High-Efficiency, Ultra-Low Emission Combustion in a Heavy-Duty Engine via Fuel Reactivity Control

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Outline

- Motivation

- Experimental Results
  - Gasoline PPC

- CFD Modeling – Fuel reactivity

- Experimental Results
  - Dual-fuel PCCI

- Conclusions
Motivation

• Concern for improved fuel efficiency – GHG, economy

• Emissions regulations
  EPA 2010 on-highway HD
  Euro 5,6

LTC (MK, PCCI, HCCI, etc.)

Advantages
  Low NOx and PM emissions
  High thermal efficiency

Disadvantages
  Load limits from high PRR
  No direct control of combustion timing

PPC – “hybrid”
  between HCCI and diesel LTC
  Kalghatgi “Mixed enough” combustion

Park & Reitz, CST, 2007
Low emissions window
Motivation

Partially Premixed Combustion

- Increase ignition delay to add mixing time

2 ways to achieve PPC

- **High EGR rates**
  - Reduce PM formation with low combustion temperatures
    (Akihama SAE 2001-01-0655)

- **Fuels**
  - Use low CN fuels and EGR to add ignition delay
    (Kalghatgi SAE 2007-01-0006)
  - Optimize fuel reactivity
    (Bessonette SAE 2007-01-0191)
## Diesel vs. gasoline compression ignition

### Kalghatgi: SAE Paper 2007-01-0006

<table>
<thead>
<tr>
<th>Engine</th>
<th>heavy-duty, flat cylinder head, shallow bowl</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore x Stroke [mm]</td>
<td>127 x 154</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>14.0</td>
</tr>
<tr>
<td>Diesel injector</td>
<td></td>
</tr>
<tr>
<td>Number of holes, diameter [μm]</td>
<td>8, 200</td>
</tr>
<tr>
<td>Operating conditions</td>
<td></td>
</tr>
<tr>
<td>Engine speed [rpm]</td>
<td>1200</td>
</tr>
<tr>
<td>Swirl ratio</td>
<td>2.4</td>
</tr>
<tr>
<td>Intake temperature [C], Pressure [bar]</td>
<td>40, 2.0</td>
</tr>
<tr>
<td>Oxygen fraction @ IVC/EGR [%]</td>
<td>15.8/25</td>
</tr>
<tr>
<td>Pilot split ratio [%]</td>
<td>30</td>
</tr>
</tbody>
</table>

### Injection profile

- Normalized injection velocity
- Crank angle [deg atdc]
- Injection profile
- 30% and 70% marks on the graph
Numerical models

**Single Zone Simulations**
SENKIN engine code
ERC reduced PRF mechanism
  41 species, 130 reactions – Ra & Reitz CNF, 2008

**Multi-Dimensional Modeling**
KIVA-3V code coupled with CHEMKIN II
RNG k-ε turbulence model
KH-RT drop break up model
Grid-independent spray models
Drop collision and coalescence
ERC reduced PRF mechanism

KIVA Modeling - Ra, Yun, Reitz
Int. J. Vehicle Design 2009
Diesel vs. gasoline - double injection

Start of injection: -137 and -6 (diesel), -9 (gasoline) deg atdc.
- Measured (Kalghatgi et al. SAE 2007)
Model: ERC KIVA-CHEMKIN w/ PRF mechanism
Diesel vs. gasoline - ignition delay

**Diesel SOI = -2**
- CA = tdc
- CA = +4
- CA = +6
- CA = +12

**Gasoline SOI = -11**
- CA = -8
- CA = tdc
- CA = +10
- CA = +12

*temp [K]*
- 2000
- 1750
- 1500
- 1250
- 1000
Additional time for mixing of gasoline offers benefits for CIDI engines!
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ERC Caterpillar engine lab

**3401E SCOTE**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement (l)</td>
<td>2.44</td>
</tr>
<tr>
<td>Geometric Comp. Ratio</td>
<td>16:1</td>
</tr>
<tr>
<td>Bore (mm)</td>
<td>137</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>165</td>
</tr>
<tr>
<td>Number of Valves</td>
<td>4</td>
</tr>
<tr>
<td>IVC (BTDC modified cam)</td>
<td>85/143</td>
</tr>
<tr>
<td>Effective Comp. Ratio</td>
<td>~12-16</td>
</tr>
<tr>
<td>Swirl Ratio (stock)</td>
<td>0.7</td>
</tr>
<tr>
<td>Piston Bowl Geometry</td>
<td>Stock</td>
</tr>
</tbody>
</table>

**Injection systems:**
- Cat HEUI 315B,
- Bosch Gen 2 Common Rail
  - 1500 bar, 0.25 mm 6-hole
Gasoline experimental conditions

**Double injection**
- A50
  - EGR
- Low Load (A25)

**Single Injection**
- A50 with 40% EGR

### Baseline Operating Conditions

<table>
<thead>
<tr>
<th>FTP Cycle Point</th>
<th>A50</th>
<th>A25</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed [rpm]</td>
<td>1300</td>
<td>1300</td>
</tr>
<tr>
<td>IMEP net [bar]</td>
<td>11</td>
<td>6.5</td>
</tr>
<tr>
<td>Pilot/Main % Split</td>
<td>30/70</td>
<td>30/70</td>
</tr>
<tr>
<td>Pilot SOI [ATDC]</td>
<td>-137</td>
<td>-137</td>
</tr>
<tr>
<td>Injection Pressure [bar]</td>
<td>1500</td>
<td>1500</td>
</tr>
<tr>
<td>Intake Temp [°C]</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>Intake Pressure [kPa]</td>
<td>200</td>
<td>152</td>
</tr>
<tr>
<td>EGR [%]</td>
<td>0-45</td>
<td>0-30</td>
</tr>
</tbody>
</table>

**Effect of EGR - gasoline**

**A50 double injection EGR**

- Simultaneous PM vs. NOx tradeoff can be achieved with sufficient EGR

- Approach EPA HD 2010 NOx and PM emissions levels at 45% EGR

- Ignition Delay increases due to combination of EGR and low CN fuel

\[ EID = SOI - CA50 \]
Effect of EGR - gasoline

- Combustion duration decreases with EGR → gasoline HCCI
  (fixed CA50 requires earlier SOI)

- Pressure rise rates increase (still lower than typical HCCI)

- Net ISFC decreases:
  - combustion phasing optimized

- 50% Indicated Thermal Efficiency approached
Gasoline single injection - A50

- Equivalence ratio stratification controls ignition and heat release profile

<table>
<thead>
<tr>
<th>Injection Strategy</th>
<th>Double</th>
<th>Single</th>
</tr>
</thead>
<tbody>
<tr>
<td>EGR (%)</td>
<td>40.8</td>
<td>41</td>
</tr>
<tr>
<td>NOx (g/kWh)</td>
<td>0.41</td>
<td>0.37</td>
</tr>
<tr>
<td>HC (g/kWh)</td>
<td>2.68</td>
<td>1.39</td>
</tr>
<tr>
<td>PM (g/kWh)</td>
<td>0.021</td>
<td>0.026</td>
</tr>
<tr>
<td>CO (g/kWh)</td>
<td>6.76</td>
<td>5.53</td>
</tr>
<tr>
<td>ISFC net (g/kWh)</td>
<td>173.5</td>
<td>167.9</td>
</tr>
<tr>
<td>IMEP net (bar)</td>
<td>11.23</td>
<td>11.62</td>
</tr>
<tr>
<td>Max PRR (bar/deg)</td>
<td>12.4</td>
<td>9.0</td>
</tr>
</tbody>
</table>

50% indicated thermal Efficiency
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  - CFD Modeling – Fuel reactivity

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- Conclusions
Comparison of diesel vs. gasoline ignition delay

- CHEMKIN – ERC PRF Mechanism
- Constant volume combustion with $T_{\text{init}} = 800-900$ K

- n-heptane (diesel) delay $\sim 6 \times$ shorter than iso-octane (gasoline)
- Diesel delay much less sensitive to pressure and equivalence ratio
- Gasoline fuel requires boosted operation and/or high intake temperature and locally rich but “mixed enough” (low swirl, low injection pressure)
Fuel reactivity control: Dual-fuel PCCI

• Bessonette (SAE 2007-01-0191) extended HCCI load range by varying fuel composition
  – 16 bar BMEP → required 27 cetane fuel: gasoline-like
  – 3 bar BMEP → required 45 cetane fuel: diesel-like

• Optimized operation requires different fuel reactivity for different operating conditions: Dual-fuel
  – Port fuel injection of gasoline
  – Direct injection of diesel fuel

← Fuel blending in-cylinder

- No DEF tank!
Modeling used for PRF & EGR selection

• SENKIN ERC-PRF simulations
  - 6, 9, and 11 bar IMEP
  - 1300 rev/min
  iso-octane → gasoline
  n-heptane → diesel

• As load is increased, minimum ISFC cannot be achieved with either neat diesel or neat gasoline

Kokjohn & Reitz – ICLASS-09
Charge preparation

KIVA GA optimization used to choose injection parameters*
- Gasoline port injection
- Diesel DI

- SOI1 ~ -60° ATDC
- SOI2 ~ -33° ATDC
- 60% of diesel fuel
  in first injection

*Ckokjohn et al.
SAE 09FFL-0107
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Experiments: Dual-fuel PCCI - 11 bar

**IMEP (bar)** | 11  
**Speed (rpm)** | 1300  
**EGR (%)** | 45.5  
**Equivalence ratio (-)** | 0.77  
**Intake Temp. (° C)** | 32  
**Intake pressure (bar)** | 2.0  
**Gasoline (% mass)** | 78 82 85  
**Diesel inject pressure (bar)** | 800  
**SOI1 (° ATDC)** | -67  
**SOI2 (° ATDC)** | -33  
**Fract. of diesel in 1st pulse** | 0.65  
**IVC (°ATDC)** | -85  

- Fuel reactivity controls ignition and heat release rate  
- Combustion phasing easily controlled
**Effect of Comp. Ratio: Dual-fuel PCCI**

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<td>Speed (rpm)</td>
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<tr>
<td>EGR (%)</td>
<td>43</td>
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<tr>
<td>Equivalence ratio (-)</td>
<td>0.5</td>
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<tr>
<td>Intake Temp. (° C)</td>
<td>32</td>
</tr>
<tr>
<td>Intake pressure (bar)</td>
<td>2 / 1.74</td>
</tr>
<tr>
<td>Gasoline (% mass)</td>
<td>78 / 82</td>
</tr>
<tr>
<td>Diesel inject pressure (bar)</td>
<td>800</td>
</tr>
<tr>
<td>SOI1 (° ATDC)</td>
<td>-58</td>
</tr>
<tr>
<td>SOI2 (° ATDC)</td>
<td>-37</td>
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<td>Fract. of diesel in 1st pulse</td>
<td>0.62</td>
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<tr>
<td>IVC (°ATDC)</td>
<td>-85 / -143</td>
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- Stock CR (IVC 143) requires more gasoline to achieve similar combustion
  - PRR controlled with gasoline fraction
Effect of Comp. Ratio: Dual-fuel PCCI

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- PRR < 10 bar/deg and net ISFC of **158 g/kW-hr**!

- NOx and soot similar for both cams → well below US 2010

![Graph showing the effect of gasoline percentage on NOx and soot emissions]
Dual-fuel PCCI – Thermal efficiency

9–11 bar IMEP

Conv. Diesel: Staples SAE 2009-01-1124; LTC: Hardy 2006-01-0026, 2006; Gas PPCI: Hanson SAE 2009-01-1442, 2009; D-F PCCI: Kokjohn SAE 09FFL-0107
Simulation results

Extended combustion duration even as load is increased
Uncharacteristic of PCI combustion
Small diesel quantity provides improved control compared to gasoline HCCI

Crank = -69.9 °ATDC
Conclusions

• PPC “Mixed enough” Gasoline
  - No traditional PM/NOx tradeoff
  - Approach 2010 EPA HD on-highway truck emission standards in-cylinder at 11 and 6 bar net IMEP
  - Low ISFC and pressure rise rate

• Dual-fuel PCCI concept used to control fuel reactivity
  - Port fuel injection of gasoline (cost effective)
  - Direct injection of diesel fuel (moderate injection pressure)
  - Possibility of traditional diesel or SI (with spark plug) operation retained for full load operation

• Dual-fuel operation at 6, 9, and 11 bar net IMEP achieved with near zero NOx and soot and reasonable Pressure Rise Rate

• 53% indicated thermal efficiency achieved while easily meeting US 2010 EPA standards in-cylinder
Dual-fuel surpasses 50% Thermal Efficiency engines

Wartsila-Sulzer RTA96-C turbocharged two-stroke diesel is the most powerful and efficient prime-mover in the world. Bore 38”, 1820 L, 7780 HP/Cyl at 102 RPM

- If technology could be applied to all US Truck and Auto engines, oil consumption could be reduced by 1/3 = oil imports from Persian Gulf
Fuel Efficiency and US Oil Consumption

US Petroleum consumption: 20.7 Million Barrels of Oil per Day*
   65% used in transportation = 13.5 MBOD

Truck and Automotive fuel usage reduction by Dual Fuel:
   4.2 MBOD Diesel: 45% → 53%
       = improvement of 18% = 0.6 million barrels saved
   9.3 MBOD Gasoline SI: 30% → 53%
       = improvement of 77% = 4.1 million barrels saved

Total saved = 4.7 MBOD = 34% of US transportation oil
   (23% of total US petroleum used ~ $1 Billion saved / 2 days)

• Could reduce transportation oil consumption by 1/3
  = US imports from Persian Gulf
    - while surpassing 2010 emissions regulations

• US DOE/EERE FreedomCar & 21st Century Truck fuel efficiency goals:
  50% increase in light-duty, 25% increase in heavy-duty

* Energy Information Administration
   Official Energy Statistics from the U.S. Government
   http://www.eia.doe.gov/