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Hybrid Systems in Automotive Engine Control

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HYSCOM
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Information Society Technologies

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Hybrid Systems in Automotive Engine Control

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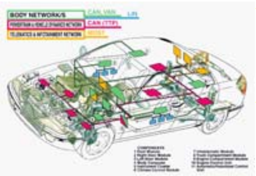
HYCON

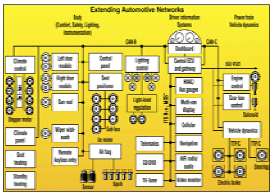
Information Society Technologies

Outline

- Automotive: a promising domain for hybrid systems
  - Model-based design
  - Derivative design
  - Design flow
- Two automotive engine control applications of hybrid systems
  - Actual Engaged Gear Identification: a Hybrid Observer Approach
  - Hybrid Modeling and Control of the Common Rail

Automotive networked control system





- Car manufacturers have to redesign their products more and more frequently to meet customers' demands on innovation
- The pressure of competitiveness is even higher for control system development, since more than 80% of innovation is in electronics
- In today cars, the electronic control system is a networked system
  - with more than 80 interconnected ECUs, some of them safety critical

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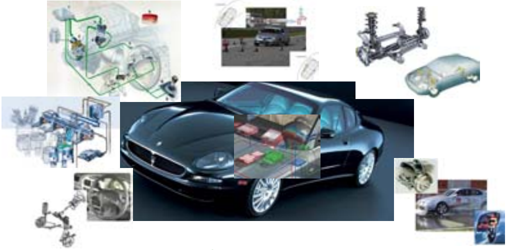
Automotive control systems architecture



- Today, a "one-subsystem one-ECU" networked control system
- This rigid partition between subsystems and electronics
  - results in a higher cost of electronics
  - prevents the design of efficiently integrated functionalities
  - it is often not efficient in terms of communication and synchronization

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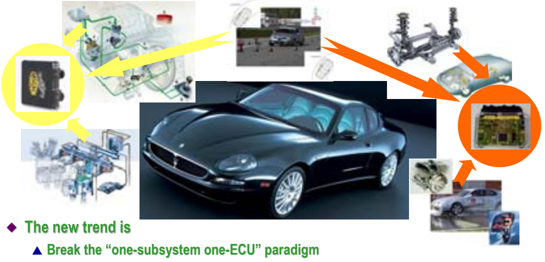
Future scenario for automotive control systems

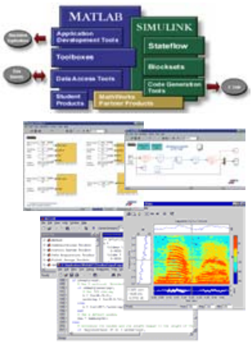


- The new trend is
  - Break the "one-subsystem one-ECU" paradigm
  - Distribute functionalities over several nodes to optimize number and cost of ECUs
- Advantages
  - flexibility, cost reduction, redundancy (fault-tolerance)
  - more sophisticated control enabled by more powerful hardware

AUTOSAR

Model-based design

- Model-based design is becoming widely used in automotive industry
  - algorithms are designed and analyzed using block diagram-based modeling tools
  - correctness of the algorithms is validated against models of the plant
  - models form the basis for all subsequent development stages
    - executable specification (instead of docs)
    - automatic code generation
- Advantages
  - Time-saving and cost-effective
  - Design choices can be explored and evaluated quickly and reliably
  - Ideally, an optimized and fully tested system is obtained



Model-based design

- However, today in the automotive industry
  - model-based design is often limited to control algorithm description
  - not complete plant modeling prevents accurate validation of algorithms
- Experimental validation is still extensively used, but
  - very expensive, time-consuming, bounded coverage
  - due to the high cost, OEM will provide less support to experimentation in Tier-1 companies
- The partial implementation of model-based design is due to
  - insufficient investments in design process innovation
  - lack of methodologies and tools suitable to address critical steps in the design flow, which are currently handled relying on the experience of the designers

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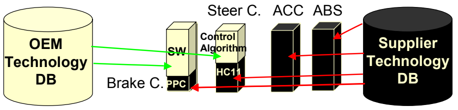
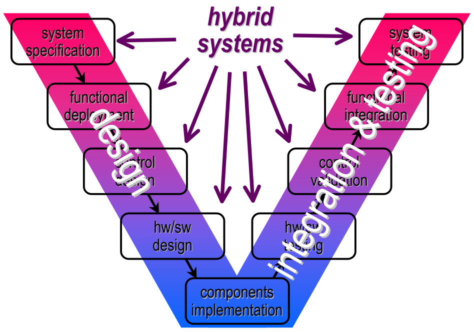
Derivative design

- The derivative design approach:
  - Every two-three years a new generation of products is designed
    - Product generations are conceived to accommodate the specification of all customers for the next years
  - For each commitment, the electronic control unit is obtained by derivation from the current generation
- In the derivative design approach, reuse is extensively employed to minimize cost and development time
  - for each class of applications, products are variants of a same originating design



Automotive industry design process



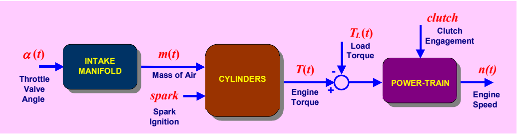
Plant modeling - model development

	DT	CT
DV	🟦	🟪
CV	🟩	🟥

- 4-stroke internal combustion engine
  - 4-stroke engine cycle (FSM + DES + CT)
    - in: spark ignition; injected fuel; air charge; EGR conc.; engine speed;
    - out: engine torque, temperature; dc events; A/F; exhaust gas;
  - fuel injection (FSM + CT)
    - inputs: fuel injection signal (rail pressure regulator command - DI);
    - outputs: injected fuel (rail pressure;fuel temperature - DI);
  - spark ignition (FSM)
    - inputs: ignition coil command; spark command;
    - outputs: spark ignition;
  - air dynamics (CT)
    - inputs: throttle valve command; EGR command; VGT command;
    - outputs: throttle valve angle; temperature; pressure; air flow rate; air charge; EGR concentration;


Maserati Spider - V8

Hybrid model of a 4-stroke engine



Controls	Time / Value
ignition *spark*	disc / disc
throttle α	cont / cont

Disturbances	Time / Value
clutch *clutch*	disc / disc
load torque T_L	cont / cont

intake manifold

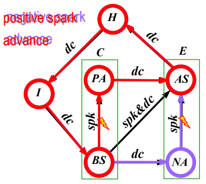
$$\dot p(t) = a_p p(t) + b_p \alpha(t)$$

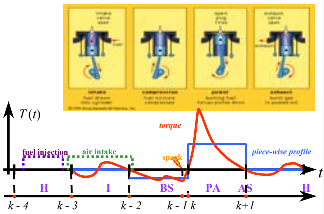
powertrain (idle gear)

$$\dot n(t) = a_n n(t) + b_n(T(t) - T_L(t))$$

$$\dot\theta(t) = 6n(t), \quad \text{if } \theta = 180 \text{ then } \theta := 0$$

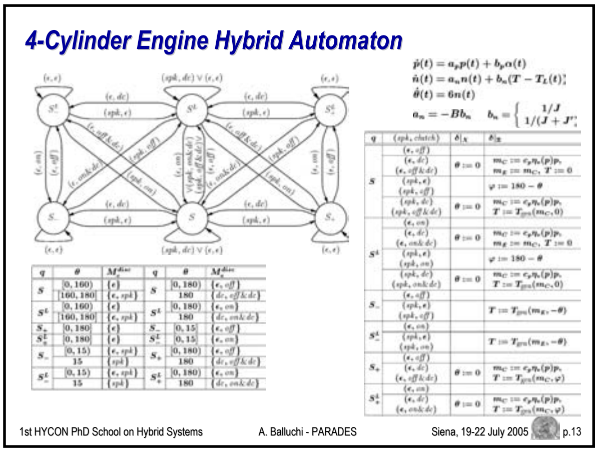
Single cylinder FSM: engine cycle





I → BS	$m := c_p \eta_v(p) p$
BS → PA	$\varphi := 180 - \theta$
PA → AS	$T := T_{\text{pot}}(m)\,\eta_e(\varphi - \varphi_{\text{opt}}, m) = T_{\text{gen}}(m, \varphi)$
NA → AS	$\varphi := -\theta$ $T := T_{\text{gen}}(m, -\theta)$
AS → H	$T := 0$

States	Time / Value
mass of air m	disc / cont
generated torque T	disc / cont
spark advance angle φ	disc / cont

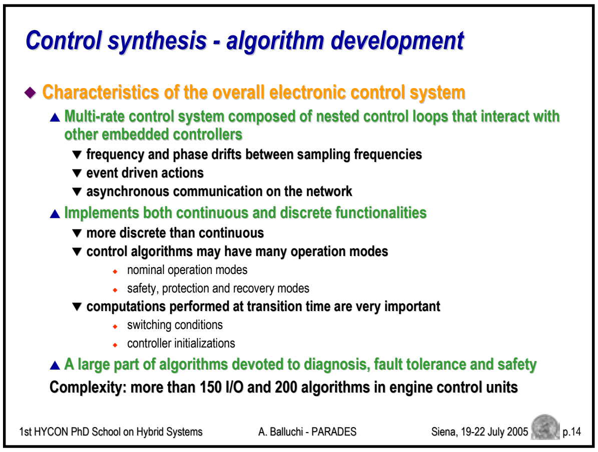


4-Cylinder Engine Hybrid Automaton

q	θ	M^{disc}_q	q	θ	M^{disc}_q
S	$[0,160)$	$\{\epsilon\}$	S	$[0,180)$	$\{\epsilon, off\}$
	$[160,180]$	$\{\epsilon, spk\}$		180	$\{dc, off\&dc\}$
$S^\downarrow$	$[0,160)$	$\{\epsilon\}$	$S^\downarrow$	$[0,180)$	$\{\epsilon, on\}$
	$[160,180]$	$\{\epsilon, spk\}$		180	$\{dc, on\&dc\}$
S_+	$[0,180]$	$\{\epsilon\}$	S_-	$[0,15]$	$\{\epsilon, off\}$
$S^\downarrow_+$	$[0,180]$	$\{\epsilon\}$	$S^\downarrow_-$	$[0,15]$	$\{\epsilon, on\}$
S_-	$[0,15)$	$\{\epsilon, spk\}$	S_+	$[0,180)$	$\{\epsilon, off\}$
	15	$\{spk\}$		180	$\{dc, off\&dc\}$
$S^\downarrow_-$	$[0,15)$	$\{\epsilon, spk\}$	$S^\downarrow_+$	$[0,180)$	$\{\epsilon, on\}$
	15	$\{spk\}$		180	$\{dc, on\&dc\}$

$$\dot p(t) = a_p p(t) + b_p \alpha(t)$$
$$\dot n(t) = a_n n(t) + b_n (T - T_L(t))$$
$$\dot\theta(t) = 6n(t)$$
$$a_n = -Bb_n \quad b_n = \begin{cases} 1/J \\ 1/(J+J') \end{cases}$$

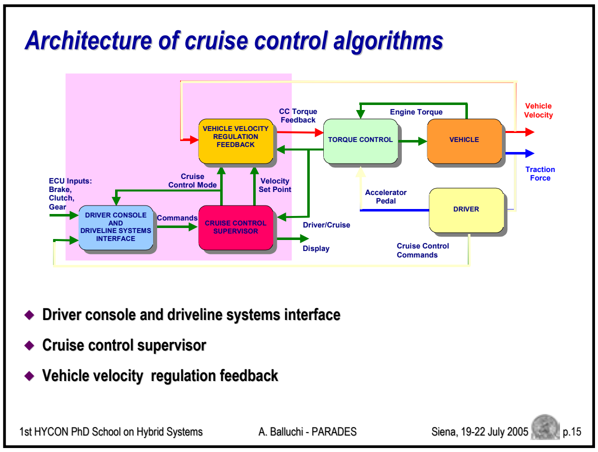
q	$(spk, clutch)$	$\delta\vert_x$	$\delta\vert_z$
S	(ϵ, off)		
	(ϵ, dc) $(\epsilon, off\&dc)$	$\theta := 0$	$m_C := c_p\eta_v(p)p$, $m_E := m_C$, $T := 0$
	(spk, ϵ) (spk, off)		$\varphi := 180 - \theta$
	(spk, dc) $(spk, off\&dc)$	$\theta := 0$	$m_C := c_p\eta_v(p)p$, $T := T_{gen}(m_C, 0)$
$S^\downarrow$	(ϵ, on)		
	(ϵ, dc) $(\epsilon, on\&dc)$	$\theta := 0$	$m_C := c_p\eta_v(p)p$, $m_E := m_C$, $T := 0$
	(spk, ϵ) (spk, on)		$\varphi := 180 - \theta$
	(spk, dc) $(spk, on\&dc)$	$\theta := 0$	$m_C := c_p\eta_v(p)p$, $T := T_{gen}(m_C, 0)$
S_-	(ϵ, off)		
	(spk, ϵ) (spk, off)		$T := T_{gen}(m_E, -\theta)$
$S^\downarrow_-$	(ϵ, on)		
	(spk, ϵ) (spk, on)		$T := T_{gen}(m_E, -\theta)$
S_+	(ϵ, off)		
	(ϵ, dc) $(\epsilon, off\&dc)$	$\theta := 0$	$m_C := c_p\eta_v(p)p$, $T := T_{gen}(m_C, \varphi)$
$S^\downarrow_+$	(ϵ, on)		
	(ϵ, dc) $(\epsilon, on\&dc)$	$\theta := 0$	$m_C := c_p\eta_v(p)p$, $T := T_{gen}(m_C, \varphi)$



Control synthesis - algorithm development

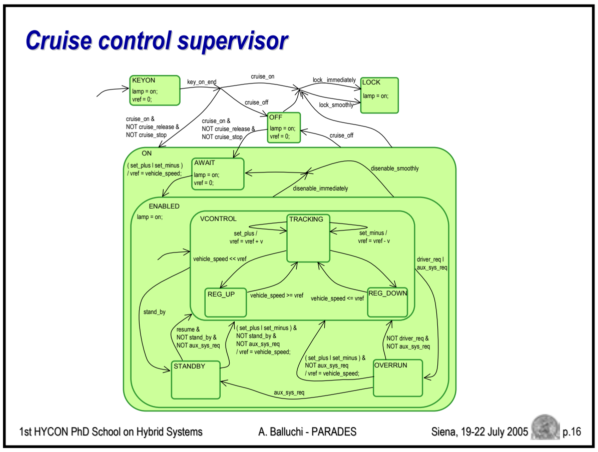
- Characteristics of the overall electronic control system
  - Multi-rate control system composed of nested control loops that interact with other embedded controllers
    - frequency and phase drifts between sampling frequencies
    - event driven actions
    - asynchronous communication on the network
  - Implements both continuous and discrete functionalities
    - more discrete than continuous
    - control algorithms may have many operation modes
      - nominal operation modes
      - safety, protection and recovery modes
    - computations performed at transition time are very important
      - switching conditions
      - controller initializations
  - A large part of algorithms devoted to diagnosis, fault tolerance and safety

Complexity: more than 150 I/O and 200 algorithms in engine control units



Architecture of cruise control algorithms

- Driver console and driveline systems interface
- Cruise control supervisor
- Vehicle velocity regulation feedback



Cruise control supervisor

Actual Engaged Gear Identification: a Hybrid Observer Approach

Applications Paper Prize

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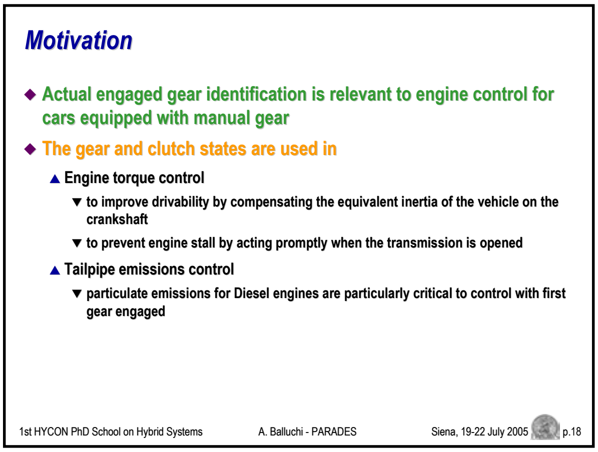
(5) Magneti Marelli Powertrain, Bologna, I

16th IFAC World Congress
Prague, July 4-8, 2005

CC "Control and Computation"



Motivation

- Actual engaged gear identification is relevant to engine control for cars equipped with manual gear
- The gear and clutch states are used in
  - Engine torque control
    - to improve drivability by compensating the equivalent inertia of the vehicle on the crankshaft
    - to prevent engine stall by acting promptly when the transmission is opened
  - Tailpipe emissions control
    - particulate emissions for Diesel engines are particularly critical to control with first gear engaged

Outline

- **Automotive driveline modeling**
  - Detailed hybrid model used for analysis and validation
    - discontinuities and nonlinear dynamics
  - Simplified hybrid model used for synthesis
    - obtained by abstraction and reduction
- **Hybrid design of the actual engaged gear identification algorithm**
- **Validation**
  - Against the detailed hybrid nonlinear model
    - robustness analysis
  - Using experimental data provided by Magneti Marelli Powertrain

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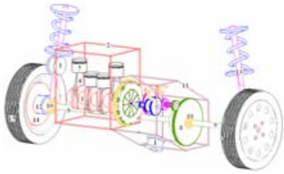
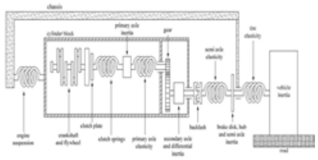
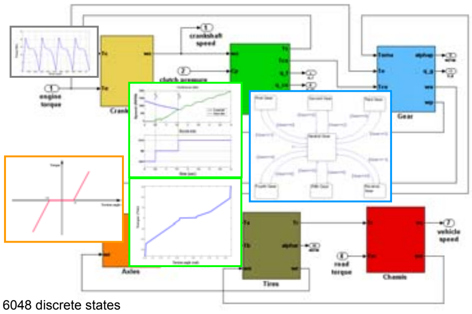
Automotive driveline



1) engine suspension
2) cylinder block
3) crankshaft
4) connecting rod
5) piston
6) clutch
7) gear
8) differential
9) semi-axle
10) flexible joint
11) gear box
12) hub
13) body suspension
14) wheel
15) tire



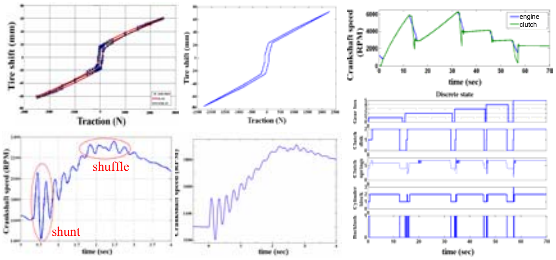
Driveline detailed hybrid model



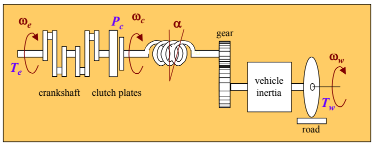
6048 discrete states
12 continuous states

Validation of the model with experimental data

experimental data **proposed model**



Driveline simplified hybrid model



7 discrete states
- gear and clutch

4 continuous states
ω_e - crankshaft speed
ω_c - clutch speed
ω_w - wheel speed
α - torsion angle

1 discrete input
- gear lever $\in \{1, 2,..5, RG, N\}$

3 continuous inputs
T_e - engine torque (m.v. known)
P_c - clutch plate pressure
T_w - wheel torque

2 continuous outputs
ω_e - crankshaft speed
ω_w - wheel speed

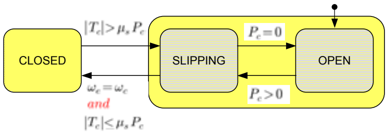
with gear engaged and clutch closed:

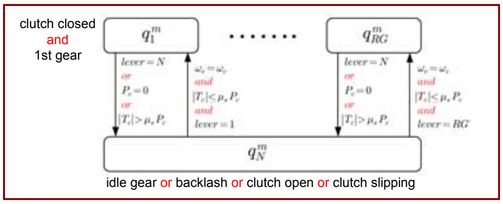
$$\dot{x}(t) = A_i x(t) + B_i u(t)$$
$$y(t) = C_i x(t)$$

$$x = \begin{pmatrix} \alpha \\ \omega_e \\ \omega_w \end{pmatrix}, \quad u = \begin{pmatrix} T_e \\ T_w \end{pmatrix}, \quad y = \begin{pmatrix} \omega_e \\ \omega_w \end{pmatrix}$$

$$\mathbf{A}_i = \begin{bmatrix} 0 & 1 & -\frac{1}{\tau_i} \\ -\frac{k_i}{j_e} & -\frac{b_e + b_i}{j_e} & \frac{b_i}{\tau_i j_e} \\ \frac{k_i}{\tau_i j_w} & \frac{b_i}{\tau_i j_w} & -\frac{b_w + \frac{b_i}{\tau_i}}{j_w} \end{bmatrix} \quad \mathbf{B}_i = \begin{bmatrix} 0 & 0 \\ \frac{1}{j_e} & 0 \\ 0 & -\frac{1}{j_w} \end{bmatrix} \quad \mathbf{C}_i = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

Abstraction of the discrete behavior

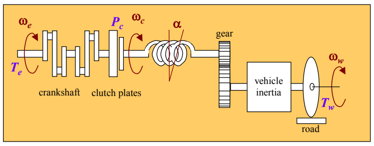


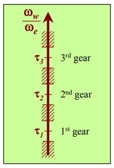


clutch closed
and
1st gear

idle gear or backlash or clutch open or clutch slipping

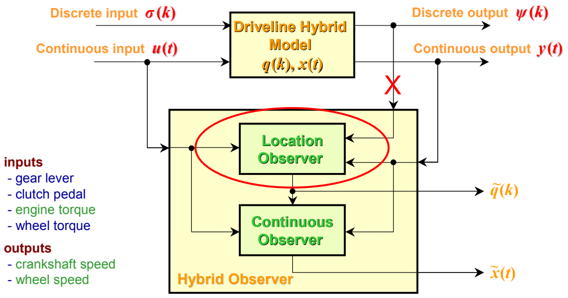
Wheel speed and engine speed comparison





- **Limitations**
  - Large time delays in the engaged gear identification
    - due to oscillations of the transmission shafts during transients
    - particularly critical for idle speed and first gear identification
  - Frequent identification errors
- **Specification**
  - Identification within a delay of 250 msec., sampling period of 12 msec.

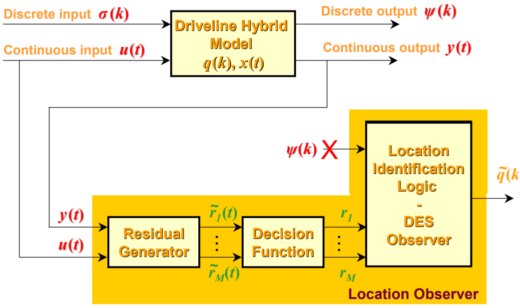
Hybrid observer approach for actual engaged gear identification



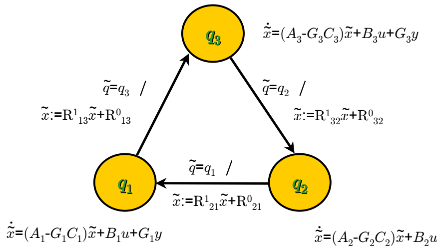
inputs
- gear lever
- clutch pedal
- engine torque
- wheel torque

outputs
- crankshaft speed
- wheel speed

Location observer



Continuous observer



- A switched Luenberger observer with resets and switching controlled by the identified plant location
- Stability can be achieved by using dwell time approach
  - a possible delay in location identification has to be taken into account

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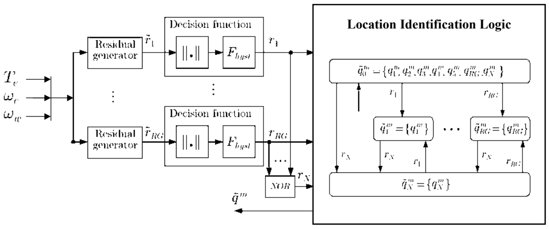
Engaged gear identification algorithm



- The location q_N cannot be detected using a residual generator due to lack of feedbacks
- The corresponding signature is obtained by negation of the others

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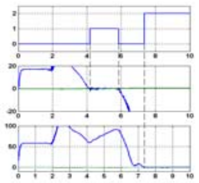
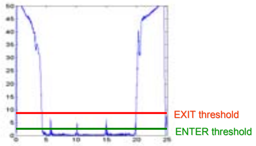
Residual generator and decision function

- **Residual generators**
  - Luenberger obs.
  - Unknown input obs.
  - Kalman filter
  - Walkcott-Zak obs.
  - Sliding mode obs.

$$\dot z_i(t) = (A_i - L_iC_i)z_i(t) + B_iu(t) + L_iy(t)$$
$$\tilde r_i(t) = C_iz_i(t) - y(t)$$

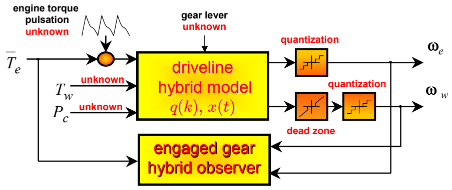


- **Decision function**
  - Passive hysteresis relay
  - Debouncing algorithm



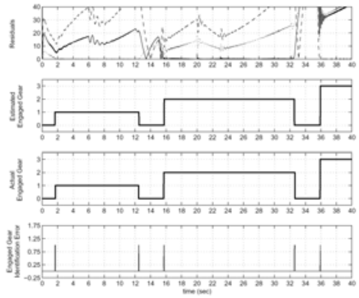
  - Thresholds function of engine torque
  - Decision function output disenabled during residual transients

Validation against the detailed hybrid model of the driveline



- **Unknown inputs**
  - wheel torque (road slope)
  - clutch plate pressure
  - engine torque pulsation
  - gear lever
- **Sensors**
  - engine speed quantization (1 RPM)
  - vehicle speed quantization (1 Km/h) and dead zone (5 Km/h)
- **Unmodel dynamics**
  - both discrete and continuous

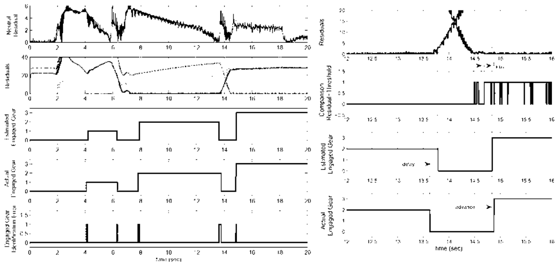
Validation against the detailed hybrid model of the driveline



Experimental results

- Obtained in Magneti Marelli Powertrain using an Opel Astra equipped with
  - a Diesel engine and a robotized gearbox SeleSpeed
- The estimated engaged gear compared to the signal from the gearbox control unit
- The proposed algorithm was tested on several maneuvers of different types, for a total of 250 gear engagements
- The actual engaged gear was successfully identified within a delay of 250 msec. in 90% of cases
- The unsuccessful cases have been obtained in very critical maneuvers such as
  - gear engagements during sharp braking
  - clutch abrupt releases
- In these cases, the residuals exhibit large oscillations

Experimental results



Conclusions

- A detailed hybrid model of the driveline has been developed
- The model has been analyzed to obtained a reduced model used for synthesis
- An algorithm for actual engaged gear identification based on hybrid observer theory has been devised
- The proposed algorithm exhibits remarkable robustness with respect to unmodel dynamics, disturbances and uncertain parameters
- The proposed algorithm has been validated by both
  - extensive simulation with the detailed hybrid model of the driveline
  - experimental data obtained with an Opel Astra equipped with SeleSpeed
- Efficient drivability control allows car manufactures to design lighter transmission systems characterized by higher elasticity, which will require the use of dynamical algorithms for actual engaged gear identification

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Hybrid Modeling and Control of the Common Rail

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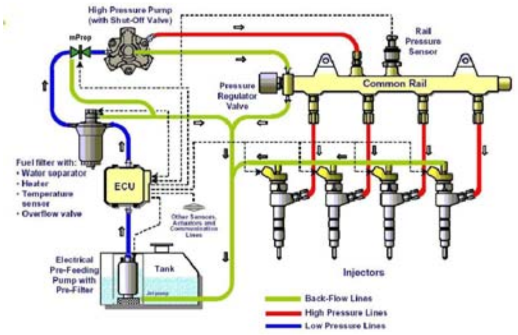
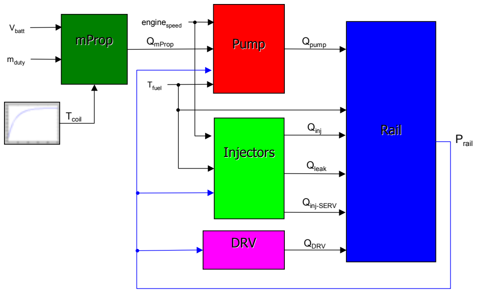
Outline

- Common rail injection system
- Detailed hybrid model of the fuel injection system
- Rail pressure controller design
- Simulation results
- Conclusions and future work

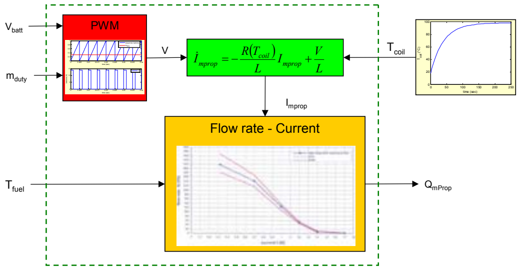
New common rail injection system developed by Magneti Marelli Powertrain



Hybrid model of the common rail injection system

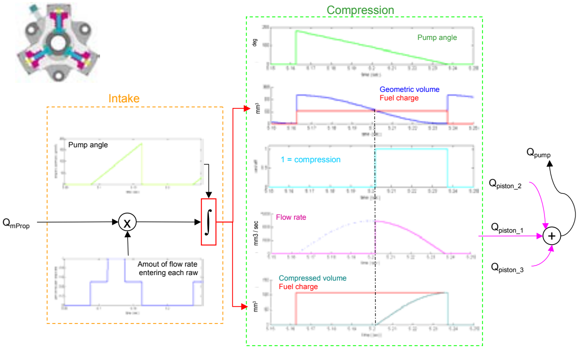


High pressure pump regulation valve (mProp)



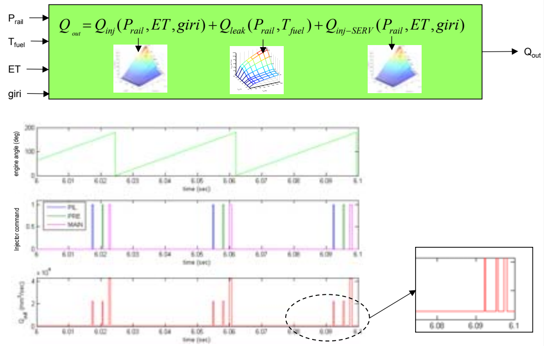
$$\dot{I}_{mprop} = -\frac{R(T_{coil})}{L} I_{mprop} + \frac{V}{L}$$

High pressure pump

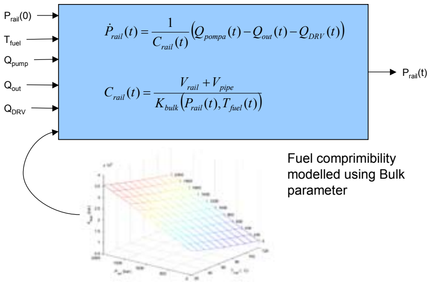


Injectors

$$Q_{out} = Q_{inj}(P_{rail}, ET, giri) + Q_{leak}(P_{rail}, T_{fuel}) + Q_{inj-SERV}(P_{rail}, ET, giri)$$



Common rail



$$\dot{P}_{rail}(t) = \frac{1}{C_{rail}(t)}\left(Q_{pompa}(t) - Q_{out}(t) - Q_{DRV}(t)\right)$$

$$C_{rail}(t) = \frac{V_{rail} + V_{pipe}}{K_{bulk}\left(P_{rail}(t), T_{fuel}(t)\right)}$$

Fuel comprimibility
modelled using Bulk
parameter

Controller design

- **Common rail pressure control**
  - **Design a tracking controller for the rail pressure that achieves tracking of a reference pressure signal generated on-line by an outer loop controller**
- **Main challenges**
  - **Time-varying delay between valve control command and rail pressure measurement**
  - **Time-varying uncertainty of the actuator electrical resistance**
- **Design approach**
  - **Smith predictor controller**
    - **loop delay compensation**
  - **Adaptive algorithm**
    - **on-line identification of the electrical resistance**
  - **First attempt: controller synthesis based on a plant mean-value model**

Mean-value model for controller synthesis

- **nonlinear CT model**

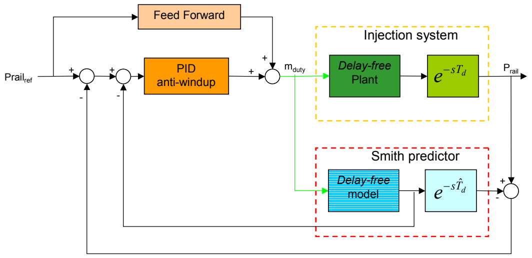
$$\begin{aligned} \dot{I}_{mProp} &= -\frac{R(T_{coil})}{L} I_{mProp} + \frac{V_{batt}}{100L} m_{Duty} \\ \dot{P}_{rail} &= \frac{K_{bulk}(P_{rail}, T_{fuel})}{V_{rail} + V_{tubi}} \cdot Q_{bil}(I_{mProp}, P_{rail}, T_{fuel}, ET, giri) \\ Q_{bil} &= Q_{pompa}(I_{mProp}, T_{fuel}, P_{rail}, giri) - Q_{inj}(P_{rail}, ET) - \\ &\quad Q_{inj-SERV}(P_{rail}, ET) - Q_{leak}(P_{rail}, T_{fuel}) \end{aligned}$$

- **piecewise polynomial CT model**

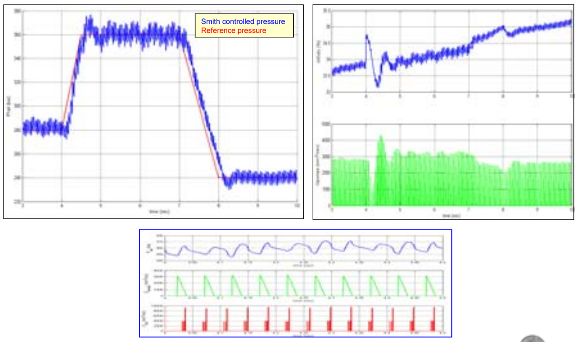
$$\begin{aligned} \dot{I} &= -\frac{R(T_{coil})}{L} I + \frac{V_{batt}}{100L} m_{Duty} \\ \dot{P} &= \frac{1}{V}\left[a_1 P^2 I + a_2 P I + a_3 P^2 + a_4 P + a_5 I + a_6\right] \end{aligned}$$

Rail pressure controller

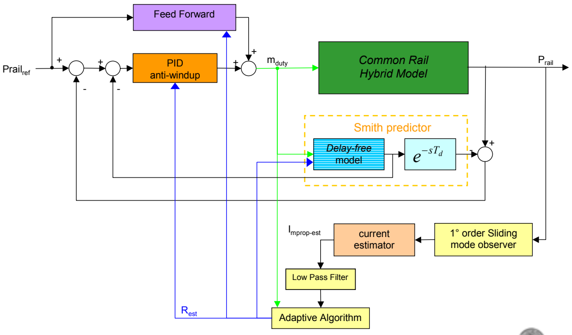
- **Sampling time 5msec**



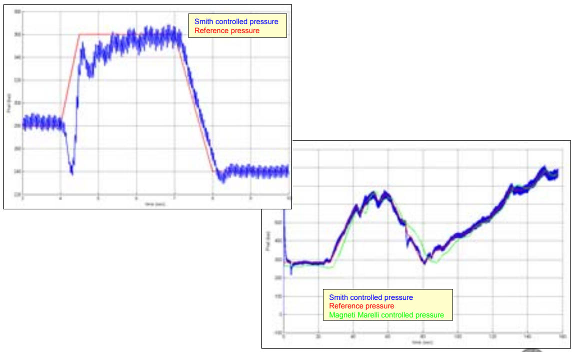
Hybrid closed-loop system simulation results



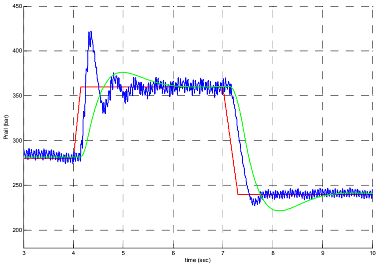
Smith Predictor and adaptive control



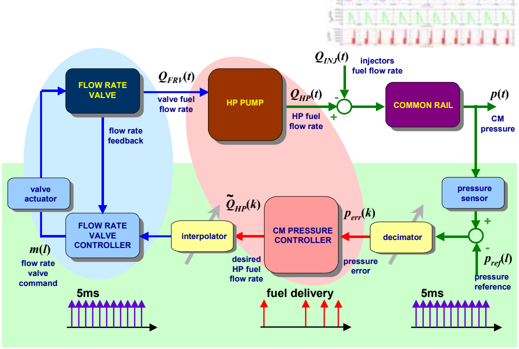
Closed-loop hybrid model simulation results



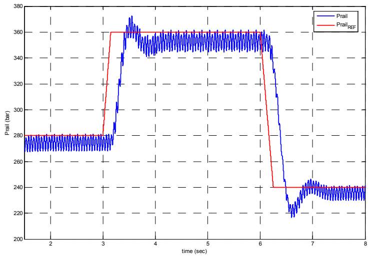
Limits of the controller designed using the mean-value model approach



Hybrid multi-rate controller



Tracking behavior of the hybrid multi-rate controller



Conclusions

- A detailed hybrid model of the common rail fuel injection system has been presented
- The hybrid model describes the pulsating evolution of the rail pressure due to HP pump supply and multiple fuel injections
- The proposed switching controller has been designed using a mean-value model of the plant and employs
  - a Smith Predictor to compensate the time-varying loop delay
  - an adaptive algorithm to adjust the static gain
- Simulation results obtained with the hybrid closed-loop model show that the controller perform satisfactorily if the reference pressure is not too fast
- Controller design based on hybrid methodologies achieves better performances and ensures tracking of fast pressure references

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