Semi-active SOBEN suspension modeling and control

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Summary

Summary

The industrial context

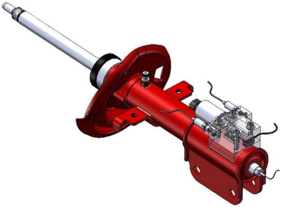
Manufacturing

- Damper prototypes
- Mass-production
- Helicopter undercarriages
- Testing benches

Tests & Validation

- Testing benches
- Testing cars
- Sensors & acquisition boards

Research & Development

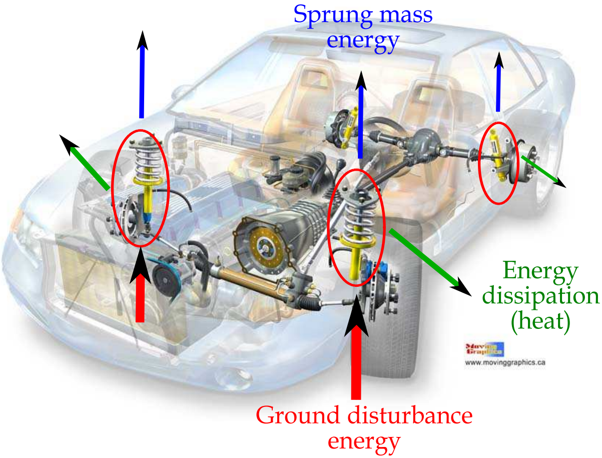


- Dampers for private and racing car
- Adjustable motorcycle dampers
- Helicopter undercarriages
- Semi-active dampers development

The collaboration context

- **SLR team - GIPSA-lab**
  - Semi-active suspension modeling and control
    - [Sammier, 2001, PhD. thesis]
  - Active suspension control toward Global Chassis Control
    - [Zin, 2005, PhD. thesis]
  - Robust multivariable LPV automotive Global Chassis Control
    - [C. Poussot-Vassal, 2008, PhD. thesis]
  - Robust vehicle dynamics LPV control : comfort and safety improvement
    - [A. Lam Do, PhD. thesis] (in progress)
- **TEC Monterrey (PCP program - 6 months visit)**
  - Magneto-rheological (MR) dampers modeling
  - Semi-active damper control

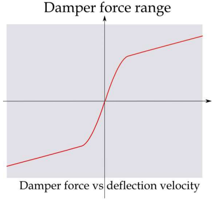
The role of suspensions



- Spring
  - Chassis holding
- Damper
  Minimize :
  - sprung mass vibrations
    - ▶ Comfort
  - wheel displacements
    - ▶ Roadholding

Various suspension types

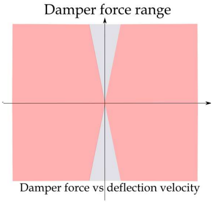
Passive



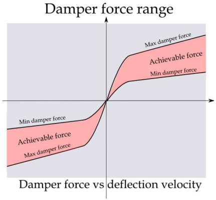
- Most widely used suspension type
- Simple and economical
- Can be optimized ▶ CRONE approach [Moreau, 1995], [Oustaloup et Mathieu, 1999]

Active

- Widely studied, rarely used [Hrovat, 1997], [Karnopp, 1983], [Sammier, 2003]
- Expensive, high power consumption, high performance



Semi-active



- Under development, already used
- Moderate cost and power consumption, improved performances

Previous studies on semi-active suspension control

- Clipped semi-active suspension control
  - LQ : complex, state measurement
    - [Tseng et al., 1994]
  - MPC based : involve optimization, state measurement
    - [Canale et al., 2006], [Giorgetti et al., 2006], [Sammier et al., 2003]
  - $\mathcal{H}_\infty$
    - [Sammier et al., 2003]
- Dedicated semi-active suspension control
  - $\mathcal{H}_\infty$/LPV control
    - [Poussot-Vassal, 2008], [Do et al., 2010]
  - ADD, Mixed SH-ADD : easily implementable, deflection velocity measurement, comfort oriented
    - [Savaresi et al., 2005, 2007]

PhD contributions

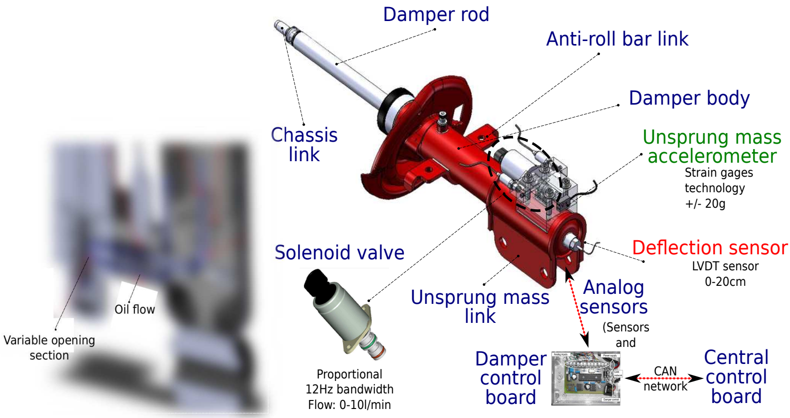
- Control oriented damper models
- $\mathcal{H}_\infty$ observer
  - Unknown input observer
  - Few sensors required
- LPV/$\mathcal{H}_\infty$ suspension control
  - Local(damper) and global(vehicle) control
  - Adjustable performance
  - Damper non linearities/abilities (LPV)
- System development
  - Sensors and actuator choice
  - Vehicle preparation (cables, sensors, dampers, control boards)
  - Control boards programing

Publications

- Conference papers
  - Vehicle modeling and analysis
    - 17th IFAC World Congress 2008
    - 7th EUROSIM Congress on modeling and simulation (co-author)
    - "Journées Automatique et Automobile" (JAA GRDMACS) 2007
  - Observer design methodology
    - IEEE Multi-Conference on Systems and Control 2009
    - IFAC Symposium Advances in Automotive Control 2010
  - $\mathcal{H}_\infty$/LPV control strategy
    - European Control Conference 2009
    - 11th Mini Conference on Vehicle System Dynamics, Identification and Anomalies 2008
- Journal papers in preparation
  - "Full vertical car observer design methodology for on-board suspension control applications"
  - "Semi-active $\mathcal{H}_\infty$/LPV control for a full vertical car equipped with industrial hydraulic dampers"

Summary

System description

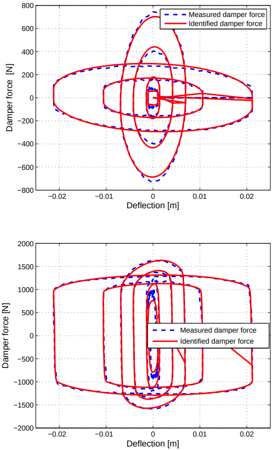


Identified damper model

- Quasi-static damper behavior [Guo et al., 2006]
  $F(t) = (A_1 u_d + A_2)\tanh(A_3 \dot{z}_{def} + A_4 z_{def}) + A_5 \dot{z}_{def} + A_6 z_{def} + A_7$
  - Constant parameters A_i identified from experimental data
  - Damper non-linearity & hysteresis
- Dynamical damper behavior

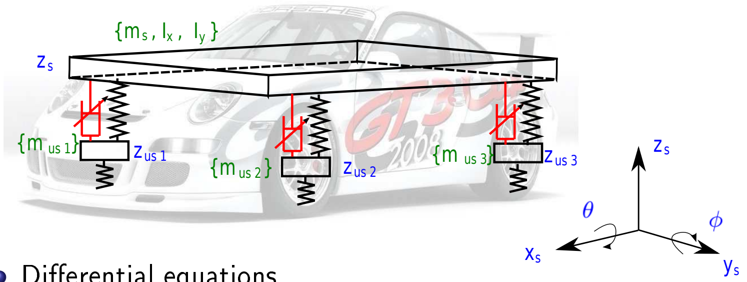
$$\frac{I(s)}{u_d(s)} = \frac{G}{\left(\frac{s}{\omega_d}\right)^2 + 2\zeta_d \frac{s}{\omega_d} + 1}$$

  - Actuator bandwidth



The vehicle model

- Vertical full-car model (7 DOF)



- Differential equations

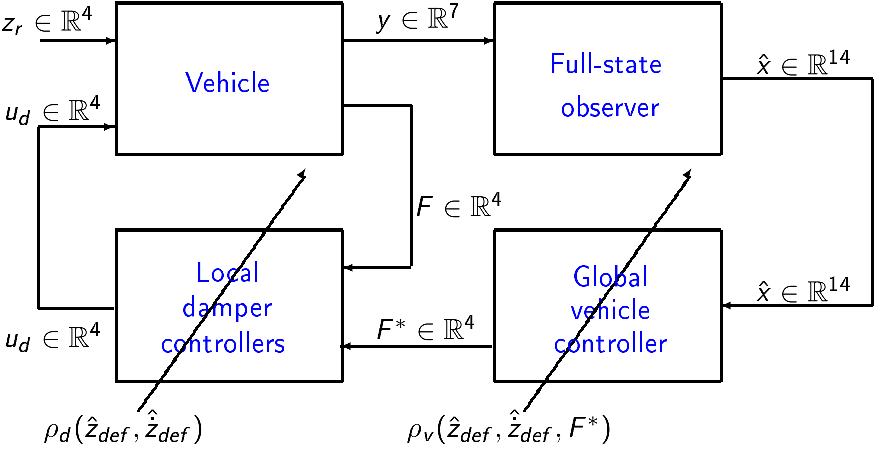
$$\begin{cases} m_s \ddot{z}_s = -(F_{s_1} + F_{s_2} + F_{s_3} + F_{s_4}) \\ m_{us_i} \ddot{z}_{us_i} = F_{s_i} - F_{t_i}, i = 1, .., 4 \\ I_x \ddot{\theta} = (F_{s_1} - F_{s_2}) t_f + (F_{s_3} - F_{s_4}) t_r \\ I_y \ddot{\phi} = (F_{s_4} + F_{s_3}) l_r - (F_{s_2} + F_{s_1}) l_f \end{cases}$$

$$\begin{cases} F_{si} = k_i z_{def_i} + F(z_{def}, \dot{z}_{def}, u_i) \\ F_{ti} = k_{t_i} (z_{us_i} - z_{r_i}) \\ z_{def_i} = z_{s_i} - z_{us_i} \end{cases}$$

$F(z_{def}, \dot{z}_{def}, u_i)$ linear for synthesis, nonlinear for simulations

Summary

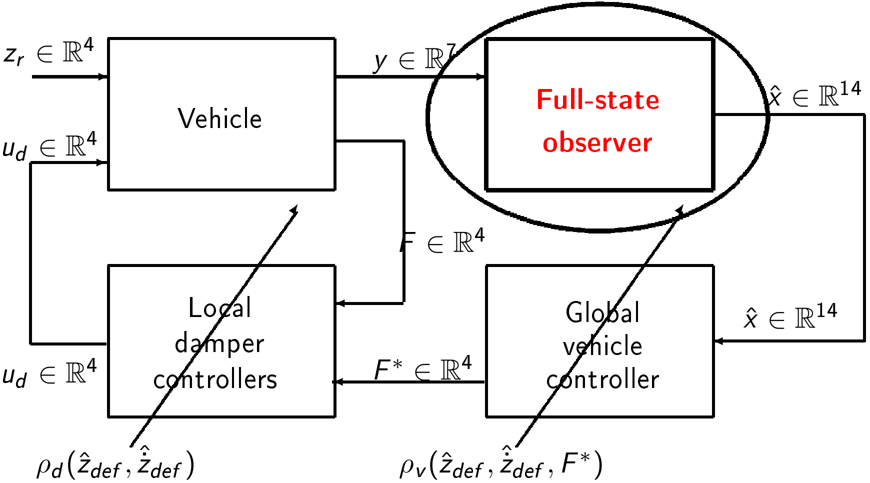
Hierarchical control architecture



Architecture characteristics

- Full-state observer
  - To reduce the number of sensors
- Global vehicule controller
  - $\mathcal{H}_\infty$ LPV state-feedback controller
  - LPV to ensure dissipativity constraint
  - Damper force reference generation for each damper
  - Comfort/roadholding trade-off optimization
- Local damper controllers
  - Mixed $\mathcal{H}_\infty/\mathcal{H}_2$ LPV dynamic output feedback controller
  - LPV to account for damper nonlinearities

Control architecture



Full vertical car observer [Koenig, 2008]

Measured variables :

4 unsprung mass accelerations, 3 sprung mass accelerations

Full-car linear model :

$$\begin{cases} \dot{x} = Ax + D_x v \\ y = Cx + D_y v \end{cases}$$

Full-state observer :

$$\begin{cases} \dot{z} = Nz + Ly \\ \hat{x} = z - Ey \end{cases}$$

$$\dot{e} = \dot{x} - \dot{\hat{x}} = Ne + (PA - KC - N)x + (PD_x - LD_y)v + ED_y\dot{v}$$

Where $P = \mathcal{I} + EC$ and $K = L + NE$.

Stability conditions :

$$\begin{cases} N \text{ is Hurwitz} \\ N = PA - KC \end{cases}$$

Decoupling conditions :

$$\begin{cases} ED_y = 0 \\ LD_y - PD_x = 0 \end{cases}$$

$LD_y - PD_x = 0$ may not be solvable or lead to inconvenient pole placement $\Rightarrow$ $\mathcal{H}_\infty$-observer $\Rightarrow$ N, L and E

$\mathcal{H}_\infty$ observer synthesis

- Method
  1. E parameterized s.t. $ED_y = 0$
  2. $N = PA - KC$

$$\dot{e} = Ne + (PD_x - LD_y)v \Leftrightarrow \dot{e} = \mathcal{A}e + \mathcal{B}v$$

- $\mathcal{H}_\infty$ disturbance attenuation

  Application of the Bounded Real Lemma [Scherer and Weiland, 1999]

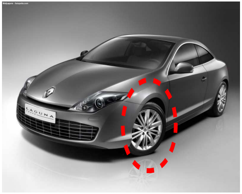
$$\left(\begin{array}{c|c|c} \mathcal{A}^T P + P\mathcal{A} & P\mathcal{B} & \mathcal{C}^T \\ \hline * & -\gamma\mathcal{I} & \mathcal{D}^T \\ \hline * & * & -\gamma\mathcal{I} \end{array}\right) \prec 0$$

- + Observer pole placement [Chilali et al., 1999]

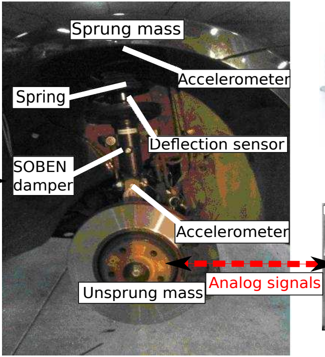
$$\mathcal{D} = \left\{z \in \mathcal{C} : L + zM + z^* M^T \prec 0\right\}$$

$$\left(\begin{array}{c|c|c} L \otimes X + M \otimes (X\mathcal{A}) & M_1^T \otimes (X\mathcal{B}) & M_2^T \otimes C^T \\ +M^T \otimes (\mathcal{A}^T X) & & \\ \hline * & -\gamma\mathcal{I} & \mathcal{D}^T \\ \hline * & * & -\gamma\mathcal{I} \end{array}\right) \prec 0$$

Experimental set-up and measured variables

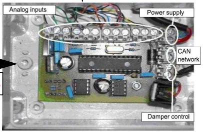


SOBEN testing car
(Renault Laguna GT)



Embedded observer

CAN network

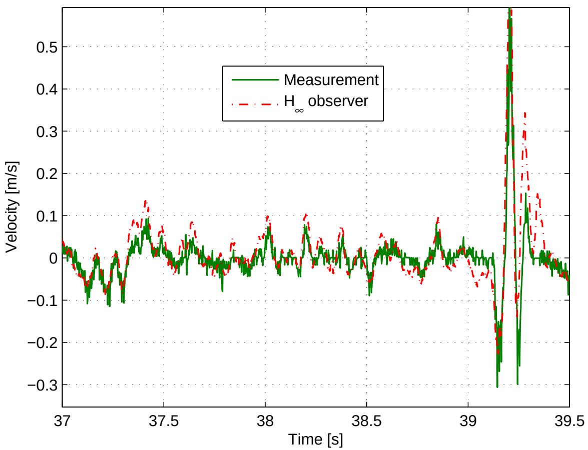


Acquisitions &
Valve power supply

Notation	Description	Full scale	Used in...
$\ddot{z}_{us_{1,2,3,4}}$	Wheel vertical accelerations	+/- 50g	▶ Observer input
$\ddot{z}_{s_{1,2,3}}$	Chassis vertical acceleration	+/- 5g	▶ Observer input
$z_{def_{1,2,3,4}}$	Front left suspension deflection	0-20cm	▶ Validation
$\dot{\theta}, \dot{\phi}$	Roll and pitch velocities	0-150deg/s	▶ Validation
$F_{1,2,3,4}$	Damper forces	Confidential	▶ Local damper control

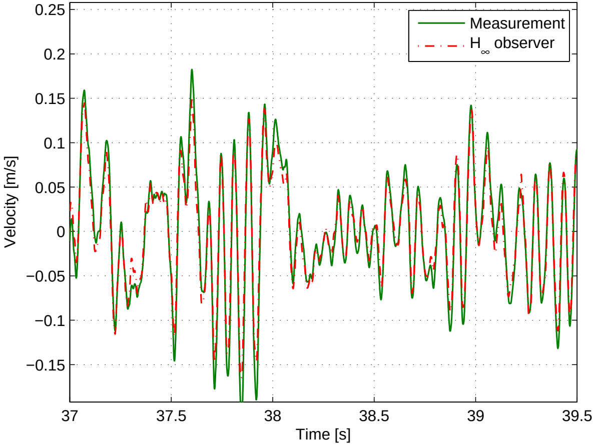
Real time estimation results from experiments

Deflection velocity (front left)



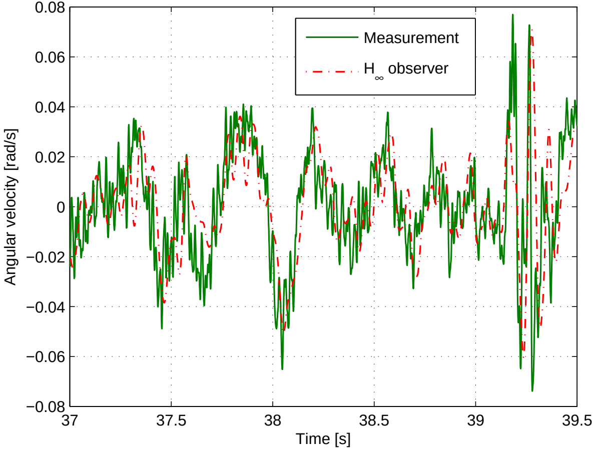
Real time estimation results from experiments

Unsprung mass velocity (front left)

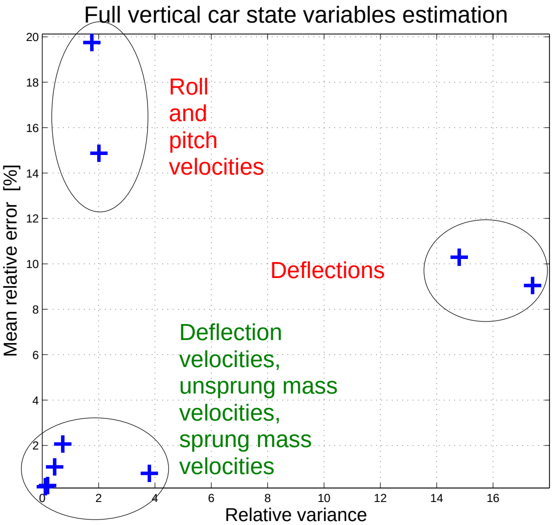


Real time estimation results from experiments

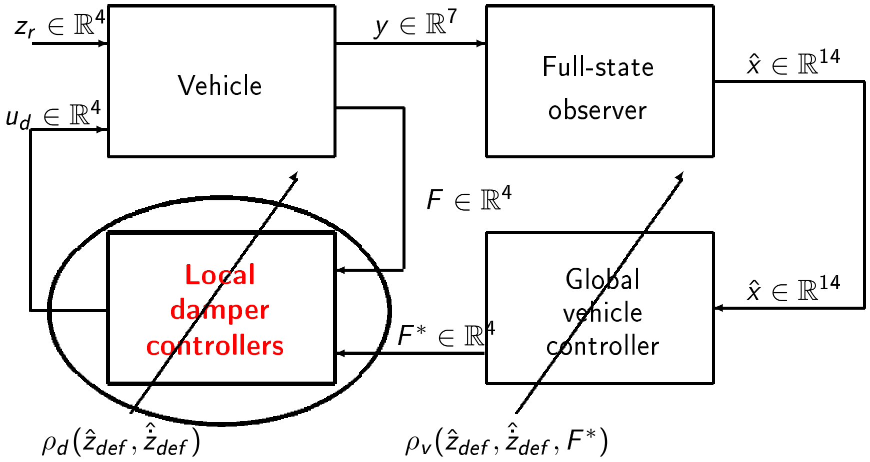
Pitch angular velocity



Real time estimation results from experiments



Control architecture



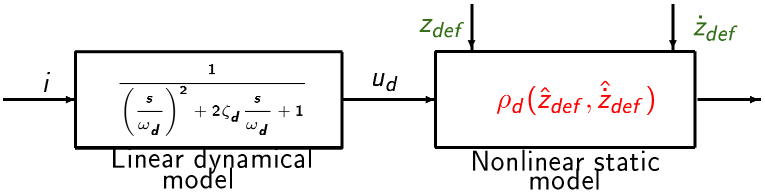
LPV damper model

- Static damper model ▶ Nonlinear static gain

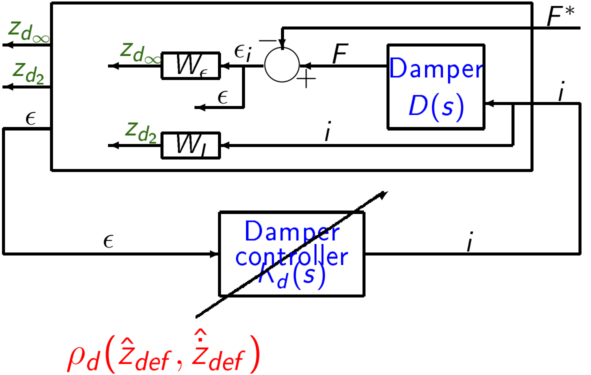
$$F = (A_1 u_d + A_2)\tanh(A_3\dot{z}_{def} + A_4 z_{def}) + A_5\dot{z}_{def} + A_6 z_{def} + A_7$$
$$= \rho_d(z_{def}, \dot{z}_{def})u_d + F_0(z_{def}, \dot{z}_{def})$$

- Damper dynamical behavior ▶ Bandwidth characterization

$$D(s) = \frac{\rho_d(z_{def}, \dot{z}_{def})}{\left(\frac{s}{\omega_d}\right)^2 + 2\zeta_d\frac{s}{\omega_d} + 1}$$



Controller synthesis



Mixed $\mathcal{H}_\infty/\mathcal{H}_2$ LPV synthesis

- Tracking performances ▶ $\mathcal{H}_\infty$ constraint
- Control signal energy bound ▶ $\mathcal{H}_2$ constraint
- Stability of the closed-loop LPV system

Damper controller synthesis

- Damper model

$$D(s) = \frac{\rho_d(z_{def}, \dot{z}_{def})}{\left(\frac{s}{\omega_d}\right)^2 + 2\zeta_d \frac{s}{\omega_d} + 1}$$

- Generalized system

$$\begin{pmatrix} \dot{x}_{d_a} \\ z_{d_\infty} \\ z_{d_2} \\ y \end{pmatrix} = \left(\begin{array}{c|c|c} A_{d_a} & B_1 & B_2(\rho_d) \\ \hline C_\infty & D_{\infty 1} & D_{\infty 2} \\ \hline C_2 & D_{21} & D_{22} \\ \hline C_y & D_{y1} & \mathcal{O} \end{array}\right) \begin{pmatrix} x_{d_a} \\ F^* \\ i \end{pmatrix}$$

- LPV controller

$$K_d(\rho_d) : \begin{pmatrix} \dot{\zeta} \\ i \end{pmatrix} = \left(\begin{array}{c|c} A_{K_d}(\rho_d) & B_{K_d}(\rho_d) \\ \hline C_{K_d}(\rho_d) & \mathcal{O} \end{array}\right) \begin{pmatrix} \zeta \\ \epsilon \end{pmatrix}$$

- Closed-loop system

$$\begin{pmatrix} \dot{x}_{cl} \\ z_{d\infty} \\ z_{d2} \end{pmatrix} = \left(\begin{array}{c|c} \mathcal{A}(\rho_d) & \mathcal{B} \\ \hline \mathcal{C}_1 & \mathcal{D}_1 \\ \hline \mathcal{C}_2 & \mathcal{D}_2 \end{array}\right) \begin{pmatrix} x_{cl} \\ F^* \end{pmatrix}$$

Synthesis inequalities

- $\mathcal{H}_\infty$ performance for LPV systems (Bounded Real Lemma) [Apkarian et al., 1995], [Scherer and Weiland, 2005]

$$\begin{pmatrix} \mathcal{A}^T(\rho_d)P + P\mathcal{A}(\rho_d) & P\mathcal{B} & \mathcal{C}_\infty{}^T \\ * & -\gamma_0\mathcal{I} & \mathcal{D}_\infty{}^T \\ * & * & -\gamma_0\mathcal{I} \end{pmatrix} \prec 0$$

- $\mathcal{H}_2$ performance for LPV systems

$$\begin{pmatrix} \mathcal{A}^T(\rho_d)P + P\mathcal{A}(\rho_d) & P\mathcal{B} \\ * & -\mathcal{I} \end{pmatrix} \prec 0, \begin{pmatrix} K & \mathcal{C}_2^T \\ * & Z \end{pmatrix} \succ 0, \; Tr(Z) > \sigma_0$$

- Pole placement constraints

$$\mathcal{D} = \left\{ z \in \mathcal{C} : L + zM + z^* M^T \prec 0 \right\}$$

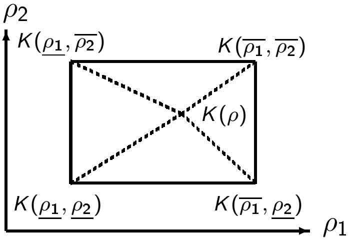
$$\begin{pmatrix} L \otimes X + M \otimes (X\mathcal{A}(\rho_d)) + M^T \otimes (\mathcal{A}^T(\rho_d)X) & M_1^T \otimes (X\mathcal{B}) & M_2^T \otimes C_1^T \\ * & -\gamma\mathcal{I} & \mathcal{D}_1^T \\ * & * & -\gamma\mathcal{I} \end{pmatrix} \prec 0$$

LPV control - Polytopic approach

- Polytopic approach [Apkarian et al., 1995], [Scherer and Weiland, 2005]
  - Extremal parameters ▶ polytope
  - Single Lyapunov function ▶ conservatism
  - Synthesis ▶ one controller for each vertex
- LPV controller

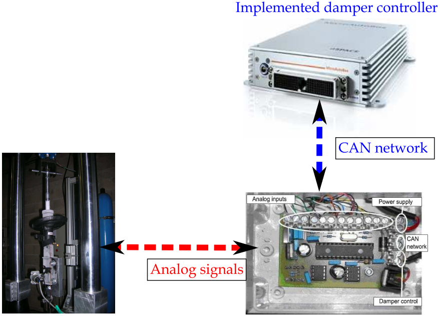
$$K_d(\rho) = \sum_{k=1}^{2^p} \alpha_k(\rho) K_{d_k}$$

$$\alpha_k(\rho) = \frac{\prod_{j=1}^{p} \mid \rho(j) - \Theta_k \mid}{\prod_{j=1}^{p} \mid \overline{\rho} - \underline{\rho} \mid}$$



Experimental set-up

Testing bench

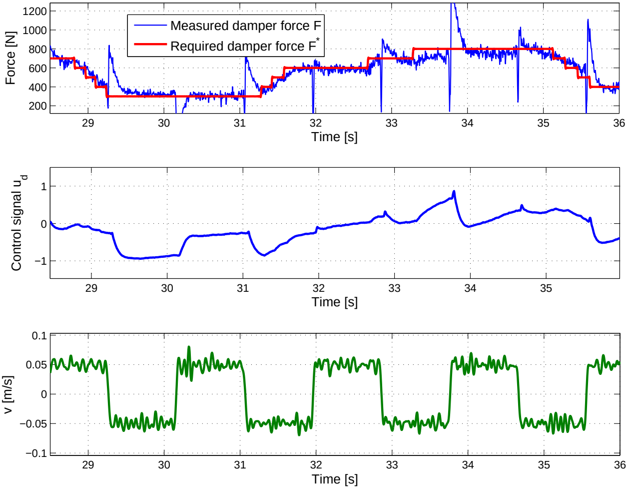


Controlled damper

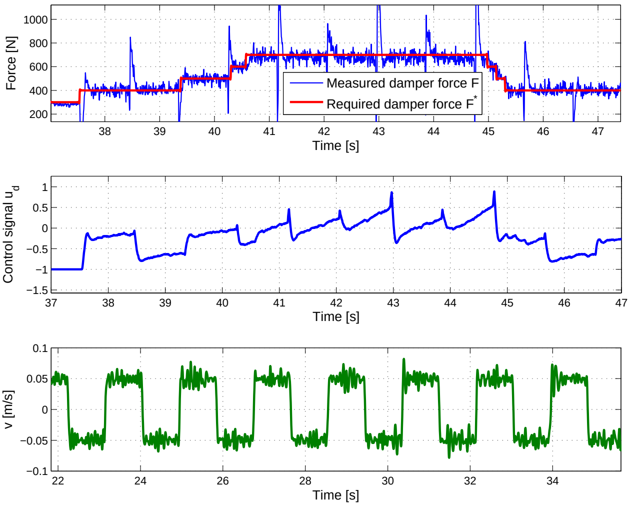
Acquisitions & Valve power suppy

- Testing bench
- Ramp deflections ($\dot{z}_{def} = 0.05 m/s$)
- Dspace acquisition & damper control board
- Closed-loop damper (measured force)

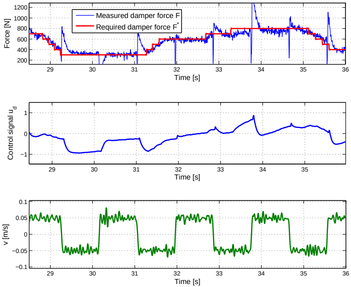
Experimental results : LTI control

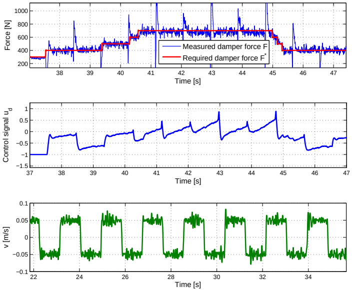


Experimental results : LPV control



Experimental results : synthesis

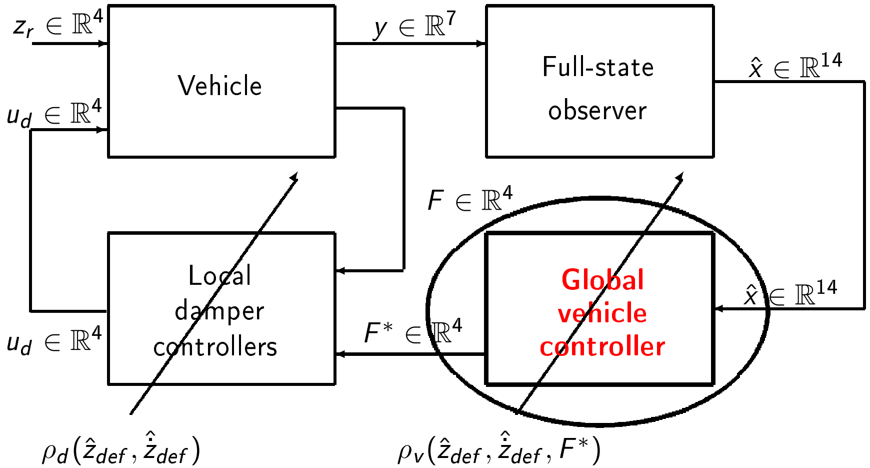




Mean relative errors [%] :

	Experiments	Simulations
LTI control	14	11
LPV control	6	5

Control architecture



Objectives

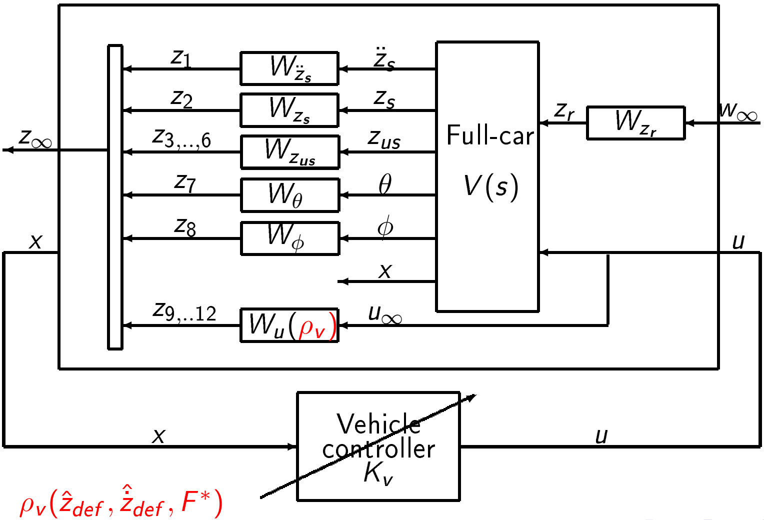
- Minimize the $\mathcal{H}_\infty$-norm of the input-output relations

Relation	Frequency range
$z_{r_i} \mapsto \ddot{z}_s,\ i \in [1,4]$	[4-20]Hz
$z_{r_i} \mapsto z_s,\ i \in [1,4]$	[0-5]Hz
$z_{r_i} \mapsto z_{us_i},\ i \in [1,4]$	[0-20]Hz
$M_x \mapsto \theta$	[0-5]Hz
$M_y \mapsto \phi$	[0-5]Hz

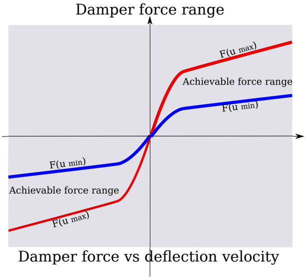
[Sammier, 2001]

- Compute achievable semi-active force references
- Ensure the dissipativity constraint

Generalized plant



Scheduling strategy to ensure dissipativity constraint



- Achievable damper force range

$$
\begin{aligned}
F(u_{min}) &= (A_1 u_{d_{min}} + A_2)\tanh(A_3 \dot{z}_{def} + A_4 z_{def}) + A_5 \dot{z}_{def} + A_6 z_{def} + A_7 \\
F(u_{max}) &= (A_1 u_{d_{max}} + A_2)\tanh(A_3 \dot{z}_{def} + A_4 z_{def}) + A_5 \dot{z}_{def} + A_6 z_{def} + A_7
\end{aligned}
$$

- Dynamical force reference saturation

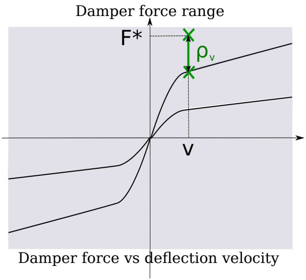
$$
\begin{aligned}
\underline{F} &= \min(F(u_{min}), F(u_{max})) \\
\overline{F} &= \max(F(u_{min}), F(u_{max})) \\
F_{rea} &= \min(\max(F^*, \underline{F}), \overline{F})
\end{aligned}
$$

- Parameter calculation

$$
W_u(\rho_v) = \rho_v = (\overline{\rho}_v - \underline{\rho}_v)\min(\overline{\epsilon}, |F_{rea} - F^*|) + \underline{\rho}_v
$$

- Extension of previous results [Poussot-Vassal, 2008]
  - Full vertical car
  - Static state-feedback controller
  - Identified damper model based scheduling

Scheduling strategy to ensure dissipativity constraint



- Achievable damper force range

$$F(u_{min}) = (A_1 u_{d_{min}} + A_2)\tanh(A_3 \dot{z}_{def} + A_4 z_{def}) + A_5 \dot{z}_{def} + A_6 z_{def} + A_7$$
$$F(u_{max}) = (A_1 u_{d_{max}} + A_2)\tanh(A_3 \dot{z}_{def} + A_4 z_{def}) + A_5 \dot{z}_{def} + A_6 z_{def} + A_7$$

- Dynamical force reference saturation

$$\underline{F} = \min(F(u_{min}), F(u_{max}))$$
$$\overline{F} = \max(F(u_{min}), F(u_{max}))$$
$$F_{rea} = \min(\max(F^*, \underline{F}), \overline{F})$$

- Parameter calculation

$$W_u(\rho_v) = \rho_v = (\overline{\rho}_v - \underline{\rho}_v)\min(\overline{\epsilon}, |F_{rea} - F^*|) + \underline{\rho}_v$$

- Extension of previous results [Poussot-Vassal, 2008]
  - Full vertical car
  - Static state-feedback controller
  - Identified damper model based scheduling

Complete control strategy simulations

Simulation of the whole system :

- Nonlinear vehicle model
- Observer $\Rightarrow$ State estimation
- Local controller $\Rightarrow$ Force reference tracking
- Global controller $\Rightarrow$ Force reference generation

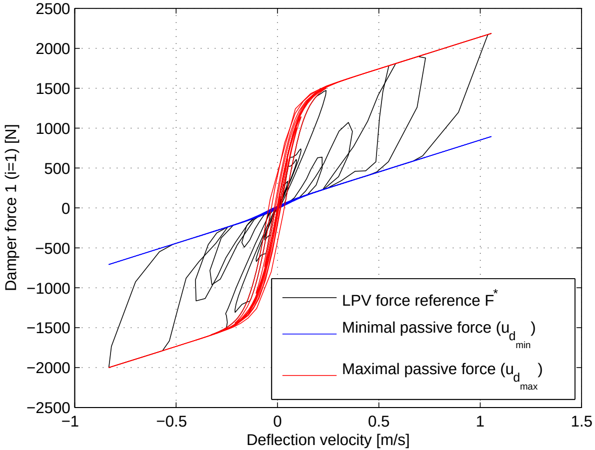
Vehicles under study :

1. Passive linear damper with minimal damping rate,
2. Passive linear damper with maximal damping rate,
3. Active damper,
4. Semi-active LPV control,
5. Semi-active ADD control (Acceleration Driven Damper) [Savaresi et al., 2005]

$$\text{damping rate} = \begin{cases} c_{max} & \text{if} \quad \ddot{z}_{s_i}(\dot{z}_{s_i} - \dot{z}_{us_i}) > 0 \\ c_{min} & \text{if} \quad \ddot{z}_{s_i}(\dot{z}_{s_i} - \dot{z}_{us_i}) < 0 \end{cases}$$

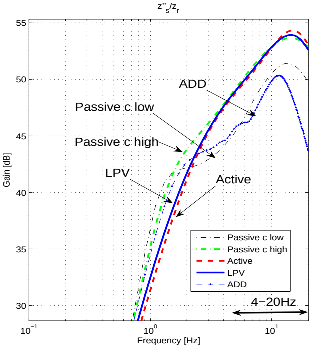
Time-domain simulations with random road disturbance

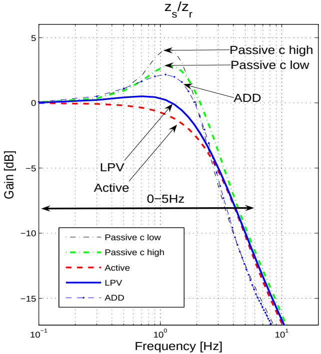
Achievable force range and force reference



Frequency-domain simulations of the nonlinear vehicle model

Classical calculation method [Savaresi et al., 2010]

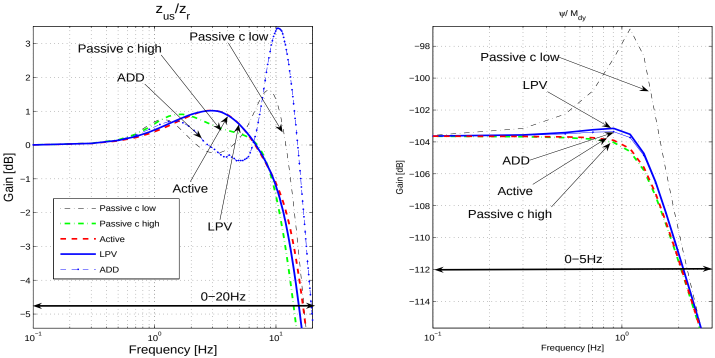




- ADD ▶ Good high-frequency comfort level
- LPV ▶ Good low frequency comfort level

Frequency-domain simulations of the nonlinear vehicle model

Classical calculation method [Savaresi et al., 2010]



- LPV ▶ Good roadholding performance
- ADD ▶ Poor roadholding performance

Performance criteria

Performance criteria :

$$PSD_{f_1 \to f_2}(X(f)) = \sqrt{\int_{f_1}^{f_2} X^2(f) \cdot df}$$

Comparison to nominal passive :

	$\ddot{z}_s/z_r$	z_s/z_r	z_{us}/z_r	ψ/z_r
Active $\mathcal{H}_\infty$	11%	17 %	24 %	2%
LPV $\mathcal{H}_\infty$	8%	15 %	22 %	-2%
ADD	18%	7 %	-7 %	-4%

Summary

Conclusion

- **Contribution**
  - Observer design methodology
  - Complete and flexible suspension control methodology
  - Improved comfort and roadholding
  - Adjustable performance specifications
  - Achievable force reference
  - Local damper control $\Rightarrow$ Nonlinearities
  - Implementable control strategy
- **Perspectives**
  - Vehicle controller : implementation and tests in practice
  - Global chassis control
  - Feedforward control

Conclusion

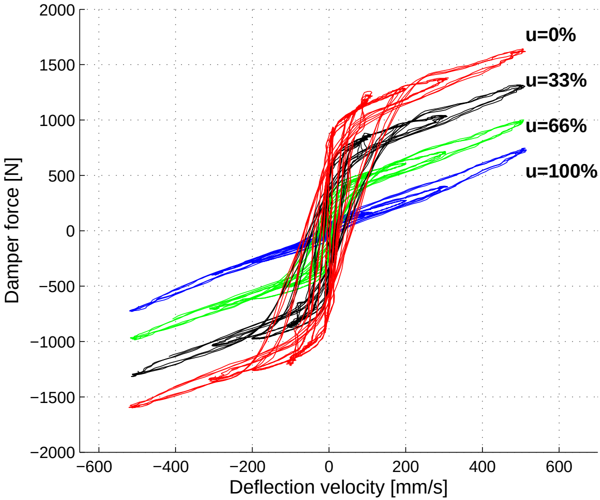
Questions

Experiment 1 : protocol

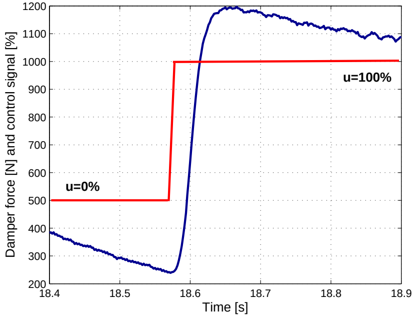
- **Sine deflections** : various amplitudes, frequencies and control signals (constant)

Sine wave	Freq. [Hz]	Amp. [mm]	Control [%]
Sine 1	1.5	1	0, 33, 66 and 100
Sine 2	1.5	10.5	0, 33, 66 and 100
Sine 3	1.5	21	0, 33, 66 and 100
Sine 4	12	1.4	0, 33, 66 and 100
Sine 5	12	4.1	0, 33, 66 and 100
Sine 6	12	6.9	0, 33, 66 and 100

Experiment 1 : results



Experiment 2

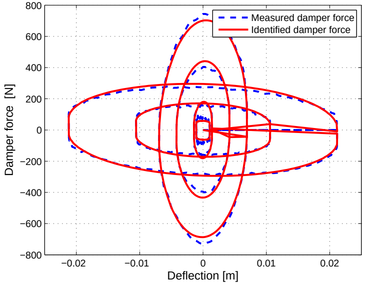


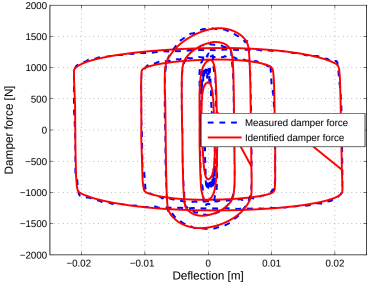
Step response :

- Testing bench
- Ramp deflections (0.1m/s and -0.1m/s)
- Control signal step : 0% to 100%

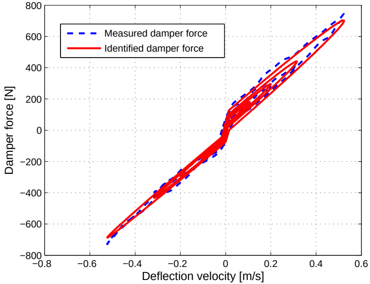
Identified model validation : force deflection

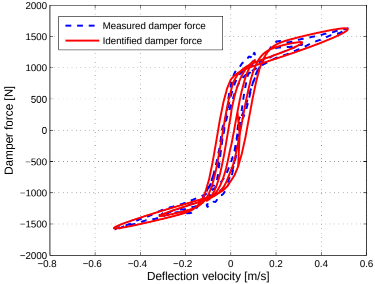
- Minimal control signal
- Other sine deflections (different amplitudes and frequencies)
  $\Rightarrow$ Max relative error : 5.3%





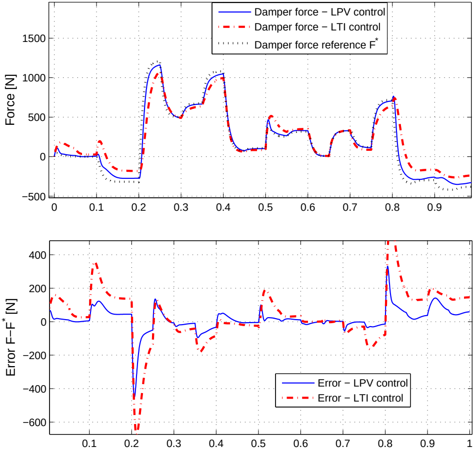
Identified model validation : force velocity





Simulation results : LTI vs LPV controller

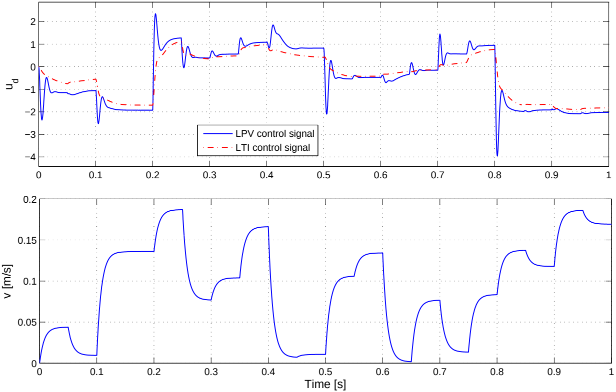
Damper force and tracking error :



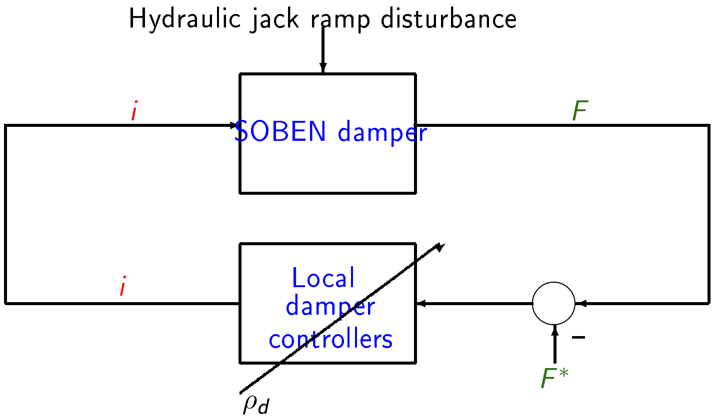
- LTI controller : Synthesized with nominal constant $\rho_{d_i}(\dot{z}_{def_i} = 0.1, z_{def_i} = 0)$
- LPV controller : Synthesized with polytopic approach, ρ_{d_i} scheduled controller

Simulation results : LTI vs LPV controller

Damper control signal and deflection velocity :



Simulations



Simulation results

- Mean relative error :

$$\epsilon_r(F - F^*) = \frac{\sum_{k=1}^{n} |F(k) - F^*(k)|}{F(k)}$$

  - F^* : Force reference
  - F : Damper force
  - n : Number of samples

- Results :

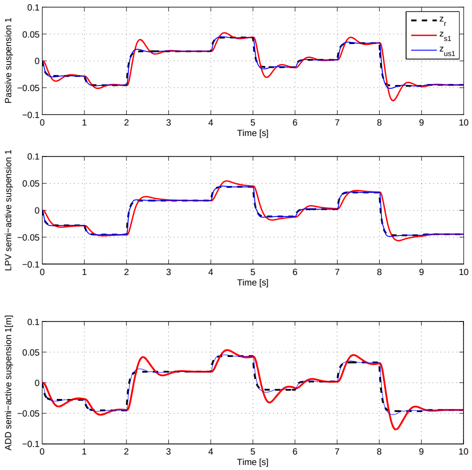
Case	MRE*
LTI	11%
LPV	5%

* Mean Relative Error

Weighting filters

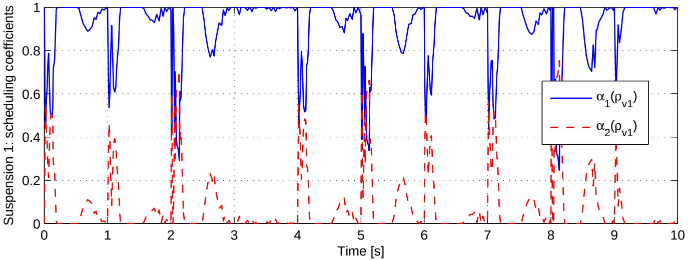
System	Filter (Frequency unit : Hz)	
$\ddot{z}_s \mapsto z_1$	$W_{\ddot{z}_s} = G_{\ddot{z}_s} \frac{s}{s+2\pi f_{\ddot{z}_s}}$	$f_{\ddot{z}_s} = 4$ $G_{\ddot{z}_s} = 0.01$
$z_s \mapsto z_2$	$W_{z_s} = G_{z_s} \frac{2\pi f_{z_s}}{s+2\pi f_{z_s}}$	$f_{z_s} = 5$ $G_{z_s} = 2$
$\theta \mapsto z_7$	$W_\theta = G_\theta \frac{2\pi f_\theta}{s+2\pi f_\theta}$	$f_\theta = 5$ $G_\theta = 2$
$\phi \mapsto z_8$	$W_\phi = G_\phi \frac{2\pi f_\phi}{s+2\pi f_\phi}$	$f_\theta = 5$ $G_\theta = 2$
$z_{us_i} \mapsto z_j$ $i \in [1,4], j \in [3,6]$	$W_{z_{us}} = G_{z_{us}} \frac{2\pi f_{z_{us}}}{s+2\pi f_{z_{us}}}$	$f_{z_{us}} = 20$ $G_{z_{us}} = 1$
$w_\infty \mapsto z_{r_i}$ $i \in [1,4]$	$W_{z_r}^{-1} = G_{z_r} \frac{2\pi f_{z_r}}{s+2\pi f_{z_r}}$	$f_{z_{r_i}} = 20$ $G_{z_{r_i}} = 1$
$u \mapsto z_j$ $i \in [1,4], j \in [9,12]$	$W_{u_i}(\rho_{v_i}) = \rho_{v_i}$	

Simulation results : Sprung and unsprung mass positions



- Sprung mass position :
  - Passive : many oscillations
  - Semi-active : smooth
  - ADD : smooth, less oscillations
- Tire deflection :
  - Passive : large values
  - Semi-active : small values
  - ADD : large values

Simulation results : scheduling coefficients



- $(\alpha_1, \alpha_2) = (1, 0)$ if F^* achievable
- $(\alpha_1, \alpha_2) = (0, 1)$ if F^* not achievable
- u automatically decreased if F^* outside the damper force range

Simulation results