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

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Diesel Emission Reduction Consortium

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

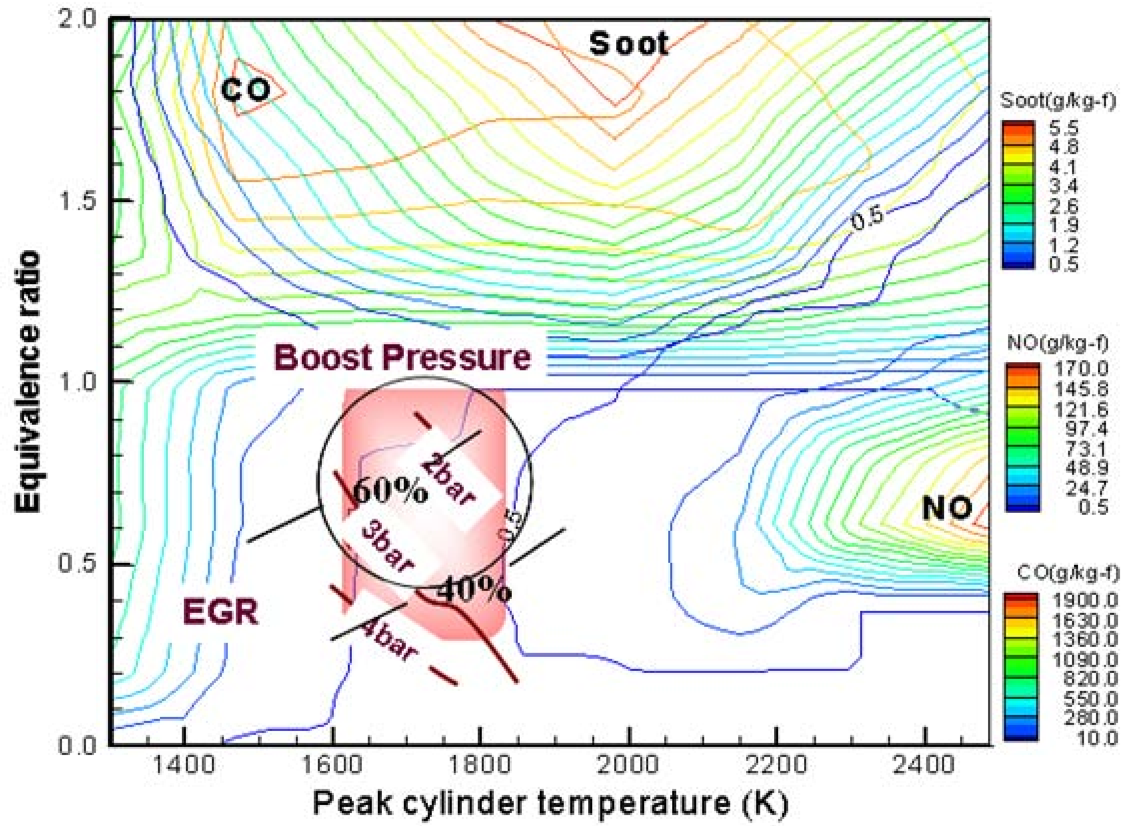
LTC (MK,PCCI,HCCI, etc.)

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

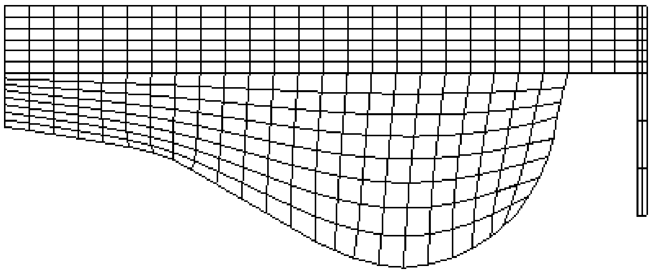
- Increase ignition delay to add mixing time

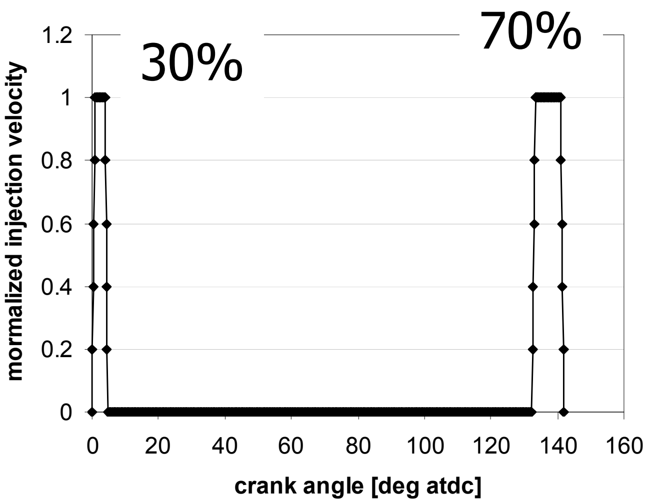
2 ways to achieve PPC

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





Injection profile

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

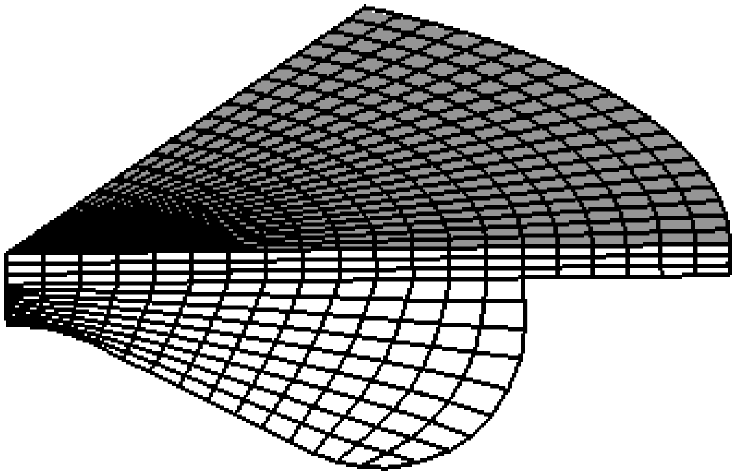
RNG k-ε turbulence model

KH-RT drop break up model

Grid-independent spray models

Drop collision and coalescence

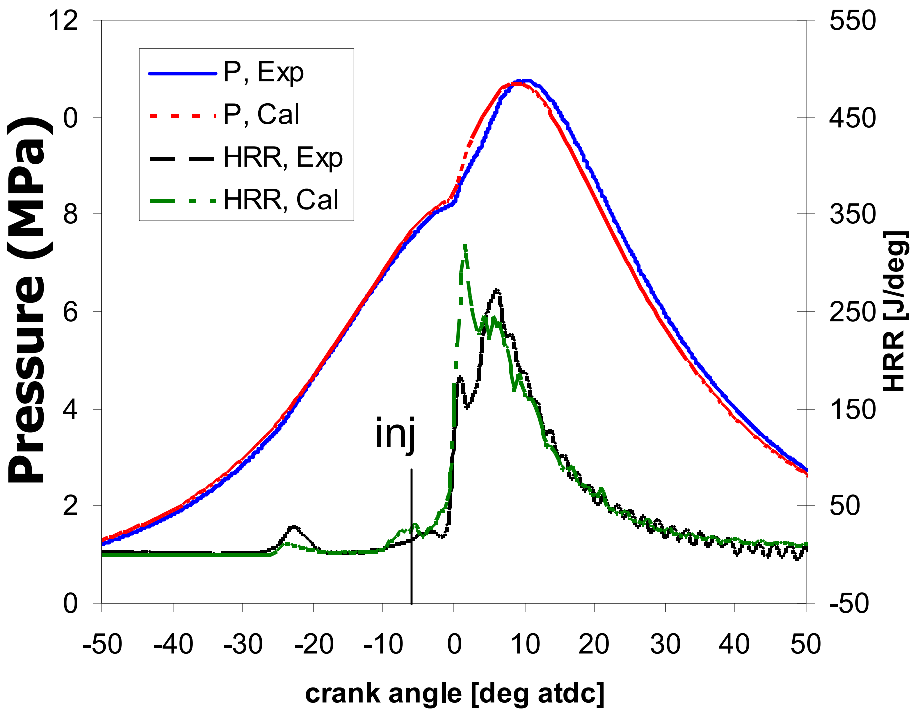
ERC reduced PRF mechanism



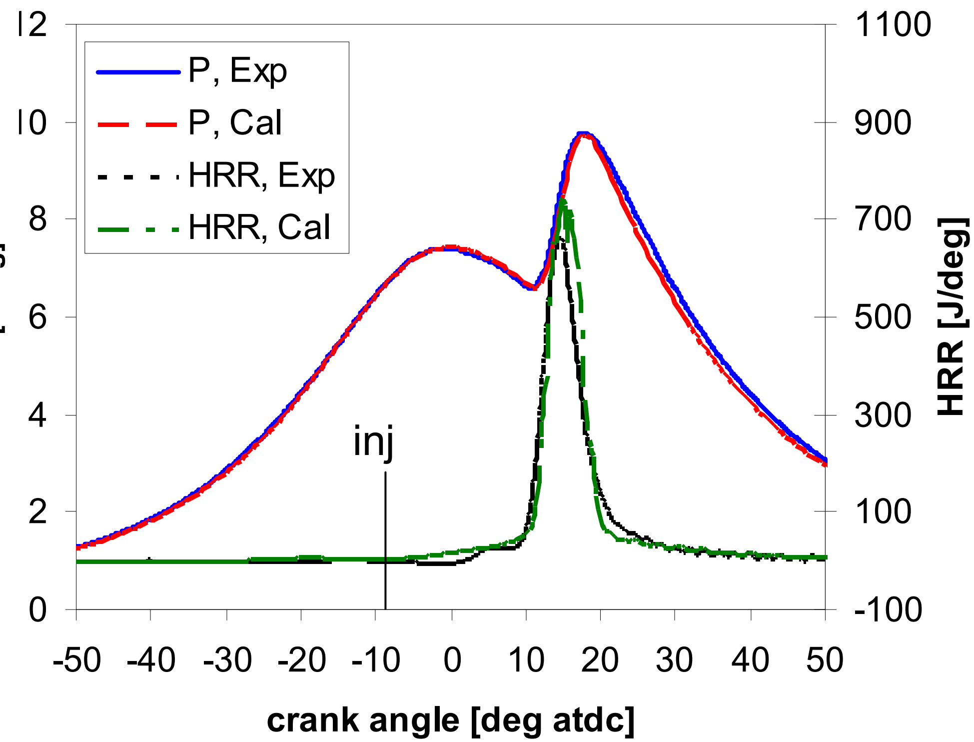
KIVA Modeling - Ra, Yun, Reitz
Int. J. Vehicle Design 2009

Diesel vs. gasoline - double injection

Diesel



Gasoline

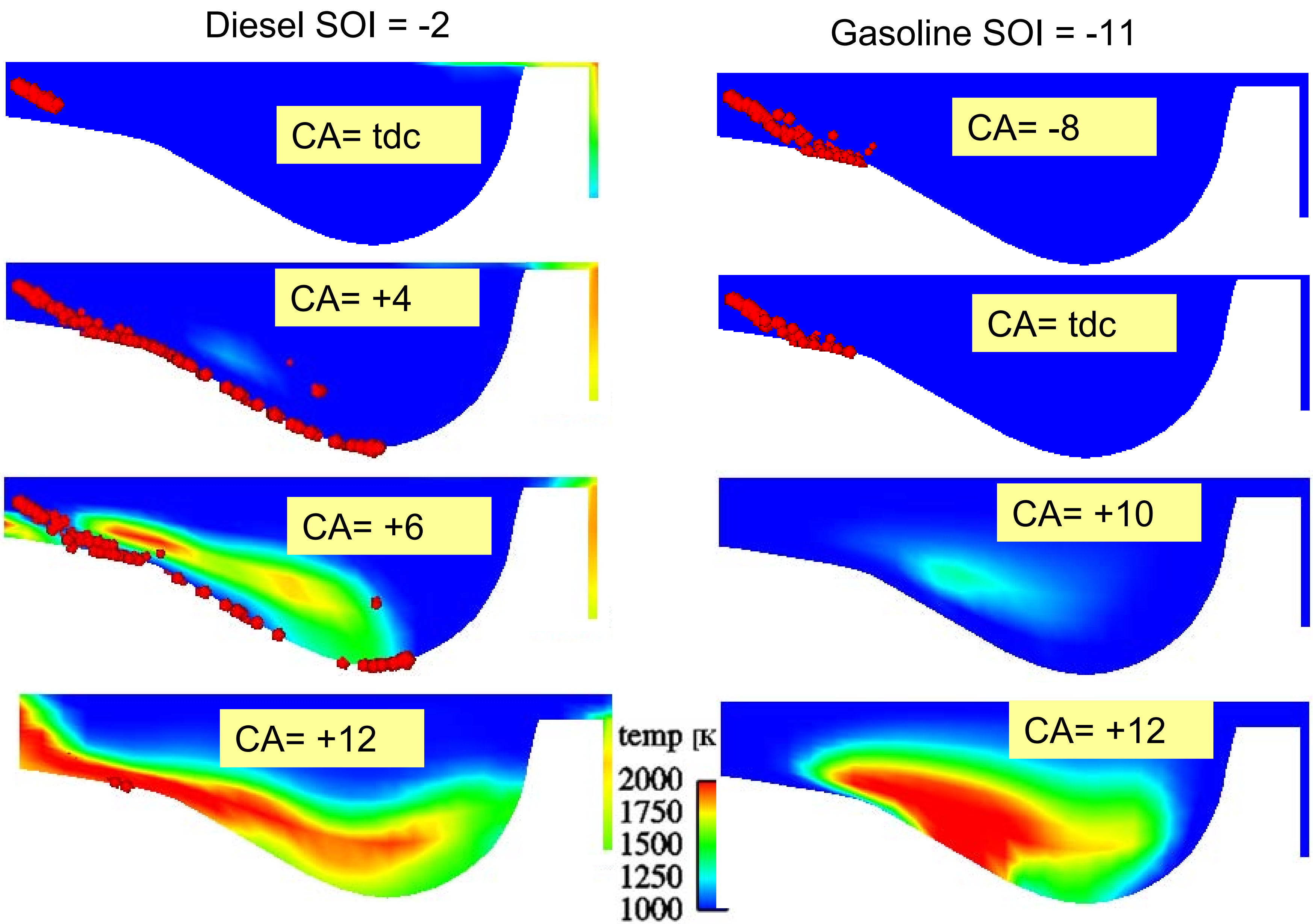


Start of injection: -137 and -6 (diesel), -9 (gasoline) deg atdc.

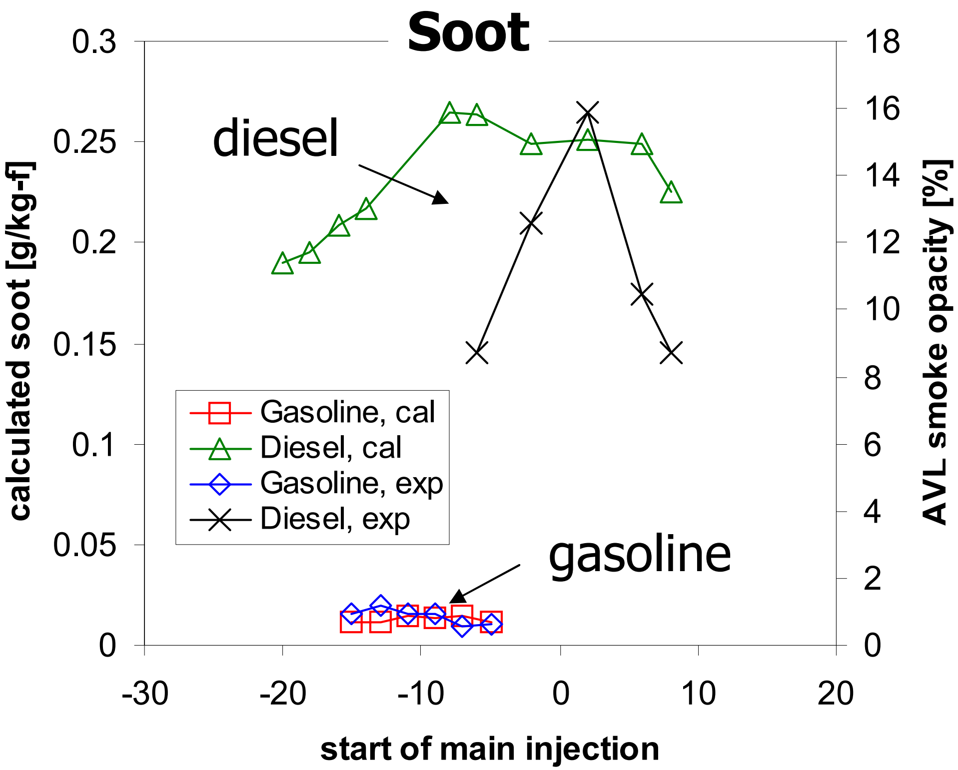
- Measured (Kalghatgi et al. SAE 2007)

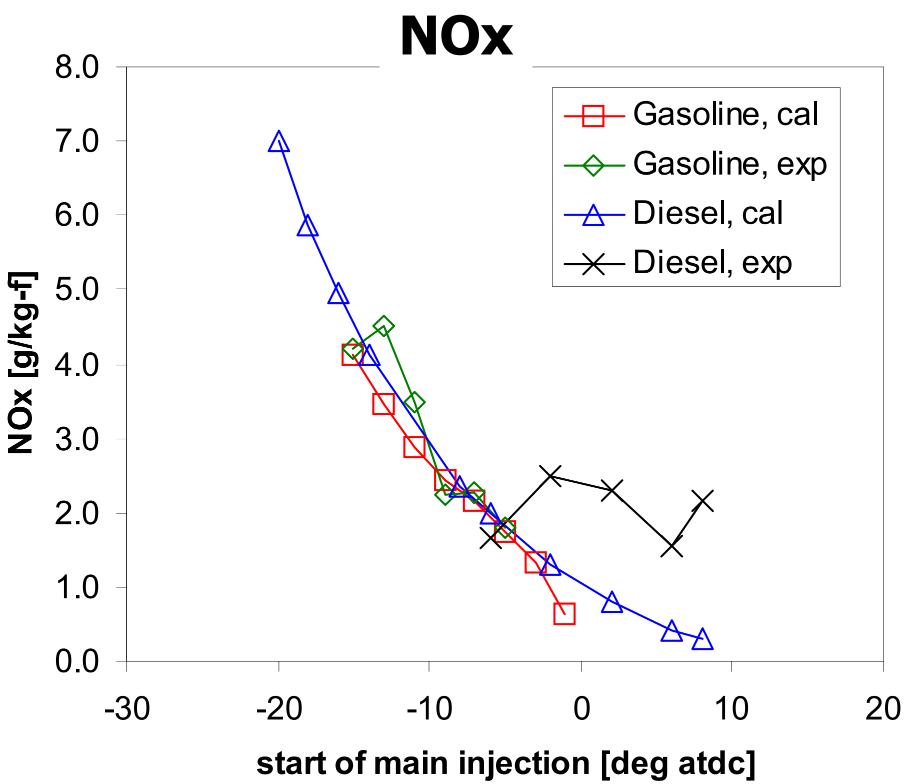
Model: ERC KIVA-CHEMKIN w/ PRF mechanism

Diesel vs. gasoline - ignition delay



Diesel vs. gasoline - emissions





Additional time for mixing of gasoline offers benefits for CIDI engines!

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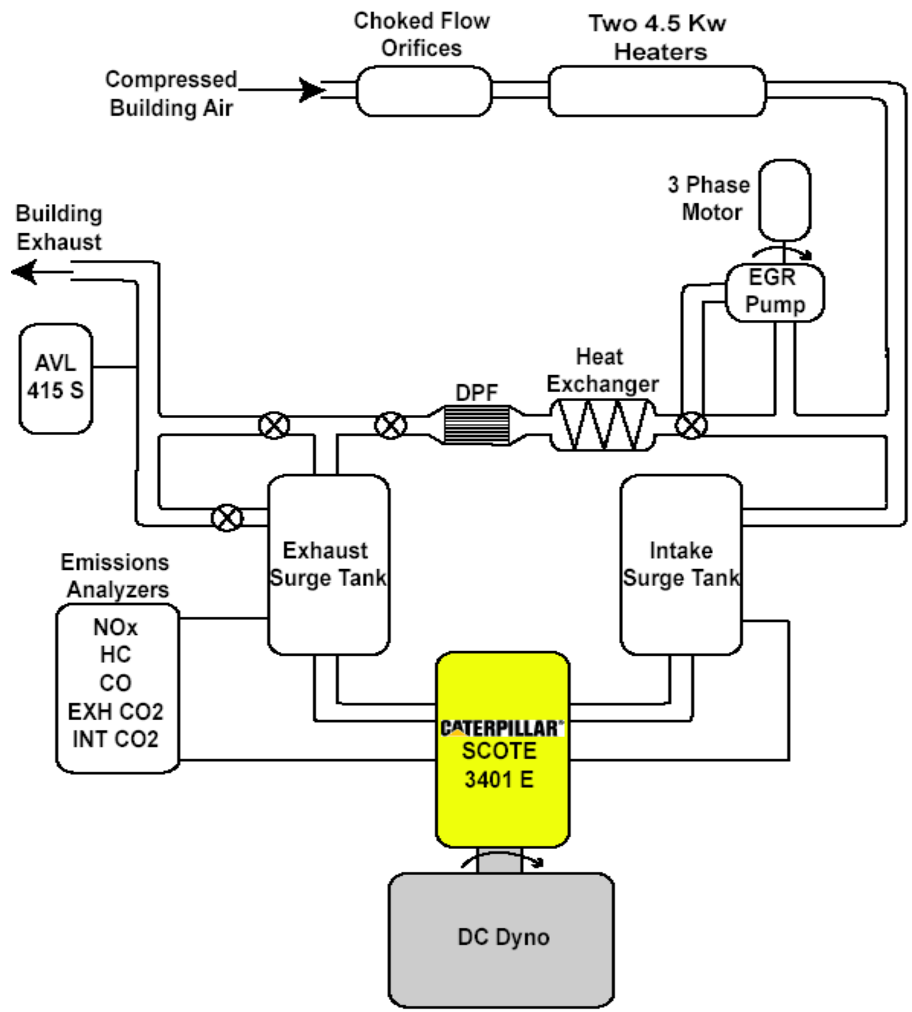
ERC Caterpillar engine lab

3401E SCOTE

Displacement (l)	2.44
Geometric Comp. Ratio	16:1
Bore (mm)	137
Stroke (mm)	165
Number of Valves	4
IVC (BTDC modified cam)	85/143
Effective Comp. Ratio	~12-16
Swirl Ratio (stock)	0.7
Piston Bowl Geometry	Stock

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

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

Hanson et al. "Operating a Heavy Duty DICI Engine with Gasoline for Low Emissions," SAE 2009-01-1442, 2009

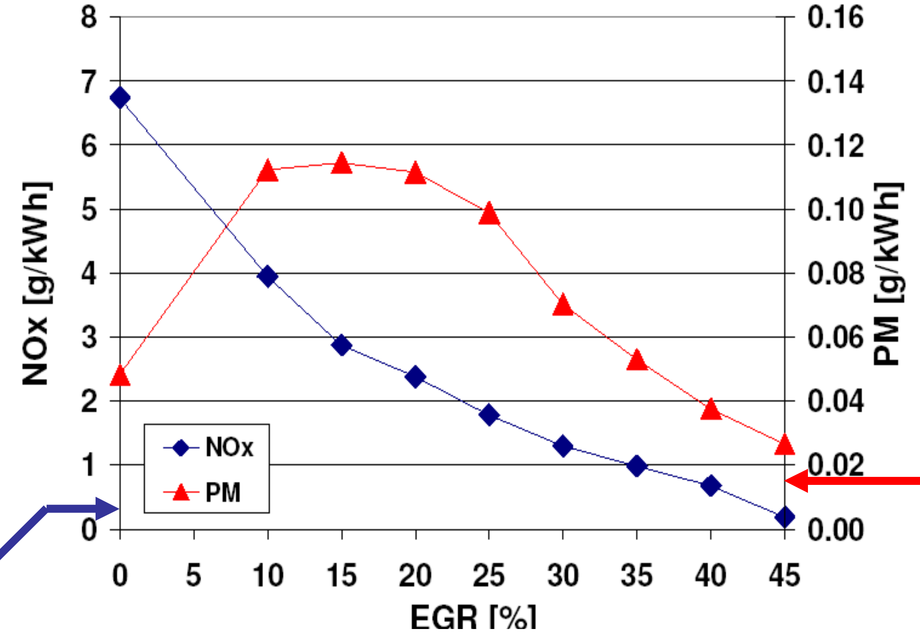
Effect of EGR - gasoline

A50 double injection EGR

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

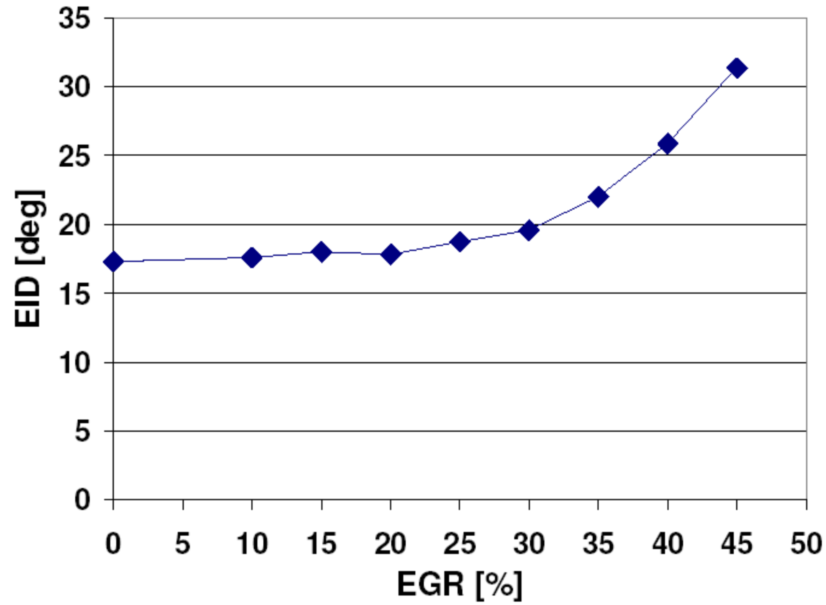
- Approach EPA HD 2010 NOx and PM emissions levels at 45% EGR

2010



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

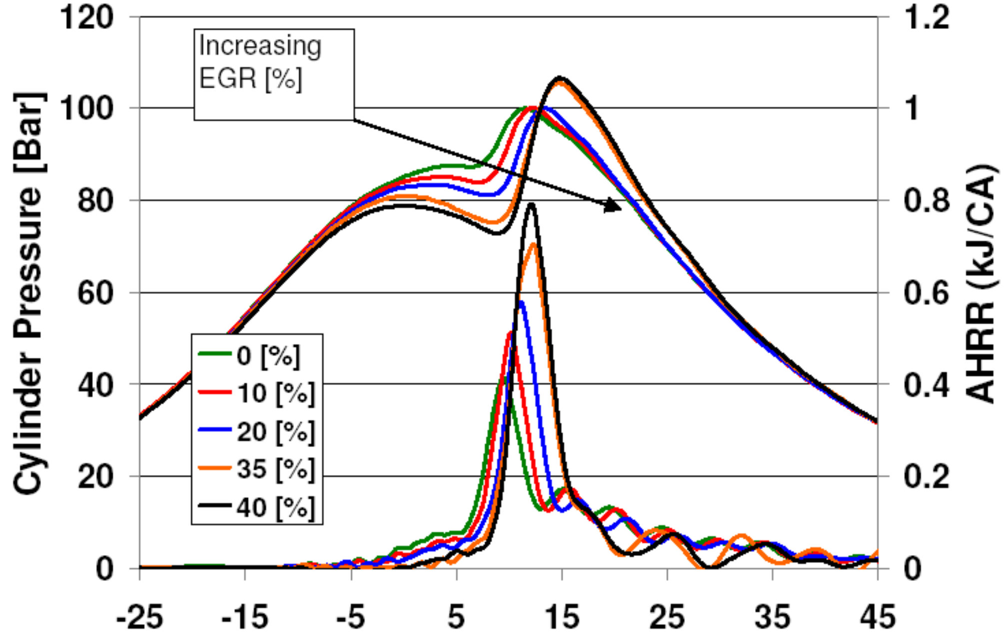
EID=SOI-CA50



Effect of EGR - gasoline

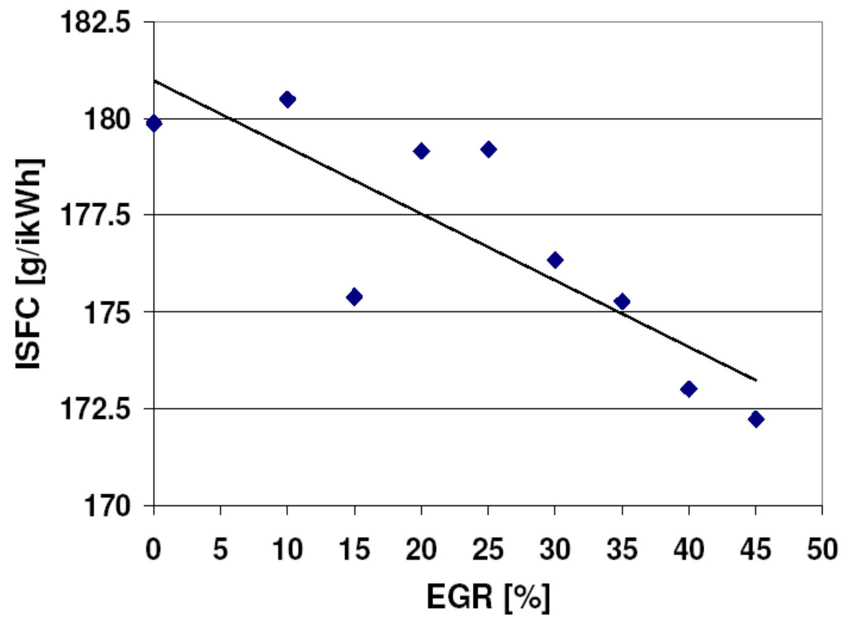
- Combustion duration decreases with EGR → gasoline HCCI

(fixed CA50 requires earlier SOI)



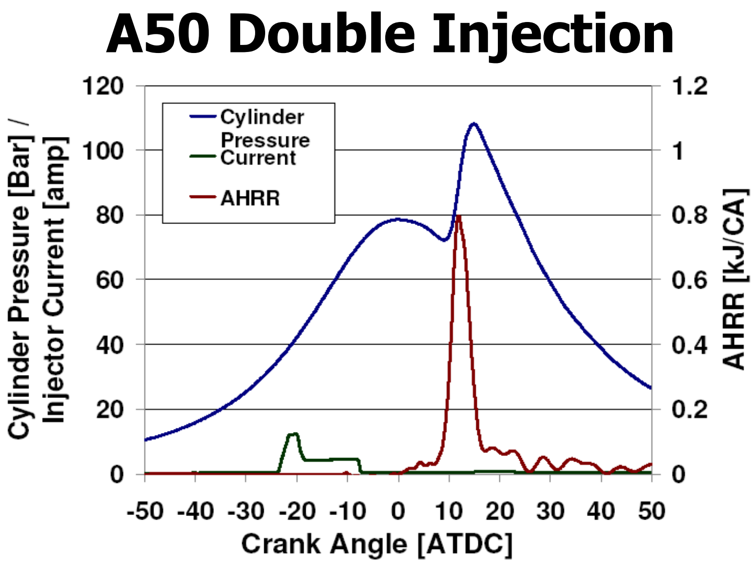
- Pressure rise rates increase (still lower than typical HCCI)

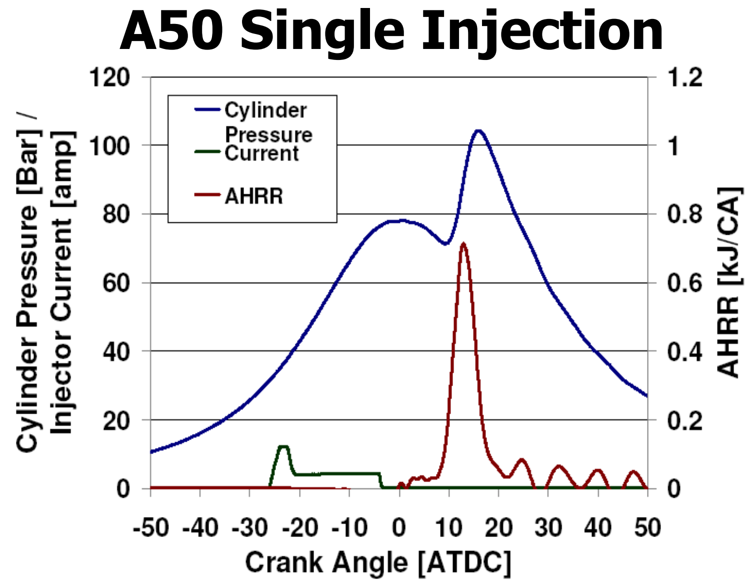
- Net ISFC decreases:
- combustion phasing optimized



- 50% Indicated Thermal Efficiency approached

Gasoline single injection - A50





- Equivalence ratio stratification controls ignition and heat release profile

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

← 50% indicated thermal Efficiency

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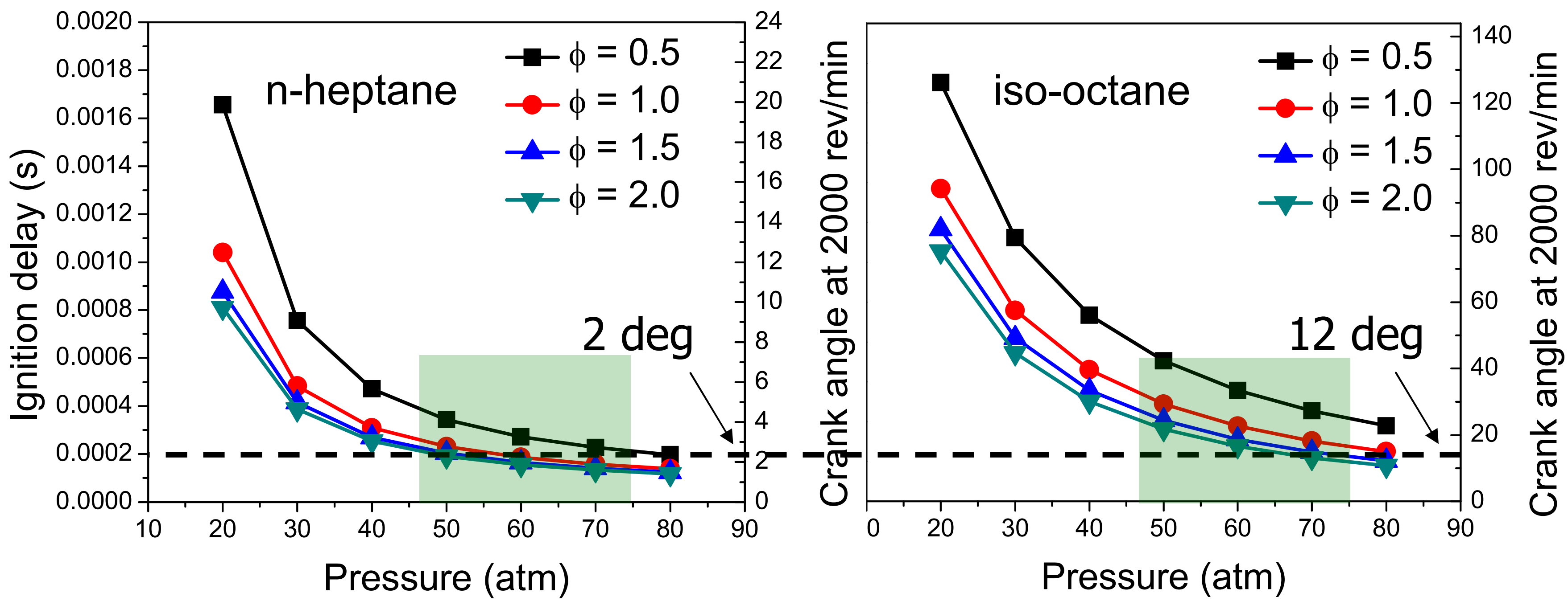
Comparison of diesel vs. gasoline ignition delay

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



- n-heptane (diesel) delay ~6 x 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

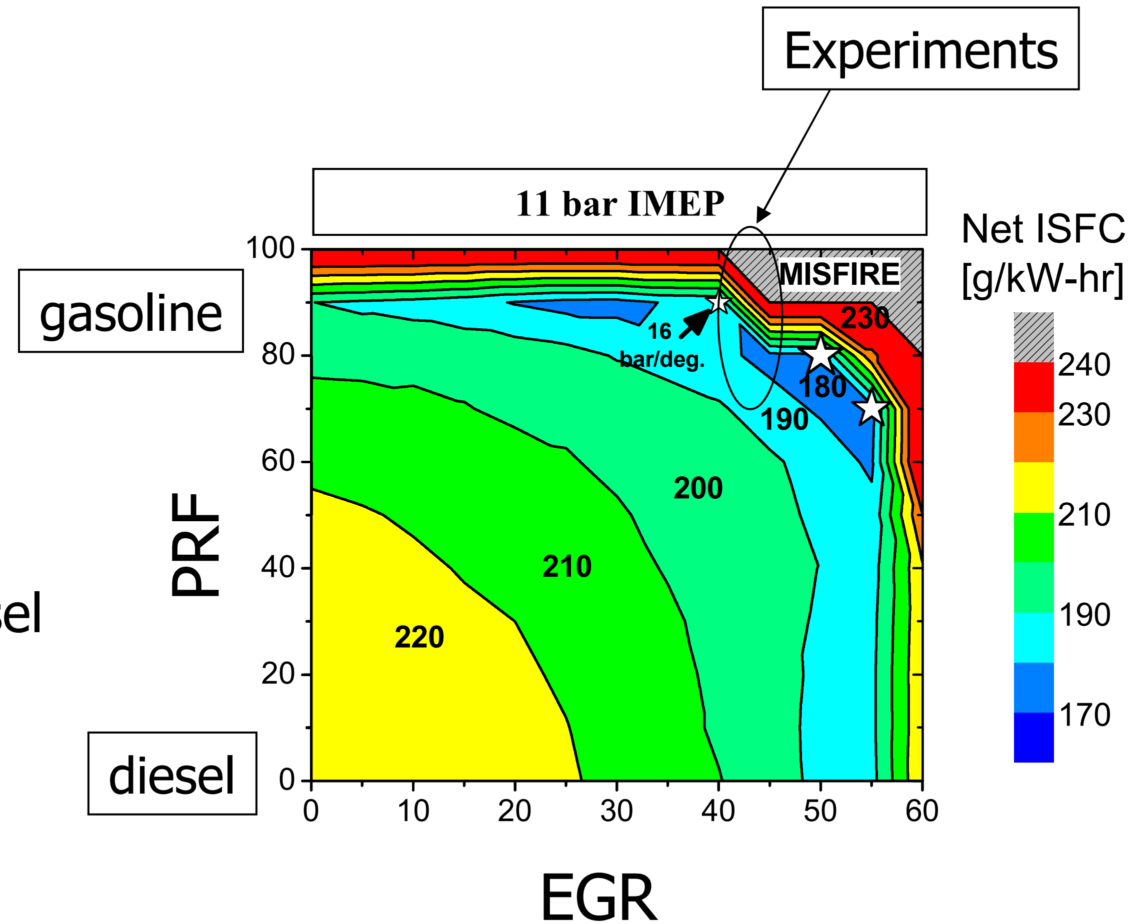
← Fuel blending in-cylinder

Diesel Exhaust "Fuel"

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

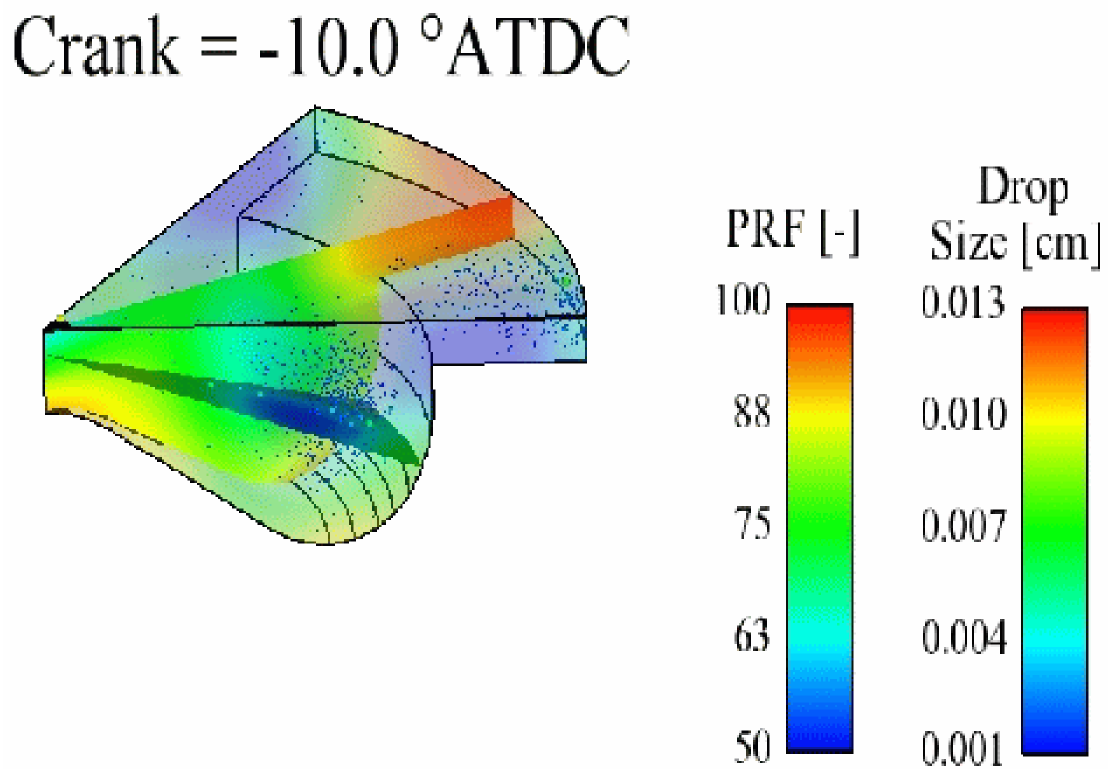
Kokjohn & Reitz – ICLASS-09

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



* Kokjohn et al. SAE 09FFL-0107

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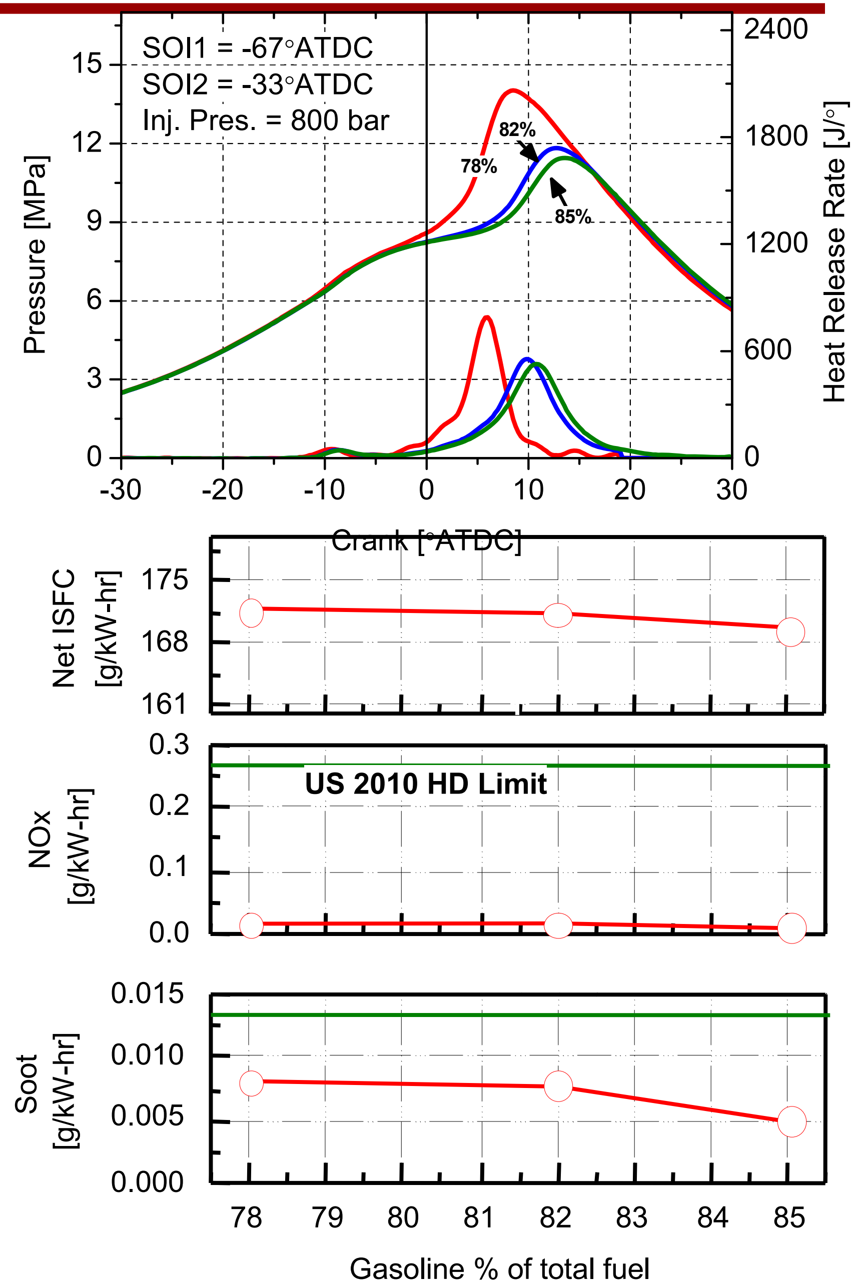
Experiments: Dual-fuel PCCI - 11 bar

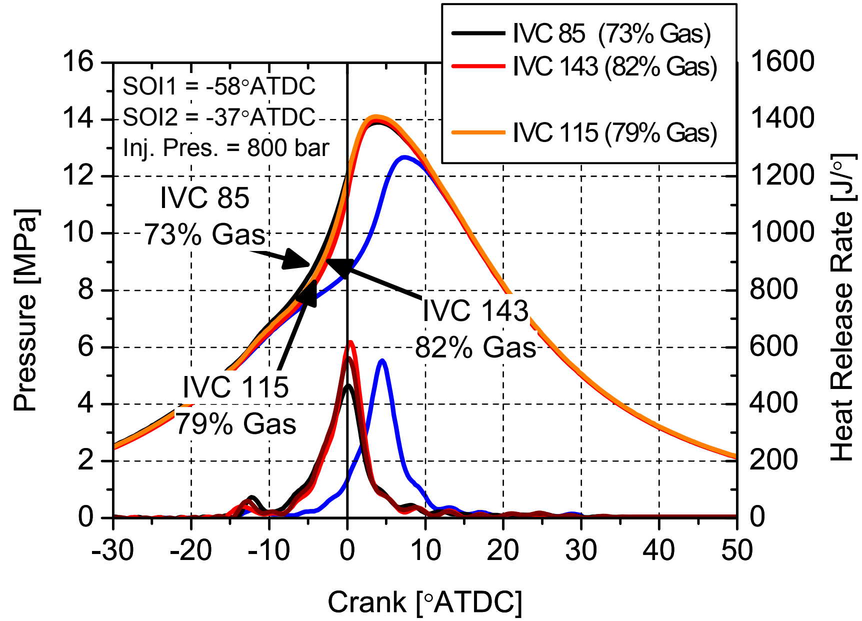
IMEP (bar)	11
Speed (rpm)	1300
EGR (%)	45.5
Equivalence ratio (-)	0.77
Intake Temp. (° C)	32
Intake pressure (bar)	2.0
Gasoline (% mass)	78 82 85
Diesel inject pressure (bar)	800
SOI1 (° ATDC)	-67
SOI2 (° ATDC)	-33
Fract. of diesel in 1st pulse	0.65
IVC (ºATDC)	- 85



- Fuel reactivity controls ignition and h
- Combustion phasing easily controlled

Effect of Comp. Ratio: Dual-fuel PCCI

IMEP (bar)	9	
Speed (rpm)	1300	
EGR (%)	43	
Equivalence ratio (-)	0.5	
Intake Temp. (° C)	32	
Intake pressure (bar)	2	1.74
Gasoline (% mass)	78	82
Diesel inject pressure (bar)	800	
SOI1 (° ATDC)	-58	
SOI2 (° ATDC)	-37	
Fract. of diesel in 1st pulse	0.62	
IVC (ºATDC)	-85	-143



- 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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Speed (rpm)	1300	
EGR (%)	43	
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Intake Temp. (° C)	32	
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SOI1 (° ATDC)	-58	
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Fract. of diesel in 1st pulse	0.62	
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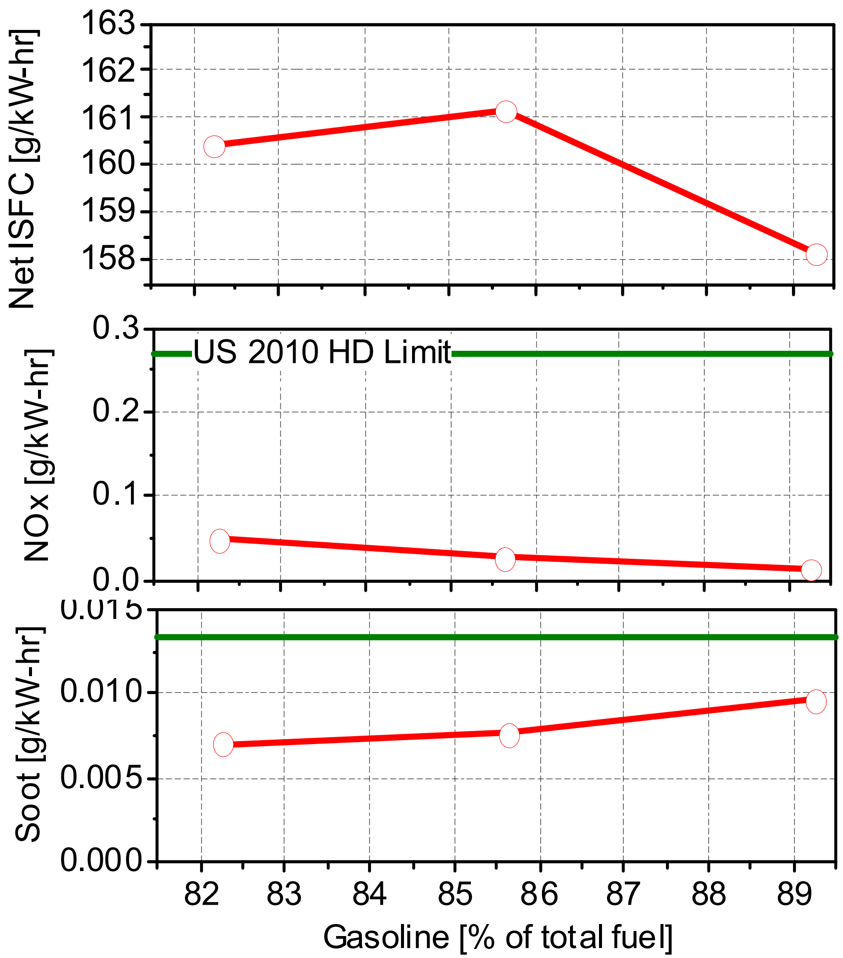
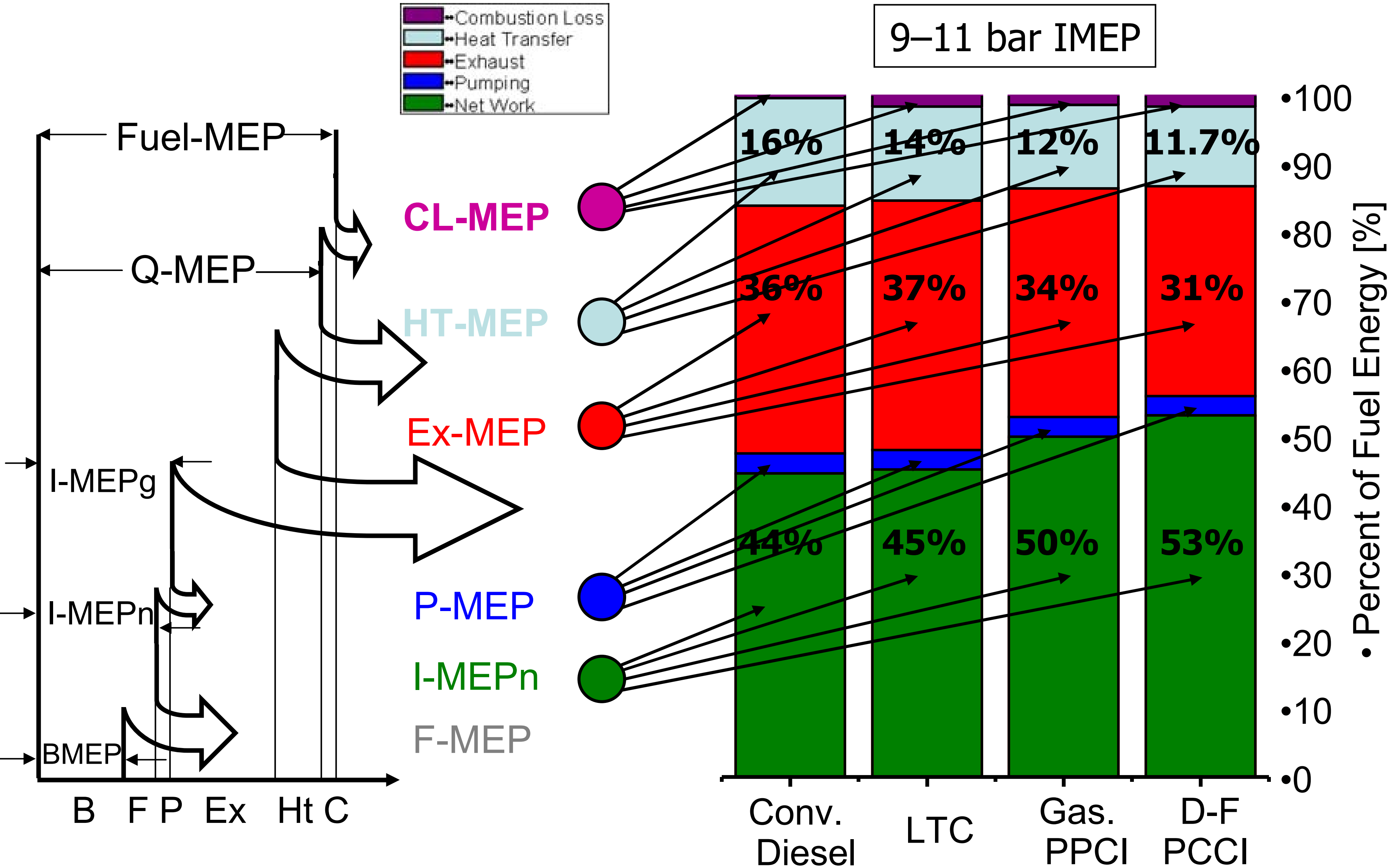
- PRR < 10 bar/deg and net ISFC of **158 g/kW-hr!**



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

Dual-fuel PCCI – Thermal efficiency



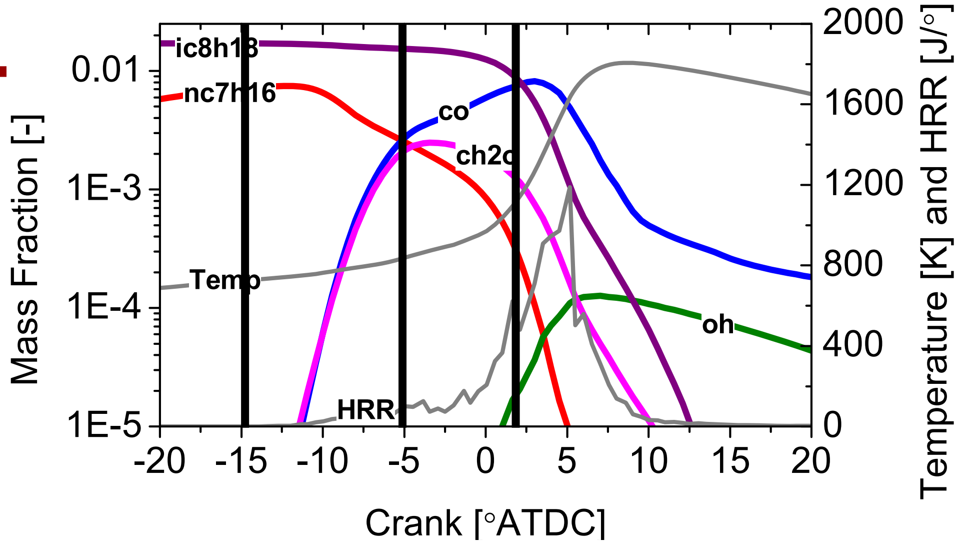
Conv. Diesel: Staples SAE 2009-01-1124; LTC: Hardy 2006-01-0026, 2006;
Gas PPCI: Hanson SAE 2009-01-1442, 2009; D-F PCCI: Kokjohn SAE 09FFL-0107

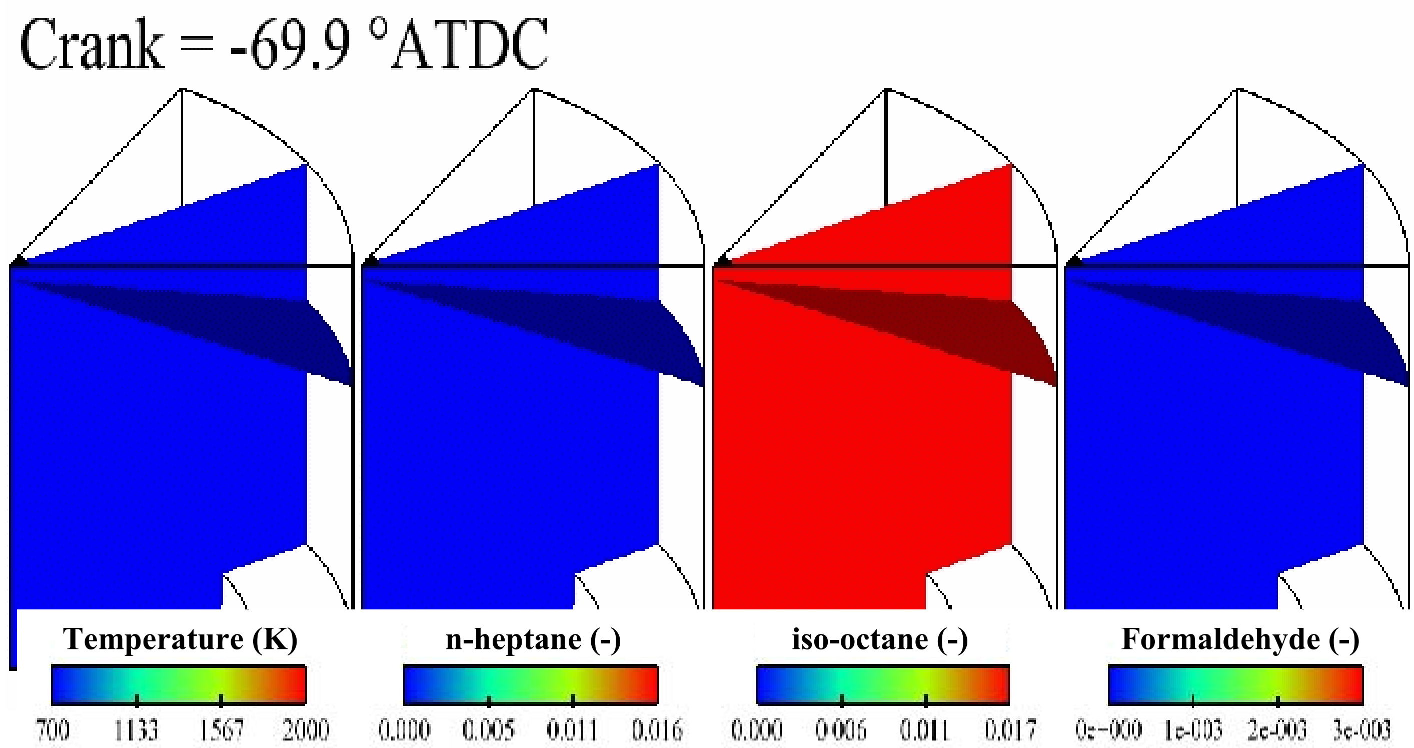
Simulation results

Extended combustion duration even as load is increased

Uncharacteristic of PCI combustion

Small diesel quantity provides improved control compared to gasoline HCCI





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 <u>while</u> 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

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