Introduction to vehicle dynamics control

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Outline

1. Introduction
2. Models
3. Intro to Towards global chassis control
4. Active safety using coordinated steering/braking control
   - Active safety
   - Objective
   - Basics on vehicle dynamics
   - Partial non linear Vehicle model
   - Lateral stability control
   - Simulations
5. Road profile estimation and road adaptive vehicle dynamics control
   - Road profile vehicle control adaptation
   - Road Adaptive controller synthesis
   - Implementation & test validation on the INOVE test bench
6. LPV FTC for Vehicle Dynamics Control
   - Towards global chassis control
   - The LPV FTC VDC... approach
   - Simulations on a full NL vehicle model
7. Conclusions and future work
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6. LPV FTC for Vehicle Dynamics Control
   - Towards global chassis control
   - The LPV FTC VDC... approach
   - Simulations on a full NL vehicle model

7. Conclusions and future work
Road safety: an international stake

- Worldwide, *1.24 million people* of road traffic deaths per year (+ 50 million of injuries). 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
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

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)
Introduction

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 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


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 $H_\infty$ 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
Models

Suspension system

Objective

- Link between unsprung \( m_{us} \) and sprung \( m_s \) masses

- Involves \textit{vertical} \( (z_s, z_{us}) \) dynamics

Passive suspension system
Suspension system

Objective

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

Semi-active suspension system \(\rightarrow\) dissipates energy through an adjustable damping coefficient
Suspension system

Objective

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

Active suspension system \(\rightarrow\) 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.

\[
\ddot{z}_s = -\frac{(F_{szfl} + F_{szfr} + F_{szrl} + F_{szrr})}{m_s}
\]

\[
\ddot{z}_{usi,j} = \frac{(F_{szij} - F_{tzij})}{m_{usi,j}}
\]

\[
\dot{\theta} = \frac{((F_{szrl} - F_{szrr})t_r + (F_{szfl} - F_{szfr})t_f)}{I_x}
\]

\[
\dot{\phi} = \frac{((F_{szrr} + F_{szrl})l_r - (F_{szfr} + F_{szfl})l_f)}{I_y}
\]
Wheel & Braking system

Objective

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

Objective

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

Extended quarter vehicle model

Normalized longitudinal tire force

\(F_{tx}/F_n\)

- Dry
- Wet
- Cobblestone
- Icy
Vehicle model - dynamical equations

Full vertical model

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

\[
\ddot{z}_s = -\left( F_{szfl} + F_{szfr} + F_{szrl} + F_{szrr} \right) / m_s
\]

\[
\ddot{z}_{usij} = \left( F_{szij} - F_{tzij} \right) / m_{usij}
\]

\[
\ddot{\theta} = \left( (F_{szrl} - F_{szrr})t_r + (F_{szfl} - F_{szfr})t_f \right) / I_x
\]

\[
\ddot{\phi} = \left( (F_{szrr} + F_{szrl})l_r - (F_{szfr} + F_{szfl})l_f \right) / I_y
\]
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 stability and security issues.

\[
\begin{align*}
\ddot{x}_s &= \frac{(F_{txfr} + F_{txfl}) + (F_{txrr} + F_{txrl})}{m} \\
\ddot{z}_s &= -\frac{(F_{szfl} + F_{szfr} + F_{szrl} + F_{szrr})}{m_s} \\
\ddot{\omega}_{usi j} &= \frac{(F_{szi j} - F_{tzi j})}{m_{usi j}} \\
\ddot{\lambda}_{ij} &= \frac{v_{ij} - R_{ij} \omega_{ij}}{\max(v_{ij}, R_{ij}, \omega_{ij})} \\
\ddot{\omega}_{ij} &= \frac{-RF_{txij}(\mu, \lambda, F_n) + T_{bij}}{I_w}
\end{align*}
\]
Wheel & Steering system

Objective

- Wheel / road contact
- Influences safety performances
- Involves lateral ($y_s$), side slip angle ($\beta$) and yaw ($\psi$) dynamics
Wheel & Steering system

Objective
- Wheel / road contact
- Influences safety performances
- Involves lateral ($y_s$), side slip angle ($\beta$) and yaw ($\psi$) dynamics

Bicycle model
Vehicle model - dynamical equations

Full vertical and longitudinal model

\[
\begin{align*}
\ddot{x}_s &= \frac{((F_{txfr} + F_{txfl}) + (F_{txrr} + F_{txrl}))}{m} \\
\ddot{z}_s &= -\frac{(F_{szfl} + F_{szfr} + F_{szrl} + F_{szrr})}{m_s} \\
\ddot{\omega}_{usij} &= \frac{(F_{szi} - F_{tzij})}{m_{usij}} \\
\dot{\theta} &= \frac{((F_{szrl} - F_{szrr})\tau_r + (F_{szfl} - F_{szfr})\tau_f)}{I_x} \\
\dddot{\phi} &= \frac{((F_{szrr} + F_{szrl})l_r - (F_{szfr} + F_{szfl})l_f + mh\ddot{x}_s)}{I_y} \\
\lambda_{ij} &= \frac{v_{ij} - R_{ij} \omega_{ij}}{\max(v_{ij}, R_{ij} \omega_{ij})} \\
\dot{\omega}_{ij} &= \frac{(-RF_{txij}(\mu, \lambda, F_n) + T_{bij})}{I_w}
\end{align*}
\]
Vehicle model - dynamical equations

Full model

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

\[
\begin{align*}
\ddot{x}_s &= \frac{((F_{txfr} + F_{txrl}) \cos(\delta) + (F_{txrr} + F_{txfl}) - (F_{tyfr} + F_{tyfl}) \sin(\delta) + m \dot{\psi} \dot{y}_s)}{m} \\
\ddot{y}_s &= \frac{((F_{tyfr} + F_{tyrl}) \cos(\delta) + (F_{tyrr} + F_{tyfl}) \sin(\delta) - m \dot{\psi} \dot{x}_s)}{m} \\
\ddot{z}_s &= -\frac{(F_{szfl} + F_{szfr} + F_{szrl} + F_{szrr})}{m} \\
\ddot{\lambda}_{uij} &= \frac{(F_{szij} - F_{tiuj})}{m_{usi}ij} \\
\ddot{\theta} &= \frac{((F_{szrl} - F_{szrr})t_r + (F_{szfl} - F_{szfr})t_f - m h \ddot{y}_s + (I_y - I_z) \dot{\psi} \dot{\phi})}{I_x} \\
\ddot{\phi} &= \frac{((F_{szrr} + F_{szfl})t_r - (F_{szfr} + F_{szfl})t_f + m h \ddot{x}_s + (I_z - I_x) \dot{\psi} \dot{\theta})}{I_y} \\
\ddot{\psi} &= \frac{((F_{tyfr} + F_{tyrl})t_f \cos(\delta) - (F_{tyrr} + F_{tyfl})t_r + (F_{txfr} + F_{txfl}) t_f \sin(\delta) + (I_x - I_y) \dot{\theta} \dot{\phi})}{I_z} \\
\lambda_{ij} &= \frac{v_{ij} - R_{ij} \omega_{ij} \cos(b_{ij})}{\max(v_{ij}, R_{ij} \omega_{ij} \cos(b_{ij}))} \\
\omega_{ij} &= \frac{(-R F_{txij}(\mu, \lambda, F_n) + T_{bij})}{I_w} \\
\beta_{ij} &= \arctan\left(\frac{\dot{x}_{ij}}{\dot{y}_{ij}}\right)
\end{align*}
\]
Vehicle model - dynamical equations

Full model

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

\[
\begin{align*}
\dot{x}_s &= \left( (F_{txfr} + F_{txfl}) \cos(\delta) + (F_{txrr} + F_{txrl}) - (F_{tyfr} + F_{tyfl}) \sin(\delta) + m \dot{y}_s - F_{dx} \right) / m \\
\dot{y}_s &= \left( (F_{tyfr} + F_{tyfl}) \cos(\delta) + (F_{tyrr} + F_{tyrl}) + (F_{txfr} + F_{txfl}) \sin(\delta) - m \dot{x}_s - F_{dy} \right) / m \\
\dot{z}_s &= -(F_{szfl} + F_{szfr} + F_{szrl} + F_{szrr} + F_{dz}) / m_s \\
\dot{\omega}_{usij} &= \left( F_{szij} - F_{tzij} \right) / m_{usij} \\
\ddot{\theta} &= \left( (F_{szrl} - F_{szrr}) l_r + (F_{szfl} - F_{szfr}) l_f - m h \dot{y}_s + (I_y - I_z) \dot{\psi} \dot{\phi} + M_{dx} \right) / I_x \\
\ddot{\phi} &= \left( (F_{szrr} + F_{szrl}) l_r - (F_{szfr} + F_{szfl}) l_f + m h \dot{x}_s + (I_z - I_x) \dot{\psi} \dot{\theta} + M_{dy} \right) / I_y \\
\ddot{\psi} &= \left( (F_{tyfr} + F_{tyfl}) l_f \cos(\delta) - (F_{tyrr} + F_{tyrl}) l_r + (F_{txfr} + F_{txfl}) l_f \sin(\delta) \right. \\
&\left. + (F_{txrr} - F_{txrl}) l_r + (F_{txfr} - F_{txfl}) l_f \cos(\delta) - (F_{txfr} - F_{txfl}) l_f \sin(\delta) \right) \right) / I_z \\
\lambda_{ij} &= v_{ij} - R_{ij} \omega_{ij} \cos \beta_{ij} \\
\dot{\omega}_{ij} &= \left( -R F_{tiij} (\mu, \lambda, F_n) + T_{bij} \right) / I_w \\
\beta_{ij} &= \arctan \left( \frac{x_{ij}}{y_{ij}} \right)
\end{align*}
\]
Vehicle model - synopsis

\[
\begin{bmatrix}
\dot{z}_s, z_s \\
\dot{z}_{us}, z_{us}
\end{bmatrix}
\rightarrow \text{Suspensions} \rightarrow \begin{bmatrix} F_{sz_{ij}} \end{bmatrix} \rightarrow \begin{bmatrix} x_s \\
y_s \\
z_s \end{bmatrix}
\]

(vehicle dynamics)

\[
\begin{bmatrix}
\ddot{x}_s, \ddot{y}_s \\
\dot{\psi}, \nu, \\
F_{sz}, z_{us}
\end{bmatrix}
\rightarrow \text{Wheels} \rightarrow \begin{bmatrix} \lambda_{ij} \\
\beta_{ij} \\
\omega_{ij} \end{bmatrix} \rightarrow \begin{bmatrix} \theta \\
\phi \\
\psi \end{bmatrix}
\]

(tire, wheel dynamics)
Vehicle model - synopsis

\[
\begin{bmatrix}
\dot{z}_s, z_s \\
\dot{z}_{us}, z_{us}
\end{bmatrix}
\rightarrow
\text{Suspensions}
\rightarrow
\begin{bmatrix}
F_{sz_{ij}}
\end{bmatrix}
\rightarrow
\begin{bmatrix}
x_s \\
y_s \\
z_s
\end{bmatrix}
\text{(external disturbances)}
\]

\[
\begin{bmatrix}
\dot{x}_s, \dot{y}_s \\
\dot{\psi}, \nu, \\
F_{sz}, z_{us}
\end{bmatrix}
\rightarrow
\text{Wheels}
\rightarrow
\begin{bmatrix}
\lambda_{ij} \\
\beta_{ij} \\
\omega_{ij}
\end{bmatrix}
\rightarrow
\begin{bmatrix}
\dot{\theta} \\
\phi \\
\psi
\end{bmatrix}
\text{(vehicle dynamics)}
\]

\[
\begin{bmatrix}
\mu_{ij}, z_{rij}
\end{bmatrix}
\rightarrow
\text{(road characteristics)}
\]
Vehicle model - synopsis

Models

Vehicle dynamics:
- \( \mathbf{F}_{\text{tx,zy}} \) and \( \mathbf{M}_{\text{dx,zy}} \) (external disturbances)

Suspensions control:
- \( u_{ij} \)
- \( \mathbf{F}_{sz,ij} \)

Braking & steering control:
- \( \mathbf{T}_{\text{bi,j}}, \delta \)
- \( \mathbf{F}_{\text{tx,zy}} \) (vehicle dynamics)

Road characteristics:
- \( \mathbf{\mu}_{ij}, z_{rij} \) (tire, wheel dynamics)

O.Sename-S.Fergani (GIPSA-lab - LAAS)
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 inappropriate 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 → collaborative based strategy.
Towards global chassis control approaches (GCC)

The GCC strategies are developed in 2 steps:

- The monitoring approach → collaborative based strategy.

- Developing coordinated control strategies → 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/$\mathcal{H}_{\infty}$)
  \[ \Rightarrow \text{MIMO internal stability & no switching} \]
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.
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Active safety using coordinated steering/braking control

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.
Vehicle safety systems:

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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
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 $\mathcal{H}_\infty$ tuning using [Bünste et al., IEEE TCST, 2004] results
- LPV Controller structure exploiting system properties to handle braking constraints
- Nonlinear frequency validations
Problematic

Problem tackled: vehicle critical situations

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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 ($\alpha$)
  2. Vertical load applied on the tire ($F_z$)
  3. Friction coefficient ($\mu$)
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
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
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 an 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:**

  \[ m v (\dot{\beta} - \dot{\psi}) = F_{yf} + F_{yr} \]  (1)

- **Equation of yaw motion:**

  \[ I_z \ddot{\psi} = l_f F_{yf} - l_r F_{yr}, \]  (2)
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.

\[
\begin{bmatrix}
\ddot{\psi} \\
\dot{\beta}
\end{bmatrix} = 
\begin{bmatrix}
-\frac{l^2_f C_f + l^2_r C_r}{l^2_f C_f + l^2_r C_r} & \frac{l_r C_r - l_f C_f}{l_r C_r - l_f C_f} \\
\frac{l_z v}{C_f + C_r} & \frac{l_z}{C_f + C_r} - \frac{l_f C_f}{mv}
\end{bmatrix}
\begin{bmatrix}
\dot{\psi} \\
\dot{\beta}
\end{bmatrix} + 
\begin{bmatrix}
\frac{l_f C_f}{mv} \\
\frac{l_z}{mv}
\end{bmatrix} \delta^* + 
\begin{bmatrix}
\frac{1}{l_z} \\
0
\end{bmatrix} M^*_z + 
\begin{bmatrix}
\frac{1}{l_z} \\
\frac{1}{mv}
\end{bmatrix} 
\begin{bmatrix}
M_{dz} \\
F_{dy}
\end{bmatrix}
\]  

(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:

$H_\infty$ synthesis extended to LPV system:

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

See paper Doumiati et al (2013)
The idea is to design a controller that:

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- Rejects yaw moment disturbances

Methodology:

$H_\infty$ synthesis extended to $LPV$ system:

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

See paper Doumiati et al (2013)
**Overall control scheme diagram**

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

**Diagram Details**:
- **Reference model (bicycle model) + Bounding limits**
- **Vehicle velocity**
- **Steering angle ($\delta_d$)**
- **Driver command**
- **Sideslip dynamics**
- **VDSC controller**
- **Vehicle simulation model**
- **External yaw disturbances**

**Symbols and Equations**:
- $T_{br}^*$
- $\delta^*$
- $\delta_d$
- $\omega$ (yaw rate target)
- $\omega$ (yaw rate (measure))

**Abbreviations**:
- AS: Steer-by-wire system
- EMB: Brake-by-wire Electro Mechanical system
Overall control scheme diagram

Vehicle velocity

Reference model (bicycle model) + Bounding limits

Driver command

Steering angle ($\delta_d$)

Sideslip dynamics

External yaw disturbances

Vehicle simulation model

AS: Steer-by-wire system

EMB: Brake-by-wire Electro Mechanical system
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
Reference model

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Steering angle ($\delta_d$)

Driver command

Sideslip dynamics

VDSC controller

$T_{br}^*$

$\delta^*$

$\delta_d$

Vehicle simulation model

External yaw disturbances

Monitor

Yaw rate target

Vehicle velocity

Yaw rate (measure)

+ -

+ +

AS

EMB

Driver command

Reference model (bicycle model) + Bounding limits

Vehicle velocity
VDSC Controller architecture

Objective: Lateral stability control

Measurements/estimation

Level 1

Level 2

Upper controller

Desired yaw moment

Lower controller

Applying brake torque to the appropriate wheel

Commanded steering angle

and/or

Steering angle

SideSlip angle

Yaw rate

O.Sename-S.Fergani (GIPSA-lab - LAAS) Intelligent Vehicles Summer School July 2-7, 2017
VDSC-upper controller: \( LPV / H_\infty \) control

Vehicle model is LTI:
- Linear bicycle model
- Synthesized considering a dry road

\( \rho \) scheduling parameter:
- \( \rho(t) \) is time dependent and known function
- \( \rho \) bounded: \( \rho \in [\rho, \bar{\rho}] \)

Generalized plant (tracking problem)

\[
\begin{align*}
  w(t) &= \begin{bmatrix} \dot{\psi}_d \cdot F_{dy} \\ \delta^* \\ M_z^* \\ e \psi \end{bmatrix} & \text{exogenous input} \\
  u(t) &= \begin{bmatrix} F_{dy} \\ \delta^* \\ M_z^* \\ e \psi \end{bmatrix} & \text{control input} \\
  y(t) &= \begin{bmatrix} e \dot{\psi} \end{bmatrix} & \text{measurement} \\
  \tilde{z}(t) &= \begin{bmatrix} W_1 \tilde{z}_1, W_2 \tilde{z}_2, W_3 \tilde{z}_3, W_4 \tilde{z}_4 \end{bmatrix} & \text{controlled output}
\end{align*}
\]
**VDSC design (cont’d)**

- \( z_1 \) : sideslip angle signal, \( \beta \): \( W_1 = 2 \). \( \leadsto \) 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 \( w_0 = 70 \text{ rad/s} \).

- \( z_3 \) braking control signal, \( M^*_z \), according to a scheduling parameter \( \rho \):
  \[
  W_3 = \frac{\rho}{s/(\alpha 2\pi f_2) + 1},
  \]

  where \( f_2 = 10 \text{ Hz} \) is the braking actuator cut-off frequency and \( \alpha = 100 \).

  \( \rho \in \{ \rho \leq \rho \leq \bar{\rho} \} \) (with \( \rho = 10^{-4} \) and \( \bar{\rho} = 10^{-2} \)).

- \( z_4 \), the steering control signal attenuation (\( f_3 = 1 \text{ Hz} \), \( f_4 = 10 \text{ Hz} \)):
  \[
  W_\delta = G_\delta 0 \frac{(s/2\pi f_3 + 1)(s/2\pi f_4 + 1)}{(s/\alpha 2\pi f_4 + 1)^2} \frac{(\Delta_f/\alpha 2\pi f_4 + 1)^2}{(\Delta_f/2\pi f_3 + 1)(\Delta_f/2\pi f_4 + 1)} \quad \text{and} \quad \Delta_f = 2\pi(f_4 + f_3)/2
  \]

  This filter is designed in order to allow the steering system to act only in \([f_3, f_4] \text{ Hz} \). At \( \Delta_f/2 \), the filter gain is unitary [Bunte et al. 2004, TCST].
VDSC design (cont’d)

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

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  \]
  \[
  G_\delta 0 = \frac{(\Delta f/2\pi f_3 + 1)(\Delta f/2\pi f_4 + 1)}{(\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
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Controller solution: $LPV/H_\infty$ control

- Mixed-Sensitivity problem
- Minimizes the $H_\infty$ 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 $\delta^*$ and $M_z^*$
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Active safety using coordinated steering/braking control

Lateral stability control

VDSC-upper controller: \( LPV / H_{\infty} \) control

Sensitivity functions:

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

Figure: Closed loop transfer functions between \( \beta \) and exogenous inputs
VDSC-upper controller: $LPV/H_\infty$ control

Sensitivity functions:

- $|e_\dot{\psi}/\dot{\psi}|$
- $|e_\psi/F_{dy}|$
- $|e_\psi/M_{dz}|$

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

- Attenuation of the yaw rate error
- Rejection of the yaw disturbance
VDSC-upper controller: \( LPV / H_\infty \) control design

Sensitivity functions:

![Graphs of closed loop transfer functions between \( M^* \) and exogenous inputs]

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

\( \rho = 0.1 \rightarrow \text{braking is activated, } \rho = 10 \rightarrow \text{braking is penalized} \)
VDSC-upper controller: $LPV/H_\infty$ control design

Sensitivity functions:

![Magnitude plots](image)

**Figure**: Closed loop transfer functions between $\delta^*$ 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

- Vehicle velocity
- Reference model (bicycle model) + Bounding limits
- Yaw rate target
- VDSC controller
  - Monitor
  - Yaw rate (measure)
- Steering angle ($\delta_d$)
- External yaw disturbances
- Driver command
- Friction estimator
- Vehicle simulation model
  - EMB
  - AS
  - Monitor
  - Yaw rate target
  - Vehicle velocity
  - Sideslip dynamics

O.Sename-S.Fergani (GIPSA-lab - LAAS)
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, $\chi$, allows accurate diagnosis of the vehicle stability.
Coordination between Steering and braking

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This criterion, $\chi$, allows accurate diagnosis of the vehicle stability.
Monitor

\[ \rho(\chi) := \begin{cases} 
\frac{\bar{\rho}}{\bar{\chi} - \chi} & \text{if } \chi \leq \bar{\chi} \text{ (steering control - Steerability control task)} \\
\frac{\chi - \bar{\chi}}{\bar{\chi} - \chi} \frac{\bar{\rho}}{\bar{\chi} - \chi} + \frac{\chi - \bar{\chi}}{\bar{\chi} - \chi} \rho & \text{if } \chi \leq \chi < \bar{\chi} \text{ (steering+braking)} \\
\rho & \text{if } \chi \geq \bar{\chi} \text{ (steering+full braking - Stability control task)} 
\end{cases} \]  

(6)

where \( \chi = 0.8 \) and \( \bar{\chi} = 1 \) (\( \chi \) is user defined)
Sideslip angle estimation

Available measurements (from ESC or reasonable cost sensors):

- Yaw rate, $\dot{\psi}$
- Steering wheel angle, $\delta$
- Wheel speeds, $w_{ij}$
- Lateral acceleration, $a_y$

$\dot{\beta}$ can be evaluated through available sensors:

$$\dot{\beta} = \frac{a_y}{v_x} - \psi$$  \hspace{1cm} (7)

Existing Methods:

- Integration of $\dot{\beta}$
- Kinematic equations (e.g., $a_y = a_x$)
- Model-based observer (Vehicle model + Estimation technique)
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Sideslip angle estimation

This study:

- Planar bicycle model (with constant velocity):
  \[
  \begin{align*}
  \dot{\beta} &= \frac{(F_y f + F_y r)}{(m v)} + \dot{\psi} \\
  \ddot{\psi} &= \frac{[l_f F_y f - l_r F_y r + M_z^*]}{I_z}
  \end{align*}
  \] (8)

- Dugoff’s tire model:
  \[F_y = -C_\alpha \times \tan(\alpha) \times f(\alpha, F_z, C_\alpha),\text{ where } f(.) \text{ is nonlinear} \] (9)

- Nonlinear filtering: Extended Kalman Filter

State-space representation:

- \( X = [\beta, \dot{\psi}]^T \)
- \( U = [M_z^*, \delta, F_z]^T \)
  - \(
  \delta = \delta^* + \delta_d.
  \)
- \( Y = [\dot{\psi}]^T \)
Sideslip angle estimation

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- Planar bicycle model (with constant velocity):
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- \( Y = [\psi]^T \)
VDSC Controller architecture

Objective: Yaw stability control

Upper controller

Commanded steering angle
Desired yaw moment

Measurements/estimation

Yaw rate
Steering angle
Sideslip angle

Lower controller

Applying brake torque to the appropriate wheel

Level 1

Level 2

O.Sename-S.Fergani (GIPSA-lab - LAAS)
Intelligent Vehicles Summer School
July 2-7, 2017
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:

- Oversteer: Brake outer rear wheel
- Understeer: Brake inner rear wheel
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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
Test 1: Results [dry road $\mu = 0.9$, $V = 100 \text{ km/h}$]

Figure: $M^*_z$ and $\rho$ variations according to $\chi$ for the double lane-change maneuver

Figure: Control signals generated by the controller
Test 2: Results [wet road $\mu = 0.5$, $V = 80 \text{ km/h}$]

Vehicle dynamic responses with and without controller
Test 2: Results [wet road $\mu = 0.5$, $V = 80 \, km/h$]

**Figure:** Response of the yaw rates versus steering wheel angle

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Outline

1. Introduction
2. Models
3. Intro to Towards global chassis control
4. Active safety using coordinated steering/braking control
   - Active safety
   - Objective
   - Basics on vehicle dynamics
   - Partial non linear Vehicle model
   - Lateral stability control
   - Simulations
5. Road profile estimation and road adaptive vehicle dynamics control
   - Road profile vehicle control adaptation
   - Road Adaptive controller synthesis
   - Implementation & test validation on the INOVE test bench
6. LPV FTC for Vehicle Dynamics Control
   - Towards global chassis control
   - The LPV FTC VDC... approach
   - Simulations on a full NL vehicle model
7. Conclusions and future work
Road profile vehicle control adaptation

Road Profile estimation strategies

- The $\mathcal{H}_\infty$ 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 identification.

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

- Road Adaptive Semi-Active Suspension for 1/4 vehicle using an LPV/$\mathcal{H}_\infty$ Controller.
- A new LPV/$\mathcal{H}_\infty$ 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 Adaptive Semi-Active Suspension for 1/4 vehicle using an LPV/$\mathcal{H}_\infty$ 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/$H_\infty$ semi-active suspension control adaptation to the type of the road profile.
Road Adaptive control

- Road profile roughness estimation to identify the type of the road.
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A shock absorber with magneto-rheological fluid that changes damping characteristics in the suspension system depending on electric current.

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

\[ \rho_1 = f(\tanh(z_{def}, \dot{z}_{def}), I) \]
\[ \rho_2 = f(\text{sat}(\tanh(z_{def}, \dot{z}_{def}), I)) \]

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

\( \rho_1 \in [-1, 1] \rightarrow \text{Nonlinearities.} \)
\( \rho_2 \in [0, 1] \rightarrow \text{Saturation.} \)
Road Adaptive control

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

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

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

\[
\hat{f}_{z_r} = \frac{\hat{z}_{rRMS}}{2\pi \cdot z_{rRMS}} \quad [Hz]
\]

\[
S_{z_r}(f_{z_r}) = \left(\hat{A}_{z_r}\right)^2/(2\Delta f)
\]

\[
\hat{A}_{z_r} = \sqrt{\alpha_1^2 + \beta_1^2}
\]
Road Adaptive control

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<table>
<thead>
<tr>
<th>Type of Road</th>
<th>Class</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth runway</td>
<td>A</td>
</tr>
<tr>
<td>Smooth highway</td>
<td>B</td>
</tr>
<tr>
<td>Highway with gravel</td>
<td>C</td>
</tr>
<tr>
<td>Rough runway</td>
<td>D</td>
</tr>
<tr>
<td>Pasture</td>
<td>E</td>
</tr>
<tr>
<td>Plowed field</td>
<td>F</td>
</tr>
</tbody>
</table>

Vehicle Model with the semi-active MR dampers

Road profile estimation reconstruction

Measurement

Internal varying Parameters $\rho_1, \rho_2$

(dissipativity, saturation)

Frequency estimation and extraction

Using ISO 8608

Road roughness Estimation $\rho_3$

PSD

ISO 8608

Road Classification

(H\text{\textsubscript{\infty}} Observer)
Road Adaptive control

- Road profile roughness estimation to identify the type of the road.
- LPV/$H_\infty$ semi-active suspension control adaptation to the type of the road profile.
Two scheduling parameters in the model:

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

- \( \rho_1 \in [-1, 1] \rightarrow \text{Nonlinearities.} \)
- \( \rho_2 \in [0, 1] \rightarrow \text{Saturation.} \)
One scheduling parameter $\rho_3$ for online suspension adaptation to the road profile:

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

where $K_{\rho_3}$ is used to bound $\rho_3$, such that

$$I(\rho_3) := \begin{cases} 
I = I_{\text{max}} & \text{if } \rho_3 \geq \bar{\rho}_3 \\
I_{\text{min}} < I < I_{\text{max}} & \text{if } \underline{\rho}_3 < \rho_3 < \bar{\rho}_3 \\
I = I_{\text{min}} & \text{if } \rho_3 \leq \underline{\rho}_3
\end{cases} \quad (11)$$
**LPV/\( \mathcal{H}_\infty \) control synthesis**

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

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

where,

\[
\alpha_k(\rho) = \frac{\prod_{j=1}^{2^3} |\rho(j) - c^c(\Omega_k)_j|}{\prod_{j=1}^{i} (\overline{\rho}(j) - \underline{\rho}(j))}
\]

\[
\sum_{k=1}^{2^3} \alpha_k(\rho) = 1 , \alpha_k(\rho) > 0
\]
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.
Road adaptive control validation & implementation

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Road classification implementation results

Vehicle Model with the semi-active MR dampers

Measurement

Road profile estimation reconstruction

Internal varying Parameters $\rho_1, \rho_2$

(dissipativity, saturation)

Frequency estimation and extraction

Using ISO 8608

Amplitude estimation

Road roughness Estimation $\rho_3$

PSD ISO 8608

Road adaptive controller $K(\rho_1, \rho_2, \rho_3)$

Road classification

(A) Road profile

Experimental data

Estimation

(B) PSD of the roughness

Estimated roughness

Thresholds for ISO roads

(C) Road identification

Implemented road

Road classification

Time [s]

$[m]$
Road classification implementation results

Table: Road profiles Classification (ISO 8608).

<table>
<thead>
<tr>
<th>Type of Road</th>
<th>Class</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth runway</td>
<td>A</td>
</tr>
<tr>
<td>Smooth highway</td>
<td>B</td>
</tr>
<tr>
<td>Highway with gravel</td>
<td>C</td>
</tr>
<tr>
<td>Rough runway</td>
<td>D</td>
</tr>
<tr>
<td>Pasture</td>
<td>E</td>
</tr>
<tr>
<td>Plowed field</td>
<td>F</td>
</tr>
</tbody>
</table>

(A) Road profile

Experimental data
Estimation

(B) PSD of the roughness

Estimated roughness
Thresholds for ISO roads

(C) Road identification

Implemented road
Road classification

Time [s]
Road profile estimation and road adaptive vehicle dynamics control
Implementation & test validation on the INOVE test bench

Road adaptive control implementation results

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

(b) Road type F, \( v_x = 30 \text{ Km/h} \)

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

(d) Road type A, \( v_x = 100 \text{ Km/h} \)
Outline

1. Introduction
2. Models
3. Intro to Towards global chassis control
4. Active safety using coordinated steering/braking control
   - Active safety
   - Objective
   - Basics on vehicle dynamics
   - Partial non linear Vehicle model
   - Lateral stability control
   - Simulations
5. Road profile estimation and road adaptive vehicle dynamics control
   - Road profile vehicle control adaptation
   - Road Adaptive controller synthesis
   - Implementation & test validation on the INOVE test bench
6. LPV FTC for Vehicle Dynamics Control
   - Towards global chassis control
   - The LPV FTC VDC... approach
   - Simulations on a full NL vehicle model
7. Conclusions and future work
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/Outputs:

\[
\begin{align*}
    w(t) &= [\psi_{\text{ref}}(v)(t), M_{dz}(t)] \\
    u(t) &= [\delta^+(t), T_{brl}^+(t), T_{brr}^+(t)] \\
    y(t) &= e_{\psi}(t) \\
    z(t) &= [z_1(t), z_2(t), z_3(t)]
\end{align*}
\]

Weighting functions for performance requirements

\( W_{\dot{\psi}_{\text{ref}}} \) and \( W_{\dot{v}_y} \) are 1st order systems.

Weighting functions for actuator coordination

- \( W_{\delta}(\rho_s) = (1 - \rho_s) \times 4\text{th order} \quad \rightarrow \text{braking (and steering) penalized if } \rho = \bar{\rho} \)
- \( W_{T_{brj}}(\rho_b) = (1 - \rho_b) \times 1\text{st order} \quad \rightarrow \text{braking (and steering) allowed if } \rho = \rho \)

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.

\[ K_s(\rho_l) := \begin{pmatrix} u_{fl}^{H_\infty}(t) \\ u_{fr}^{H_\infty}(t) \\ u_{rl}^{H_\infty}(t) \\ u_{rr}^{H_\infty}(t) \end{pmatrix} = \begin{pmatrix} 1 - \rho_l & 0 & 0 & 0 \\ 0 & \rho_l & 0 & 0 \\ 0 & 0 & 1 - \rho_l & 0 \\ 0 & 0 & 0 & \rho_l \end{pmatrix} C_c^0(\rho_s)x_c(t) \]

\( \rho_l \) allows to generate the adequate suspension forces in the 4 corners of the vehicle depending on the load transfer (left ⇐ 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 $t = 2s$ to $t = 6s$)

- Faulty left rear braking actuator:
  saturation = 75N

- 5cm Road bump from $t = 0.5s$ to $t = 1.5s$ and from $t = 4s$ to $t = 5s$)

- Faulty front left damper: force limitation of 70%

- Lateral wind occurs at vehicle’s front generating an undesirable yaw moment
  (from $t = 2.5s$ to $t = 3s$).
Monitoring parameters

- $\rho_b$: handles the braking efficiency
- $\rho_s$: activation of the steering actuator
- $\rho_l$ (LTR): Coordination of suspension control and on-line modification of the suspension performances
Fault Tolerant Yaw control through an efficient coordination of braking/steering actuators.
Fault Tolerant Yaw control through an efficient coordination of braking/steering actuators.
Suspension control distribution

Suspension Forces

Roll displacement

With Damper Fault
Without Damper Fault

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, ...)

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Thank you for your attention
References


• Rossi and Lucente (2004) : Rossi, C. and Lucente, G. "$H_\infty$ control of automotive semi-active suspensions", *1st IFAC Symposium on Advances in Automotive Control (AAC)*. Salerno, Italy.


• **Falcone et al. (2007a)**: Falcone, P.; Borrelli, F.; Tseng, H.; Asgari, J. & Hrovat, D. Integrated braking and steering model predictive control approach in autonomous vehicles, *5th IFAC Symposium on Advances on Automotive Control (AAC)*, 2007


