HIERARCHICAL MODELING FOR DESIGN AND OPTIMIZATION OF DIESEL ENGINE CONTROL STRATEGIES

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Applicazioni e Prospettive del controllo nei veicoli
Dei Politecnico di Milano
May 10th, 2007
Agenda

- Context and Objectives
- Common Rail benefits
- Modeling approach
- Multi-Zone Models
  - Description
  - Parameters identification
  - Results
- Two-Zone Models
  - Description
  - Parameters identification
  - Results
- Optimization and hierarchical structure
- Conclusions
**Context and Objectives**

- Meet stringent emissions standards for NOx and Soot, retain fuel economy benefits of Diesel engines.
- Improve Electronic Control for Diesel engines, critical due to the large number of control variables.
- Need to cut experiments for control strategies development to limit time and costs.
Context and **Objectives**

- **Model based structures:**
  - Hierarchical modeling
  - Real-time application
    - HIL
    - Virtual/Rapid Prototyping
    - On-board application

- Development of simulation models with satisfactory **accuracy**.

- **Boost computational speed.**

- Balanced precision among sub-models to be consistent with embedded applications.
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• Conclusions
Common Rail Control

CONTROL VARIABLES

- Injection pressure
- # of strikes
- SOI
- Pulse widths
- Dwell time
- EGR

OPEN ISSUES

- Find optimum combination(s) of parameters for given Load, Speed, EGR ratio and Boost Pressure.
- Large calibration effort.
- Introducing Model-Based optimization to reduce experiments.

Control Design
Agenda

• Context and Objectives
• Common Rail benefits
• **Modeling approach**
  • Multi-Zone Models
    – Description
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  • Two-Zone Models
    – Description
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• Optimization and hierarchical structure
• Conclusions
Phenomenological models are based on a simplified description of the physical phenomena vs multi-dimensional (3-D) approach.

A set of parameters guarantees the accuracy for different engine operations and geometry.
Hierarchical Structure

- Multi-Zone
- Two-Zone
- Black Box

ECU

- Steady State Strategy Optimization
- Dynamic Real-Time On-Board

<10 Cycles

>20 Cycles

>100 Cycles
## Models Hierarchy

<table>
<thead>
<tr>
<th>SINGLE ZONE</th>
<th>TWO ZONES</th>
<th>MULTI ZONE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single Zone</td>
<td>Mixing</td>
<td>Jet</td>
</tr>
<tr>
<td><img src="image1" alt="Single Zone" /></td>
<td><img src="image2" alt="Mixing" /></td>
<td><img src="image3" alt="Jet" /></td>
</tr>
</tbody>
</table>

- **Single Zone**
  - SAE 2004-01-1877

- **Two Zones**
  - Mixing
  - Air
  - SAE 2005-01-1121

- **Multi Zone**
  - Jet
  - Air
  - SAE 2006-01-1384

**Experiments Parameters**

**Comp. resources**
Approaches comparison

<table>
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</table>

- **SINGLE ZONE**: 16 experiments, 0.1 parameters, 120 - 3 injs
- **TWO ZONES**: 15 experiments, 5.0 parameters, 30 - 1 injs
- **MULTI ZONE**: 1 experiment, 3 parameters, 120,0

**SAE 2004-01-1877**
**SAE 2005-01-1121**
**SAE 2006-01-1384**
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Multi-zone Model
Features and Structure

FEATURES

• High precision vs. computational time.
• Easy-to-hand parameters identification.
• Decoupling of some phenomena.
• Modularity.
• Balanced precision.

MODULES (sub-models)

Evap.
Multi Zone Model

Energy
\[
\dot{E}_i = \dot{Q}_i - \dot{W}_i + \sum_{j, i \neq j} \dot{m}_{i,j} h_{i,j}
\]

Volume
\[
V_{cyl} = V_a + \sum_i V_i
\]

Variables
\[
[p, T_{i,j}^{vb}, T_{i,j}^{vu}, T_a]
\]
Injection Delay

<table>
<thead>
<tr>
<th>ET</th>
<th>Energizing Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>ED</td>
<td>Energizing Delay</td>
</tr>
<tr>
<td>COD</td>
<td>Control Valve Opening Delay</td>
</tr>
<tr>
<td>CCD</td>
<td>Control Valve Closing Delay</td>
</tr>
<tr>
<td>NOD</td>
<td>Needle Opening delay</td>
</tr>
<tr>
<td>ISD</td>
<td>Injection Start Delay</td>
</tr>
<tr>
<td>NCD</td>
<td>Needle Closing Delay</td>
</tr>
<tr>
<td>IED</td>
<td>Injection End Delay</td>
</tr>
<tr>
<td>EID</td>
<td>Effective Injection Duration</td>
</tr>
</tbody>
</table>

ISD = \( \frac{C_1}{V_{f,\text{inj}}} \)

EID = \( C_2 \cdot V_{f,\text{inj}} \cdot ET \)
Jet Development

- equal mass and momentum fluxes as the equivalent real spray at the same axial location;
- uniform velocity profile;
- constant injection velocity;
- no velocity slip between the fuel and the entrained air;
- conical shape.
- non-dimensional penetration by integrating the non-dimensional velocity (Naber and Siebers, 96)

The air entrained by each zone is computed from the momentum conservation:

\[ m_{ae} = -C_3 \frac{m_{f,inj} U_f}{\left( \frac{dS}{dt} \right)^2} \frac{d^2S}{dt^2} \]
Evaporation

After the break-up, the fuel evaporation rate is derived from mass diffusion and heat transfer for a spherical droplet with initial diameter equal to the SMD (Jung & Assanis, 2001):

\[
\dot{m}_{f,v} = \pi d_1 N D_v S_h \frac{p}{R_v T^m} \ln\left(\frac{p}{p - p_{v,\text{surf}}}\right)
\]

\[
\dot{q} = \pi d_1 N k^m (T^{v,u} - T_1) \text{Nu} \left(\frac{Z}{e^Z - 1}\right)
\]

The droplet temperature is assumed homogeneous

Energy balance \(\rightarrow\)

\[
\frac{dT_1}{dt} = \frac{1}{m_1 c_{p,l}} \left(\dot{q} - \lambda \frac{dm_{f,v}}{dt}\right)
\]
Ignition Delay

The ignition delay is computed with an Arrhenius-like model (Heywood, 88; Jung & Assanis, 95).

\[ \tau_{id} = 3.45 \cdot 10^{-3} p^{-1.02} \cdot \exp \left( \frac{2100}{T} \right) \]

To account for pressure and temperature variation the following integral is solved with respect to SOC

\[ \int_{\text{SOI}}^{\text{SOC}} \frac{dt}{\tau_{id}} = 1 \]
Turbulence

The turbulence is described assuming isotropic homogeneous turbulence and equilibrium between production and dissipation of turbulent kinetic energy.

The combustion does not influence directly the turbulence.

\[
\begin{align*}
\frac{dk}{dt} &= \frac{2}{3} \frac{k}{\rho} \frac{dp}{dt} - \varepsilon \\
\frac{d\varepsilon}{dt} &= \frac{4}{3} \frac{\varepsilon}{\rho} \frac{dp}{dt} - \frac{2\varepsilon^2}{k}
\end{align*}
\]

The initial value of \( k \) is proportional to the mean piston speed.

\[
k(IVC) = \frac{3}{2} \left( B_1 V_{mp} \right)^2
\]

The initial value of \( \varepsilon \) is derived from the equilibrium hypothesis (\( L_1 \) valve lift).

\[
\varepsilon(IVC) = \frac{\left[ k(IVC) \right]^{3/2}}{L_1(IVC)}
\]
Combustion Model

The combustion rate is modeled as function of a “characteristic-time”, which is the weighted sum of the laminar and turbulent combustion time scales.

\[
\frac{dm_b}{dt} = \frac{m_e - m_b}{\tau_b} \quad \tau_b = \tau_{b,\text{lam}} + \gamma \tau_{b,\text{turb}}
\]

The laminar time scale is derived from an Arrhenius-like relationship (Kaario et al., 2002). The turbulent combustion time scale is assumed proportional to the eddy turnover.

\[
\tau_{b,\text{lam}} = \left[ 4 \times 10^3 \left( n_{fv} \right)^{-0.75} \left( n_{O_2} \right)^{1.5} \exp\left( -\frac{E}{\tilde{R} T^{vb}} \right) \right]^{-1} \]

\[
\tau_{b,\text{turb}} = 0.142 \frac{k}{\varepsilon}
\]
The **NO model** is based on the well known Zeldovich mechanism:

\[ O + N_2 \leftrightarrow NO + N \]
\[ N + O_2 \leftrightarrow NO + O \]
\[ N + OH \leftrightarrow NO + H \]

**Formation rate**

\[ \frac{1}{V_b} \frac{dn_{NO}}{dt} = \frac{2 R_f \left\{ 1 - \left( \frac{[NO]}{[NO]_e} \right)^2 \right\}}{1 + \left( \frac{[NO]}{[NO]_e} \right) R_f / (R_2 + R_3)} \]

The **Soot model** is based on the Hiroyasu approach:

**Net soot mass rate**

\[ \frac{dM_s}{dt} = \frac{dM_{sf}}{dt} - \frac{dM_{so}}{dt} \]

**Mass formation rate**

\[ \frac{dM_{sf}}{dt} = K_f M_{fv}^n \]

**Mass oxidation rate**

\[ \frac{dM_{so}}{dt} = K_o M_s x_{o2} \]
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Engine

<table>
<thead>
<tr>
<th>Engine</th>
<th>FIAT 1.9 JTD 16v</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cycle</td>
<td>Diesel</td>
</tr>
<tr>
<td>Strokes</td>
<td>4</td>
</tr>
<tr>
<td>Cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Valves</td>
<td>16</td>
</tr>
<tr>
<td>Bore (mm)</td>
<td>82</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>90.4</td>
</tr>
<tr>
<td>Displacement (cm³)</td>
<td>1909</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>18</td>
</tr>
<tr>
<td>Connecting rod to stroke ratio</td>
<td>0.3177</td>
</tr>
</tbody>
</table>
The parameters related to "general" physical phenomena (e.g. evaporation, combustion, heat transfer) have been taken from the literature.

The parameters characterizing the injection system ($C_1$, $C_2$) and the jet-air-geometry interaction ($C_3$) have been identified in one reference point.

- 2000 @ 9
- EGR 13%
- 2 inj.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_1$ [$\mu$s mm$^3$]</td>
<td>3618.4</td>
</tr>
<tr>
<td>$C_2$ [1/mm$^3$]</td>
<td>0.0475</td>
</tr>
<tr>
<td>$C_3$ [/]</td>
<td>1.8</td>
</tr>
</tbody>
</table>
Parameters Identification

Engine: Fiat 1.9 16 V M-Jet

The unknown parameters have been identified by comparing predicted and measured pressure cycle.

$p_{\text{max,m}} - p_{\text{max,c}} = 0.26 \text{[bar]} \quad \Delta \theta \bigg|_{p_{\text{max}}} = 0.3 \text{[deg]}$
The model has been tested versus a wide set of the experimental data composed of 89 engine cycles.

**CONTROL VARIABLE | RANGE**
--- | ---
Injections | 1; 3
Injection timing | 47°; -2° BTDC
Dwell angle | 10°; 32°
Fuel injected/cycle | 5; 71 mm³
Fuel injected/strike | 1; 70 mm³
$P_{\text{rail}}$ | 300; 1400 bar
EGR | 0; 45 %

Average relative error  **1%**
Max relative error  **<10%**
Standard deviation  **4%**
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Results – Engine Cycle

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed (rpm)</td>
<td>1500</td>
</tr>
<tr>
<td>Brake Mean Effective Pressure (bar)</td>
<td>5</td>
</tr>
<tr>
<td>EGR Ratio (%)</td>
<td>27</td>
</tr>
<tr>
<td>Number of injections</td>
<td>2</td>
</tr>
</tbody>
</table>
Results – Engine Cycle

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed (rpm)</td>
<td>2000</td>
</tr>
<tr>
<td>Brake Mean Effective Pressure (bar)</td>
<td>5</td>
</tr>
<tr>
<td>EGR Ratio (%)</td>
<td>26</td>
</tr>
<tr>
<td>Number of injections</td>
<td>2</td>
</tr>
</tbody>
</table>
Results – Engine Cycle

Discretization
20 zones

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>2000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake Mean Effective Pressure (bar)</td>
<td>13</td>
</tr>
<tr>
<td>EGR Ratio (%)</td>
<td>0</td>
</tr>
<tr>
<td>Number of injections</td>
<td>2</td>
</tr>
</tbody>
</table>

Discretization
10 zones
Results – Engine Cycle

- Engine Speed (rpm): 4000
- Brake Mean Effective Pressure (bar): 9
- EGR Ratio (%): 0
- Number of injections: 1

Heat Release In. Cond.
Results – NO Emissions

Comparison between measured and estimated NO

Frequency distribution of the NO Relative Error

$R^2 = 0.94$

1000@10

4500@13

Frequency

Cumulative

Distribution

70%

30%
Results – NO Emissions

![Graph showing NO emissions with and without EGR and temperature increase.]

- **Engine Speed (rpm)**: 1500
- **2 Injections with EGR**
- **3 Injections without EGR**
- **Temperature increase**
Results – NO Emissions

![Graph showing NO emissions with and without EGR with engine speed of 2500 rpm.](image-url)
Results – Soot Emissions

- 2 Injections with EGR
- 2 Injections without EGR
- Temperature increase

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>2500</th>
</tr>
</thead>
</table>
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Two Zone Model

- Liquid fuel
- Fuel Preparation
- Air
- Combustion
Thermodynamic Model

Energy Equation
\[ \dot{E}_i = \dot{Q}_{w,i} - \dot{W}_i + \sum_{j, \ i \neq j} \dot{m}_{i,j} \cdot h_{i,j} \]
\[ \forall i = a, m \quad \forall j = a, l, m \]

Volume Equation
\[ V_{cyl} = V_a + \sum_i \sum_j V_{i,j} \]
\[ \forall i = a, l, m \quad \forall j = 1 \ldots n_{nh} \]

\[ \dot{p} = f(p, T_a, T_m, \Phi, \theta) = \frac{A + B \cdot L_a + C \cdot L_m}{G_a} - \frac{D - B \cdot F_a}{G_a} - \frac{C \cdot F_m}{G_m} \]
\[ \dot{T}_a = g(p, T_a, T_m, \Phi, \theta) = \frac{L_a + F_a \cdot \dot{p}}{G_a} \]
\[ T_m = h(p, T_a, T_m, \Phi, \theta) = \frac{L_m + F_m \cdot \dot{p}}{G_m} \]
Thermodynamic model

Injection Delay

Spray Model

\[ \frac{d s_{bb}}{d \theta} = \frac{C_D}{\omega} \sqrt{\frac{2 \cdot (p_{rail} - p)}{\rho_i}} \]

\[ \frac{d s_{ab}}{d \theta} = 1.475 \left( \frac{p_{rail} - p}{\rho_a} \right)^{0.25} \sqrt{\frac{d_n}{\omega} \cdot (\theta - \theta_{SOI})^{-0.5}} \]

\[ t_b = 4.351 \frac{\rho_i \cdot d_n}{C_D \cdot \sqrt{\rho_a \cdot \Delta p}} \]
The time for fuel atomization, vaporization and micromixing with entrained air is described by means of a fuel preparation rate [Whitehouse & Way]:

\[
\dot{m}_{f,p}(\theta) = C_1 \left( \frac{\int_0^\theta dm_{f,inj} d\theta}{d\theta} \right)^{\frac{1}{3}} \cdot \left( p_{02}(\theta) \right)^{0.4} \cdot \left( \frac{\int_0^\theta dm_{f,inj} d\theta}{d\theta} - \int_0^\theta \frac{dm_{f,p}}{d\theta} d\theta \right)^{\frac{2}{3}}
\]

The fuel burning rate depends on the amount of available fuel, weighted by an Arrhenius term [Whitehouse & Way]:

\[
\dot{m}_{f,b}(\theta) = \frac{C_2 \cdot p_{02}(\theta)}{N' \cdot \sqrt{T_{\text{mean}}(\theta)}} \cdot e^{\left( \frac{T_A}{T_{\text{mean}}(\theta)} \right)} \cdot \int_0^\theta \left( \frac{dm_{f,p}}{d\theta} - \frac{dm_{f,b}}{d\theta} \right) d\theta
\]
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Parameters Identification
Fiat 1.9 16 V M-Jet

The parameters related to “general” physical phenomena (e.g. spray, combustion, heat transfer, NO and Soot) have been taken from the literature.

The parameters characterizing the injection system ($t_{\text{inj,d}} ; C_D$) have been identified in 9 reference points.

Multiple regressions have been derived to express parameters variation vs. engine operation.

<table>
<thead>
<tr>
<th>No</th>
<th>Engine Speed [rpm]</th>
<th>bmep [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4500</td>
<td>0.5</td>
</tr>
<tr>
<td>2</td>
<td>4500</td>
<td>3</td>
</tr>
<tr>
<td>3</td>
<td>4500</td>
<td>5</td>
</tr>
<tr>
<td>4</td>
<td>4500</td>
<td>9</td>
</tr>
<tr>
<td>5</td>
<td>4500</td>
<td>12</td>
</tr>
<tr>
<td>6</td>
<td>4000</td>
<td>11</td>
</tr>
<tr>
<td>7</td>
<td>4000</td>
<td>16</td>
</tr>
<tr>
<td>8</td>
<td>3500</td>
<td>9</td>
</tr>
<tr>
<td>9</td>
<td>3500</td>
<td>12</td>
</tr>
</tbody>
</table>

\[
t_{\text{inj,d}}[ms] = b_1 - b_2 \cdot (q_{\text{fuel}} \cdot N)
\]

\[
C_D = a_1 - a_2 \cdot (q_{\text{fuel}} \cdot N) \quad \text{if} \quad q_{\text{fuel}} \cdot N \geq \frac{1-a_1}{a_2}
\]

\[
C_D = 1 \quad \text{if} \quad q_{\text{fuel}} \cdot N < \frac{1-a_1}{a_2}
\]
Generalization Test

The model has been tested versus a wide set of experimental data composed of 81 engine cycles.

<table>
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<td>Injections</td>
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<tr>
<td>Dwell angle</td>
<td>10°; 32°</td>
</tr>
<tr>
<td>Fuel injected/cycle</td>
<td>5; 71 mm³</td>
</tr>
<tr>
<td>Fuel injected/strike</td>
<td>1; 70 mm³</td>
</tr>
<tr>
<td>P_{rail}</td>
<td>300; 1400 bar</td>
</tr>
<tr>
<td>EGR</td>
<td>0; 45 %</td>
</tr>
</tbody>
</table>

\[ R^2 = 0.985 \]
Validation – In-Cylinder Pressure

Engine Speed  1500 rpm
BMEP  5 bar
EGR  27 %
Pre and main injection

Engine Speed  2000 rpm
BMEP  9 bar
EGR  12.7 %
Pre and main injection
Validation – In-Cylinder Pressure

**Engine Speed** 3000 rpm
BMEP 3 bar
EGR 16.8 %
Pre and main injection

**Engine Speed** 4000 rpm
BMEP 15 bar
EGR 0 %
Main injection
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Results – NO emissions

- **1500 rpm**
- **2500 rpm**
- **3000 rpm**
- **4000 rpm**
Results – Soot emissions

- **1500 rpm**
- **2500 rpm**
- **3000 rpm**
- **4000 rpm**
Results – Effects of Prail

![Graph showing the effects of Prail on soot and NO emissions with load and IMEP at 2500 rpm. The graph includes data points for different IMEP values with their respective confidence intervals.

- **Base condition**
- **Prail**

**Key Points**
- IMEP=7 bar +/-1.5%
- IMEP=11.5 bar +/-2.1%
- IMEP=15.3 bar +/-2.8%
- IMEP=17.2 bar +/-4.2%
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DPF filter enhances significant reduction of soot emissions
Optimization

An Optimization analysis has been performed on 6 operating conditions aimed at minimizing NO emissions with constraints on Soot and IMEP.

The optimization is effectively done in 20 minutes per point on a IBM Xeon 3.2 GHz; computational time per cycle varies from less than 1 second (1 inj.) to 3 seconds (2 inj.) and to 7 seconds (3 inj.)

\[
\min_{V_{inj}, \theta_{inj}, P_{rail}, EGR} NO(V_{inj}, \theta_{inj}, P_{rail}, EGR)
\]

\[
\left\{ \begin{array}{l}
\text{Soot} - \text{Soot}_{\text{base}} < 5% \\
\Delta \text{IMEP} < 1%
\end{array} \right.
\]
Optimization

Soot

- Base: +4.6%
- Optimal: +4.5%

Rail Pressure

- Base: -15.5%
- Optimal: -29.2%
Hierarchical Structure Application

- A first attempt has been performed to identify the Two-Zone model via Multi-Zone generated pressure cycles.
- The hierarchical modeling structure guarantees an accuracy level comparable to direct identification from experiments.

![Graph showing Predicted IMEP (bar) vs. Measured IMEP (bar) with R^2 = 0.9907 and 21 cycles.](image)
Hierarchical Structure Application

Multi-Zone → Two-Zone

21 cycles

NO [ppm]

Two-zone
Measured
Multi-zone

bmeq [bar]

0
5
10
15
20
25

0
500
1000
1500
2000
2500

2500 rpm

Soot [g/cycle]

Two-zone
Measured
Multi-zone

bmeq [bar]

0
5
10
15
20
25

2.5
3
3.5

2500 rpm

1500 rpm
Hierarchical Structure Application

- <10 Cycles
  - Multi-Zone

- >100 Cycles
  - Neural Network
  - Vehicle Dynamic Simulation
    - \( u(t) \) → \( y(t) \)
Mean Value Model for transient simulation of Turbocharged CI engine
Mean Value Model for transient simulation of Turbocharged CI engine
Results

massa di combustibile iniettato [mm³/colpo]

regime del turbocompressore [rpm]

pressione collettori [bar]

temperatura nei collettori [K]
Results

- ECE/EUDC Driving Cycle
- Reference Vehicle: Alfa Romeo 147 1.9 JTDm
- Fuel consumption ECE/EUDC cycle = 16.84 km/l
  - [provided by Alfa Romeo = 16.95 km/l]
- Acceleration 0-100 km/h = 8.3 s
  - [provided by Alfa Romeo = 8.8 s]
Results

Air Flow Rate [kg/s]

Torque [Nm]

EGR Flow Rate [kg/s]

Injected Fuel [mm³/cycle]
Conclusions

• A Hierarchical modeling structure has been developed for the design and optimization of Diesel engine control strategies.

• A satisfactory compromise between accuracy, computational time and experimental effort has been effectively achieved by “cascading” phenomenological models (Multi/Two-zone).

• The models accurately simulate pressure cycles and emissions (NOx and Soot) in Common-Rail Multi-Jet Diesel Engine.

• In the Multi-Zone model three parameters (Ignition delay and air entrainment) have been identified using one engine cycle.

• In the Two-Zone model two parameters (Ignition delay and discharge coefficient) have been identified using nine experimental cycles or 21 cycles generated via multi-zone model.

• The Two-Zone model can be used for investigating the effects of control parameters and for optimization analyses aimed at improving fuel efficiency and emissions.

• A Mean Value Model has been developed for the dynamic simulation of engine/vehicle transients and tested vs. literature data.
Air Entrainment

Momentum conservation

\[
\left( m_i \cdot \frac{dS_{bb}}{d\theta} \right)_0 = \left( m_i + m_{ae} \right) \cdot \frac{dS_{ab}}{d\theta}
\]

Before Break-up  After Break-up

Air mass entrainment

\[
m_{ae} = m_i \left( \left( \frac{dS_{bb}}{d\theta} \right)_0 \cdot \left( \frac{dS_{ab}}{d\theta} \right)^{-1} \right) - 1
\]

Air mass flow entrainment

\[
\frac{dm_{ae}}{d\theta} = -m_i \left( \frac{dS_{bb}}{d\theta} \right)_0 \cdot \left( \frac{dS_{ab}}{d\theta} \right)^{-2} \left( \frac{d^2 S_{ab}}{d\theta^2} \right)
\]