Combustion control for reduced fuel consumption and emissions

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New engine lab
Control room
Test cell
History

• $\lambda$-control first modern application
  – Adjust fuel injection based on oxygen level in exhaust

• Initially
  – Throttle body injection
  – Oxygen sensor far downstream in exhaust

• Both actuation and sensing gradually moving closer to cylinder
Conventional combustion

• Conventional combustion concepts
  – Spark ignition
    • Hot stoichiometric flame propagation
      – \( NO_x \) emissions
    • Throttling at part load
      – Poor efficiency
  – Diesel combustion
    • Hot stoichiometric diffusion flame
      – \( NO_x \) emissions
New combustion concepts

• Low temperature combustion
  – Avoids high temperature through dilution
  – Avoids soot through lean operation

• Homogeneous charge compression ignition (HCCI)
  – Completely premixed, kinetically controlled

• Partially premixed combustion (PPC)
  – Partially premixed, mix of injection control and kinetic control

• Reactivity controlled compression ignition (RCCI)
  – Pilot injection of reactive fuel triggers combustion
HCCI needs feedback control!

- No direct control of ignition
  - Combustion starts when the conditions are right
- Ignition process very sensitive
  - Impossible to map with sufficient accuracy
- Instability in certain operating points
  - Operating point can not be maintained without active control
  - Feedback control necessary for stabilization
New combustion concepts

Spark Ignition (SI) engine (Gasoline, Otto)
- + Clean with 3-way Catalyst
- - Poor low & part load efficiency

Compression Ignition (CI) engine (Diesel)
- + High efficiency
- - Emissions of NO\(_x\) and soot

Homogeneous Charge Compression Ignition (HCCI)
- + High efficiency
- + Ultra low NO\(_x\)
- - Combustion control
- - Power density

Spark Assisted Compression Ignition (SACI) Gasoline HCCI
- + Injection controlled
- - Less emissions advantage

Partially premixed combustion (PPC) Diesel HCCI
CA50 based on cylinder pressure

\[
\frac{dQ}{dCA} = \frac{\gamma}{\gamma - 1} p \frac{dV}{dCA} + \frac{1}{\gamma - 1} V \frac{dp}{dCA}
\]

Heat release trace steepest at CA50

\[\Rightarrow\]
Minimal crank angle error
Ion current measurement
Heat release – Ion Current

Crank angle [CAD]

50%
CA50 – 50% Ion rise

![Graph showing the relationship between Ion rise 50% and CAD 50% burned.](image-url)
Dual-fuel HCCI

- Two fuels with different reactivity can be used to control combustion timing

![Graph showing the relationship between combustion timing and octane number](chart.png)
System outline

- Based on heavy duty Scania Diesel engine
- PFI of two fuels
- Pressure sensors in all cylinders
- Intake air heating
- Unaltered combustion geometry, CR 18:1
Method

• Cylinder individual PID controllers, two per cylinder
  – One for CA50
  – One for IMEP (Load)

• Gain scheduling based on sensitivity model for CA50
Ignition Step

Step Response

- CA 50% HR
- Octane Number
- net IMEP [bar]

Cycles
HCCI control with VVT

Both EIVC and LIVC possible for control of CA50
System identification

Multi-level pseudo-random sequence used as excitation

Subspace method
2nd order model
LQG control of HCCI using LIVC

Good disturbance rejection

Significant difference in IVC requirement between different cylinders
Control of Diesel/ethanol fumigation

- Port injection of ethanol
- Direct injection of Diesel fuel
- Load control using ethanol
- Noise control using Diesel fuel
Improvement with control
Midranging Control

- Multivariable control method
- Suitable for systems with
  - One slow actuator with high authority
  - One fast actuator with limited authority
- Two controllers
  - $C_1$ controls system output
  - $C_2$ controls fast actuator to middle of its range
- Assures high-bandwidth operation at all times
Midranging HCCI control based on IVC and EGR

- Fast actuator: IVC timing
- Slow actuator: EGR ratio (backpressure)
- Controlled output: CA50
- Multi-cylinder engine ⇒ Many IVC timings
  - Control e.g. the mean IVC timing to middle of range
Midranging of IVC using EGR
Step change of CA50

Constant EGR

![Graph of Constant EGR]

- $\alpha_{50}$
- $u_{IVC}$
- $\bar{u}_{IVC}$
- $u_{EGR}$

Midranging using EGR

![Graph of Midranging using EGR]

- $\alpha_{50}$
- $u_{IVC}$
- $\bar{u}_{IVC}$
- $u_{EGR}$

Engine cycle
Midranging of IVC using EGR

Speed transient

Constant EGR

Midranging using EGR
PPC definition

- Mixing period (MP)
  - CA difference between EOI and SOC
- PPC=positive MP
System identification

• Model inputs
  – Start of injection (SOI)
  – Injection duration (ID)

• Model outputs
  – CA50
  – IMEP
  – MP
Model fidelity

![Graphs showing model fidelity comparison between measured output and model output over engine cycles.](image)
Controller design

• Model predictive control (MPC
  – Essentially constrained online optimization
• Setpoints for
  – IMEP (driver demand)
  – MP (>0 for PPC)
• Constraints for
  – CA50
Controller performance

Load steps

EGR step
Cylinder-individual efficiency

- IMEP computed from cylinder pressure
- Self-tuning heat release
  - Assume complete combustion $\Rightarrow$ Fuel input
- Ratio between the two gives efficiency
IMEP

- Normalized measure of work output
- Work performed by cylinder gases on piston
- Computed from measured cylinder pressure

\[ IMEP \cdot V_d = W_c = \int_{cykel} p \, dV \]

\[ IMEP = \frac{W_c}{V_d} = \frac{1}{V_d} \int_{cykel} p \, dV \]
Self-tuning heat release

• Heat losses implicitly modeled by using estimated polytropic exponent instead of specific heat ratio in HR

• Estimation during
  – Compression
  – Expansion
Control architecture

- Extremum seeking added to existing control
- Perturbs combustion phasing (CA50) and evaluates result
- Moves in favorable direction
Extremum seeking algorithm

$$\Delta CA50_{\text{ref}}(j) = \Delta CA50_{\text{ref}}(j-1) + \bar{\sigma}(j)$$

$$\bar{\sigma}(j) = \begin{cases} 
\sigma(j), & |\sigma(j)| \leq \sigma_{\text{max}} \\
\sigma_{\text{max}} \cdot \text{sgn}(\sigma(j)), & |\sigma(j)| > \sigma_{\text{max}} 
\end{cases}$$

$$\sigma(j) = \frac{1}{M} \sum_{k=(j-1)M}^{jM-1} \rho(k)$$

$$\rho(k) = (\eta_{\text{est}}(k) - \eta_{\text{est}}(k-1)) \cdot (CA50(k) - CA50(k-1))$$
Experimental setup

- 13 liter Volvo Diesel engine
- Cylinder pressure measured in all cyls
- AC dynamometer maintains engine speed
Control evaluation

• Extremum seeking control evaluated for steady-state engine speed and load over 2000 engine cycles
  – 1200 rpm
  – 8 bar IMEP

• Two cases
  – Forced sinusoidal excitation
  – Natural excitation from cycle-cycle variation
Extremum seeking – Forced excitation

• Extremum seeking switched on at cycle 100
  – Sinusoidal excitation with 1 CAD amplitude and 10 cycle period

• Convergence to optimal CA50 in approx. 500 cycles
Extremum seeking – Natural excitation

- Extremum seeking switched on at cycle 100
  - No forced excitation
  - Natural cycle-cycle variation utilized
- Convergence to optimal CA50 in approx. 2000 cycles
- Less disturbance on IMEP than with forced excitation
Dilution-limit control

- The throttling of an SI engine can be reduced by increasing the dilution
  - Air – lean
  - EGR – inert exhaust gas

- But ignition quality deteriorates
- Apply control to target e.g. 5% COV
What type of EGR?

- Two types of EGR
  - Short-route – high-pressure side of turbo charger
    - Tighter packing but less flexible EGR / Airpath system
  - Long-route – low-pressure side of turbo charger
    - More bulky piping but more flexible

- What about dynamics?
  - EGR path longer with long route
    - Maybe slower dynamics…
  - But short-route affects flow through turbine
    - Could also cause slower dynamics
Long-route vs. Short-route

- Much faster increase with long-route
  - Short-route has to wait for turbo to speed up
- Somewhat faster decrease with short-route
  - EGR valve closes EGR flow immediately
Transient COV

\[ COV = \frac{\sum_{k=1}^{n}(IMEP_k - \overline{IMEP})}{n - 1} \]

- Problems with normal COV during transients
  - COV – normalized standard deviation over large number of cycles
    - Long time delay – reduces bandwidth of closed-loop system
  - Mean value changes during transients
    - Trend interpreted as cycle-cycle variation
  - Different levels of solutions
    - Filtered mean value
    - Filtered mean value and variance
Filtered mean value

- Replace mean value by filtered IMEP to remove deterministic change from COV

\[ p_{m,f}(k + 1) = \lambda_m p_{m,f}(k) + (1 - \lambda_m) p_m(k) \]
Efficiency improvement

- Approximately 5% increase of efficiency throughout load range
Future trends

• Continued trend with better actuation inside combustion chamber
  – Faster injectors (piezo)
  – Continuously variable injection rate
• Cylinder pressure sensing becomes main stream
• Increased model complexities
  ⇒ More computational power required
Ultimate combustion control system

- Feedback control of combustion while it happens
  - in-cycle combustion control
  - Shape combustion as it happens!

\[ \text{Controller} \rightarrow \text{Combustion} \rightarrow \text{Sens} \]

\[ \text{Inj} \rightarrow \text{Combustion} \rightarrow \text{Sens} \]

\[ \sim 10-100 \, \mu s \, \text{loop time} \]
Computation for In-cycle combustion control

- Processor based systems slow
  - One instruction per clock cycle
- Arrange time consuming computation in hardware
  - Almost infinite number of operations per clock cycle
- Convenient to use reconfigurable hardware
  - FPGA
Virtual heat release sensor

Simulated engine

Signal conditioning

~100 ns

Oscilloscope
Algorithm

Designed using Simulink

\[ \frac{1}{\gamma - 1} p(\theta)V(\theta) + \int_{\theta_{\text{start}}}^{\theta} p(\theta) \frac{dV}{d\theta} d\theta \]

Calculated on-line
Pressure and Heat release acquired simultaneously using oscilloscope.
Performance

• Very high performance
  – Current FPGA HR “algorithm” introduces a delay of ~12 clock cycles => 120 ns delay
  – Engine @ 1200rpm moves 0.000864 CAD within 100ns, (0.01728 CAD @ 24000rpm)

• Negligible latency, virtual Q sensor!
Summary – Actuators

• Actuators move closer and closer to combustion
  – Reduces time between actuation and combustion response
  – Increases quality of actuation
    • Injection timing and pressure control
    • Multiple injections
    • Valve control
Summary – Sensors

• Sensors move closer and closer to combustion
  – Reduces time between combustion and measurement
  – Increases quality of measurement
    • Direct combustion measurement using pressure or ion current
Summary – Computation

• More and more computational power required
  – Advanced control methods computationally expensive
    • E.g. MPC
  – Low computational latency necessary as time between actuation and measurement shrinks