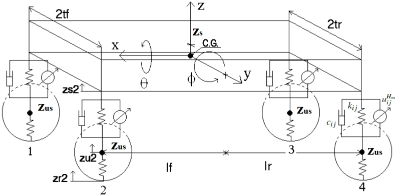
$$\begin{cases} m_s\ddot{z}_s &= -F_{sfl} - F_{sfr} - F_{srl} - F_{srr} + F_{dz} \\ I_x\ddot{\theta} &= (-F_{sfr} + F_{sfl})t_f + (-F_{srr} + F_{srl})t_r + mha_y + M_{dx} \\ I_y\ddot{\phi} &= (F_{srr} + F_{srl})l_r - (F_{sfr} + F_{sfl})l_f - mha_x + M_{dy} \\ m_{us}\ddot{z}_{us_{ij}} &= -F_{s_{ij}} + F_{tz_{ij}} \end{cases} \tag{18}$$

Suspension force:

$$F_{s_{ij}} = k_{ij}(z_{s_{ij}} - z_{us_{ij}}) + c_{ij}(\dot{z}_{s_{ij}} - \dot{z}_{us_{ij}}) + u_{ij}^{H_\infty} \tag{19}$$

Tire force:

$$F_{tz_{ij}} = -k_{t_{ij}}(z_{us_{ij}} - z_{r_{ij}}) \tag{20}$$



A 7 dof full vertical vehicle model $[z_s\ \theta\ \phi\ z_{usfl}\ z_{usfr}\ z_{usrl}\ z_{usrr}]$:

$$\begin{cases} m_s\ddot{z}_s &= -F_{sfl} - F_{sfr} - F_{srl} - F_{srr} + F_{dz} \\ I_x\ddot{\theta} &= (-F_{sfr} + F_{sfl})t_f + (-F_{srr} + F_{srl})t_r + mha_y + M_{dx} \\ I_y\ddot{\phi} &= (F_{srr} + F_{srl})l_r - (F_{sfr} + F_{sfl})l_f - mha_x + M_{dy} \\ m_{us}\ddot{z}_{us_{ij}} &= -F_{s_{ij}} + F_{tz_{ij}} \end{cases} \tag{18}$$

Rewrite (18) in the state space representation form:

$$\dot{x}(t) = Ax(t) + B_1 w(t) + B_2 u \tag{4}$$

where: $x = [z_s\ \theta\ \phi\ z_{usfl}\ z_{usfr}\ z_{usrl}\ z_{usrr}\ \dot{z}_s\ \dot{\theta}\ \dot{\phi}\ \dot{z}_{usfl}\ \dot{z}_{usfr}\ \dot{z}_{usrl}\ \dot{z}_{usrr}]^T$,

$$w = [F_{dz}\ M_{dx}\ M_{dy}\ z_{rfl}\ z_{rfr}\ z_{rrl}\ z_{rrr}]^T\ ,\ u = [u_{fl}^{H_\infty}, u_{fr}^{H_\infty}, u_{rl}^{H_\infty}, u_{rr}^{H_\infty}]^T.$$

The vehicle dynamic scheme with semi-active suspension:

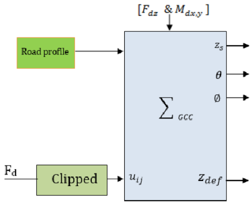
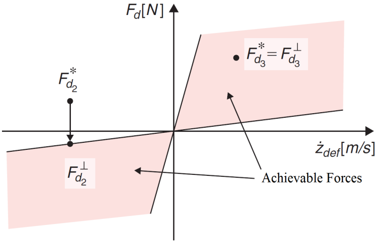


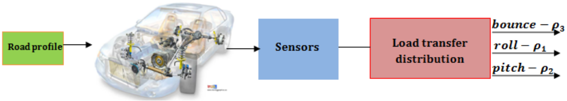
Figure: Vehicle dynamic scheme

Semi-active suspension using the "clipped strategy":



Motion detection based on the load transfer distribution

The motion detection is based on the calculation of the load transfer distribution coefficients. The coefficients are used as the scheduling parameters of the weighting functions (detailed in the sequel).



Roll monitoring by the lateral load transfer (ρ_1)

$$\rho_1 = \left|\frac{(F_{z_l} - F_{z_r})}{(F_{z_l} + F_{z_r})}\right| \qquad (a)$$

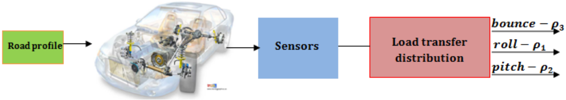
with:

$$\begin{cases} F_{z_l} = m_s \times \frac{g}{2} + m_s \times h \times \frac{a_y}{l_f} \\ F_{z_r} = m_s \times \frac{g}{2} - m_s \times h \times \frac{a_y}{l_r} \end{cases} \qquad (19)$$

where F_{z_l} and F_{z_r} are the vertical forces, a_y is the lateral acceleration. When $\rho_1 \rightarrow 0$, no lateral load transfer, no roll motion.
Conversely when $\rho_1 \neq 0$, the vehicle is in roll motion.

Motion detection based on the load transfer distribution

The motion detection is based on the calculation of the load transfer distribution coefficients. The coefficients are used as the scheduling parameters of the weighting functions (detailed in the sequel).



Pitch monitoring by the longitudinal load transfer (ρ_2)

$$\rho_2 = \left| \frac{(F_{z_f} \times \frac{L}{l_r} - F_{z_r} \times \frac{L}{l_f})}{(F_{z_f} \times \frac{L}{l_r} + F_{z_r} \times \frac{L}{l_f})} \right| \qquad (b)$$

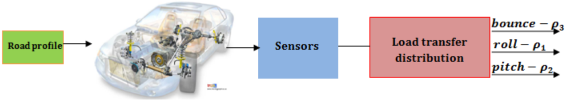
where the front and rear forces are given by:

$$\begin{cases} F_{z_f} = m_s \times (\frac{l_r}{L}.\cos(\phi) + \frac{h}{L}.\sin(\phi)) - m_s \times a_x \times \frac{h}{L} \\ F_{z_r} = m_s \times (\frac{l_f}{L}.\cos(\phi) - \frac{h}{L}.\sin(\phi)) + m_s \times a_x \times \frac{h}{L} \end{cases} \qquad (19)$$

where a_x is the longitudinal acceleration. When $\rho_2 \to 0$, no longitudinal load transfer, no pitch motion.
Conversely when $\rho_2 \neq 0$, the pitch motion is detected.

Motion detection based on the load transfer distribution

The motion detection is based on the calculation of the load transfer distribution coefficients. The coefficients are used as the scheduling parameters of the weighting functions (detailed in the sequel).



Bounce monitoring (ρ_3)

Thanks to the lateral and longitudinal load transfers, one defines ρ_3 as:

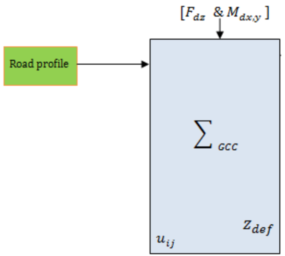
$$\rho_3 = |(1 - \rho_1 - \rho_2)| \qquad (c)$$

When $\rho_3 \neq 0$, the bounce motion is taken into account.

ρ_1, ρ_2, ρ_3 are varying parameters for the LPV system.

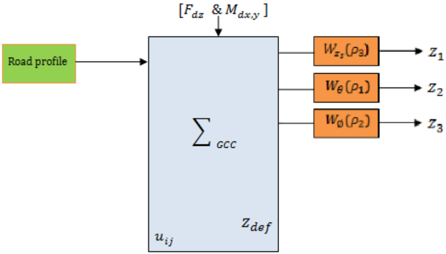
General structure for the LPV suspension controller:

LTI system + parameter dependant weighting functions = generalized LPV system



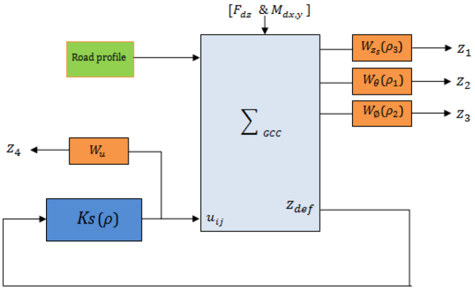
General structure for the LPV suspension controller:

LTI system + parameter dependant weighting functions = generalized LPV system



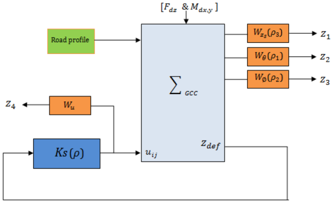
General structure for the LPV suspension controller:

LTI system + parameter dependant weighting functions = generalized LPV system



The controller K is synthesized in the H_∞/LPV framework

Design of the LPV control in the H_∞/LPV framework:



The use of parameter dependant weighting functions allows to modify on-line the performance specifications according to the vehicle motion:

- $W_{z_s}(\rho_3) = \rho_3 \frac{3}{s/(2\pi f_1)+1}$: $\Rightarrow$ bounce motion
- $W_\theta(\rho_1) = \rho_1 \frac{2}{s/(2\pi f_2)+1}$: $\Rightarrow$ roll motion
- $W_\phi(\rho_2) = \rho_2 \frac{2}{s/(2\pi f_3)+1}$: $\Rightarrow$ pitch motion

Remember that: $\rho_1, \rho_2, \rho_3 \in [0 \quad 1]$
Ex: if $\rho_3 \longrightarrow 1$, the gain of the weighting function is high, bounce (z_s) is penalized.
Conversely, when $\rho_3 \longrightarrow 0$, bounce motion is not limited.
Other weighting functions (for actuator constraints)

- $W_u = 10^{-2}$: avoids too large control signals.
- $W_{z_{rij}} = 3.10^{-2}$: shapes the road profiles (z_{rij}).

Simulation results

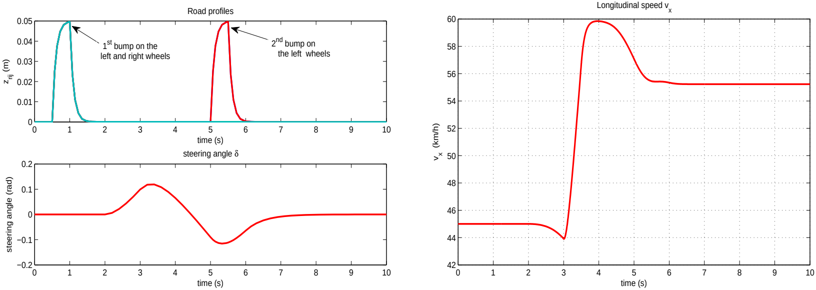
INOVE Automotive Toolbox at GIPSA Lab - Grenoble INP.
Using a full nonlinear vehicle model, validated on a real car "Renault Mégane Coupé " coll. MIPS lab [Basset]

Symbol	Value	Unit	Signification
m_s	350	kg	suspended mass
$m_{us_{fj}}$	35	kg	front unsprung mass
$m_{us_{rj}}$	32.5	kg	rear unsprung mass
$I_x;I_y;I_z$	250; 1400; 2149	$kg.m^2$	roll, pitch, yaw inertia
I_w	1	$kg.m^2$	wheel inertia
$t_f;t_r$	1.4; 1.4	m	front, rear axle
$l_f;l_r$	1.4; 1	m	COG-front, rear distance
R	0.3	m	nominal wheel radius
h	0.4	m	chassis height
k_{fj}	29500	N/m	front suspension stiffness
k_{rj}	20000	N/m	rear suspension stiffness
c_{fj}	1500	$N/m/s$	front suspension damping
c_{rj}	3000	$N/m/s$	rear suspension damping
$k_{t_{ij}}$	208000	N/m	tire stiffness
$c_{t_{ij}}$	10	$N/m/s$	tire damping
b_t	8.3278	—	lateral tire parameter
c_t	1.1009	—	lateral tire parameter
d_t	2268	—	lateral tire parameter
e_t	-1.1661	—	lateral tire parameter
g	9.81	m/s^2	gravitational constant

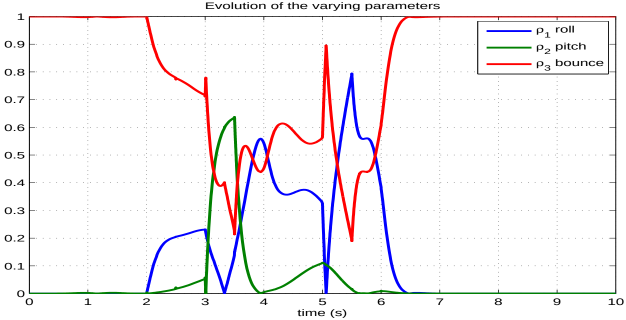
TABLE I
RENAULT MÉGANE COUPÉ PARAMETERS

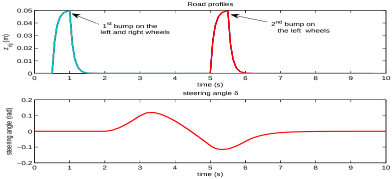
[Simulation scenario:]

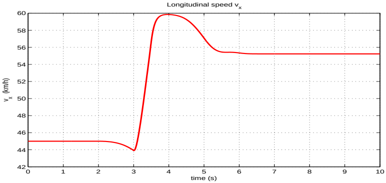
- The vehicle runs at $45 km/h$, and accelerates from $t = 3s$ to $t = 3.5s$ which induces a pitch motion (traction induced pitch motion).
- A $5cm$ bump occurs simultaneously on the left and right wheels (from $t = 0.5s$ to $t = 1s$) to excite the bounce motion.
- A double lines change is performed from $t = 2s$ to $t = 6s$).
- And a $5cm$ bump on the left wheels (from $t = 5s$ to $t = 5.5s$) during the manoeuvre that causes the roll vibration



Motion detection and scheduling parameters:

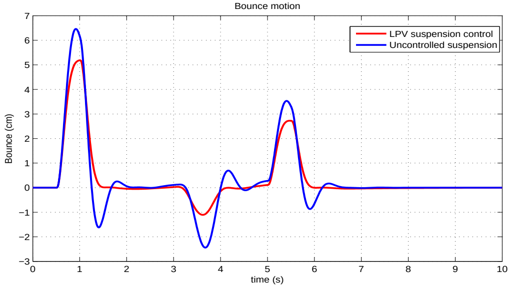




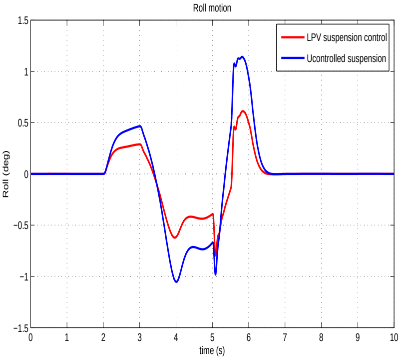


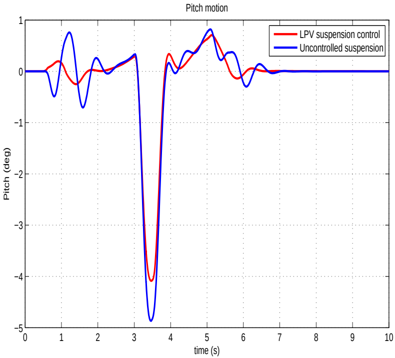
Comparison of:

- H_∞/LPV: Renault Mégane Car equipped with a semi-active suspension controlled by the proposed methodology.
- Passive (Uncontrolled suspension): Renault Mégane Car equipped with an optimized nonlinear passive suspension ($u_{ij}^{H_\infty} = 0$).



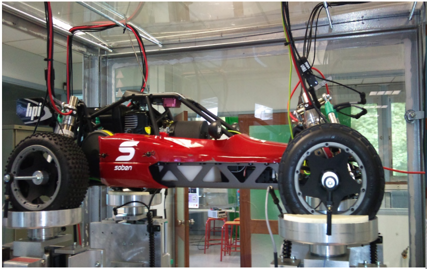
⇒ Bounce motion in the controlled case is reduced w.r.t the uncontrolled case.





$\Rightarrow$ Mitigation of the roll and pitch motions thanks to the motion detection strategy, associated to the gain scheduling LPV controller.

INOVE testbed



Studies

- Modelling of semi-active dampers
- Modelling of the 7 DOF vertical dynamics
- Estimation of the road impact and observation of the vertical dynamics state variables
- Detection of sensors and actuators faults using observers and parity equations methods
- Control of semi-active dampers and of the vertical dynamics using Linear Parameter Varying approaches

Equipment

- 4 semi-active Electro-Rheological dampers
- independent road profile, 4 DC motors
- Sensors: an inertial measurement unit, 4 accelerometers for the wheel vertical behaviors, 4 suspension deflection sensors, pitch and yaw angle sensors, 4 force sensors at the tyre/road contact; 4 ER damper force sensors.

INOVE test

Monitoring parameters

Fact: in the testbed lateral acceleration is very small and the roll motion, detected by the load transfer approach, may be then neglected.
Alternative solution: take into account the differences between suspension deflections at the left and right sides (roll) and at front and rear axle (pitch)

- **Roll monitoring parameter:**

$$\rho_1 = \left| \frac{(z_{def_{fl}} + z_{def_{rl}}) - (z_{def_{fr}} + z_{def_{rr}})}{\left|z_{def_{fl}}\right| + \left|z_{def_{rl}}\right| + \left|z_{def_{fr}}\right| + \left|z_{def_{rr}}\right|} \right| \tag{19}$$

- **Pitch monitoring parameter:**

$$\rho_2 = \left| \frac{(z_{def_{fl}} + z_{def_{fr}}) - (z_{def_{rl}} + z_{def_{rr}})}{\left|z_{def_{fl}}\right| + \left|z_{def_{rl}}\right| + \left|z_{def_{fr}}\right| + \left|z_{def_{rr}}\right|} \right| \tag{20}$$

- **Bounce monitoring parameter:** For the bounce motion supervision, one still chooses as previously, i.e:

$$\rho_3 = \frac{2 - \rho_1 - \rho_2}{2} \tag{21}$$

Implemented Control scheme

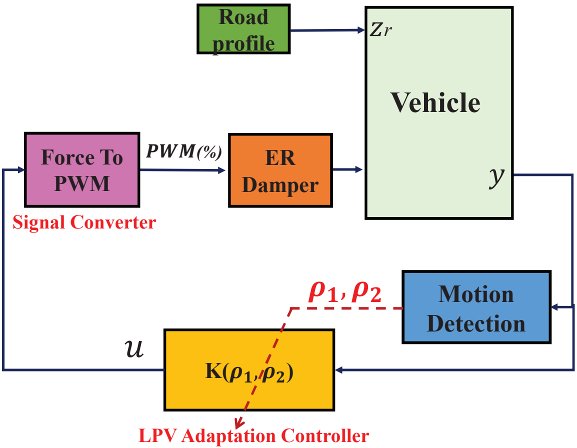
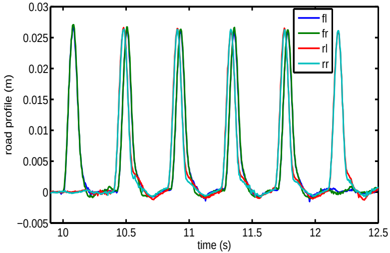


Figure: Suspension control Implementation scheme

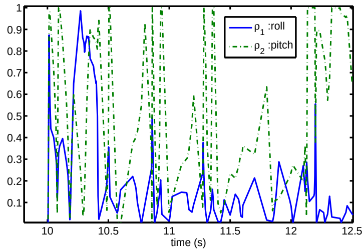
Experimental scenario 1: bumps

In order to assess the LPV controller, a five consecutive bumps road profile is send to the four corners of the car (delayed phases between the front and rear corners) that induce the pitch motion

Five bumps road profile



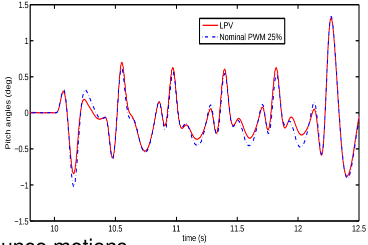
Motion detection: roll and pitch motions



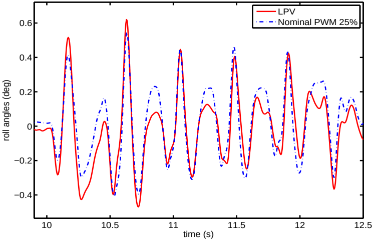
It emphasizes the motion detection of the SOBEN car, in particular the pitch motion which is the main motion of the car in this excitation.

Experimental scenario 1: bumps

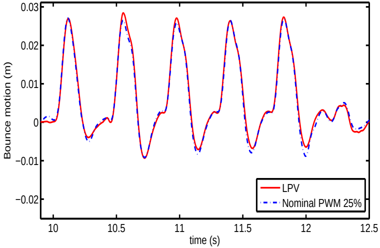
Pitch motions



Roll motions



Bounce motions

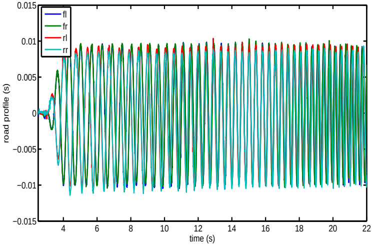


Promising results but need to include the damper modelling in control

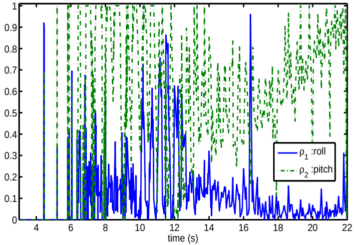
Experimental scenario 2: chirp

The SOBEN Car is excited by a chirp signal road profile from 0-3Hz (also delayed phases between the front and rear corners):

Five bumps road profile



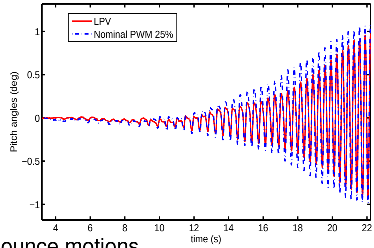
Motion detection: roll and pitch motions



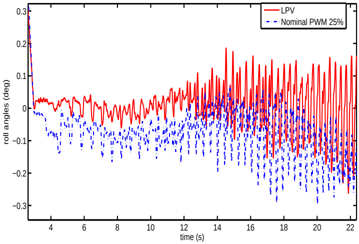
It emphasizes the motion detection of the SOBEN car that is quite perturbed

Experimental scenario 2: chirp

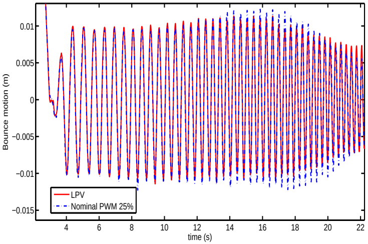
Pitch motions



Roll motions



Bounce motions



Promising results but need to include the damper modelling in control

Outline

Conclusions

About today's presentation:

Many interests of the LPV approach

- Modelling of complex systems (but still less than nonlinear formulation)
- Control design with varying performances, ensuring internal stability and robust-like performances
- LPV Observer/Filter design... for FDI
- A tool to design adaptive FTCS
- Can be extended to mixed-objectives problems (e.g$\mathscr{H}_\infty$, $\mathscr{H}_2$...) through LMI (and/or nonsmooth) tools

With grateful thanks to ...


Juan-Carlos Tudon


Charles Poussot


Anh-Lam DO


Manh Quan Nguyen


and of course to our colleague
Dr Luc Dugard


Thank you for your attention


Merci pour votre attention

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