Giornata di Studio “Giorgio Minelli”
sui Motori a Combustione Interna

Bologna, 20 aprile 2018

Analisi CFD dei motori Diesel in modalità Dual-Fuel

M.Cristina Cameretti **Roberta De Robbio** **Raffaele Tuccillo**

Dipartimento di Ingegneria Industriale (D.I.I) Univesità di Napoli “Federico II

ATTIVITA' DI RICERCA – Università di Napoli FEDERICO II

- **CFD Dual Fuel Diesel Engine**
- **Diesel Spray Modeling**
- **Variable Geometry Turbocharging and e-Turbocharging**

Prof. M.Cristina Cameretti
Prof. Raffaele Tuccillo
Dr. Roberta De Robbio

Experimental activities on Turbochargers

Prof. Massimo Cardone
Prof. Marcello Manna
Dr. Rodolfo Bontempo

- **Engine Model**
- **Engine Calibration**
- **Hybrid Powertrain**
- **Vehicles Energetic Efficiency Evaluation**

Prof. Alfredo Gimelli
Dr. Muccillo M.
Dr. Giardiello G.
Dr. Pesce G.
Dr. Maccario A.
Dr. Pennacchia O.

- **1D Engine Simulation**
- **Vehicle modeling**
- **Turbocharging**

Prof. Fabio Bozza
Dr. Vincenzo de Bellis
Dr. Luigi Teodosio
Dr. Daniela Tufano
Dr. Enrica Malfi

- **Hydraulic Circuits: Lubrication, Cooling and Injection**
- **Piston Cooling Jets**
- **Fuel tank evaporation**

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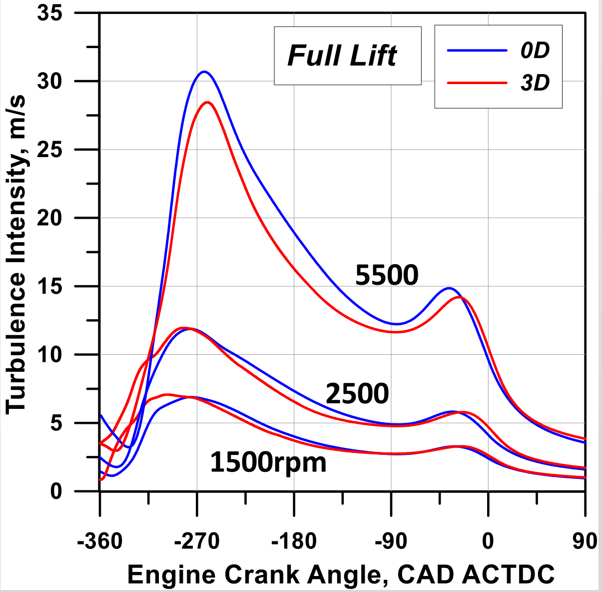
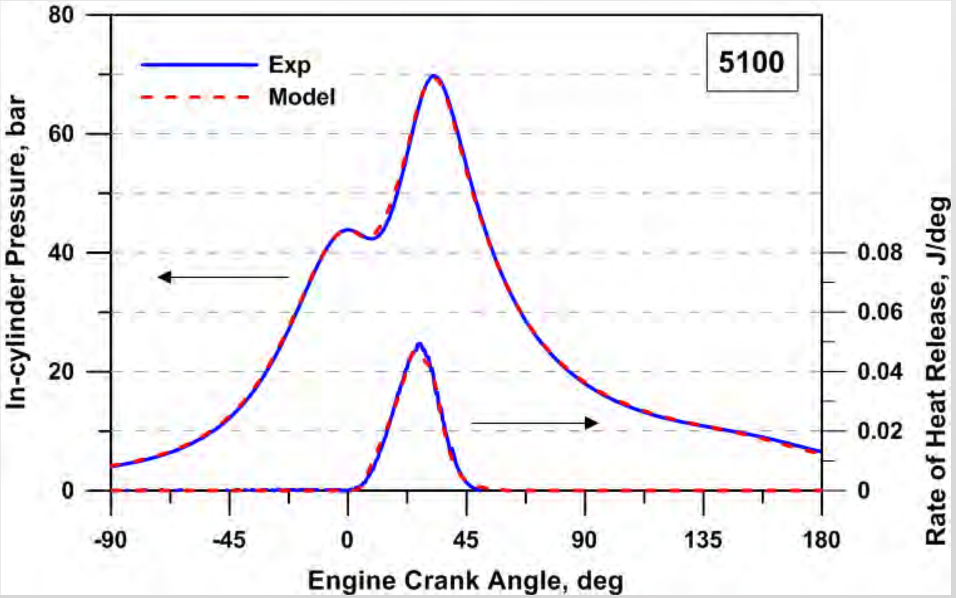
1D Engine Simulation

- Models of in-cylinder phenomena (turbulence, combustion, cycle-by-cycle variations, knock)
- Strategies for "virtual engine calibration"

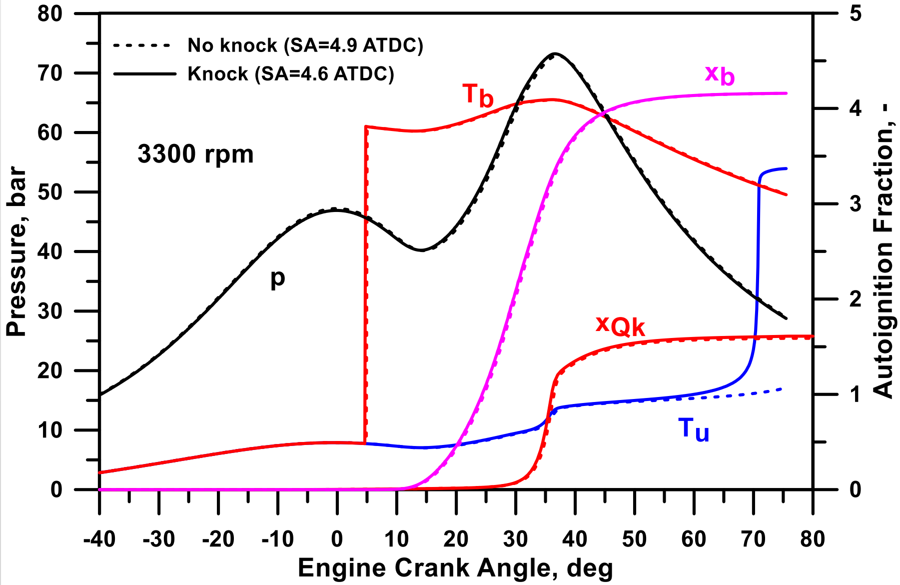
Turbulence



Combustion



Knock



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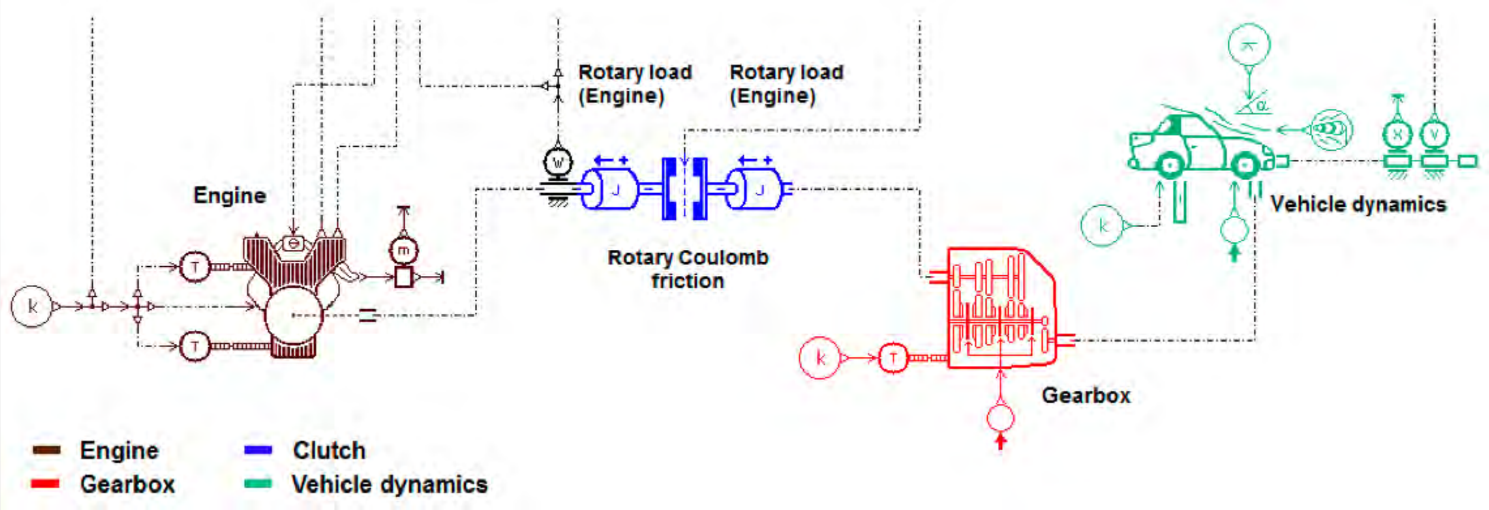
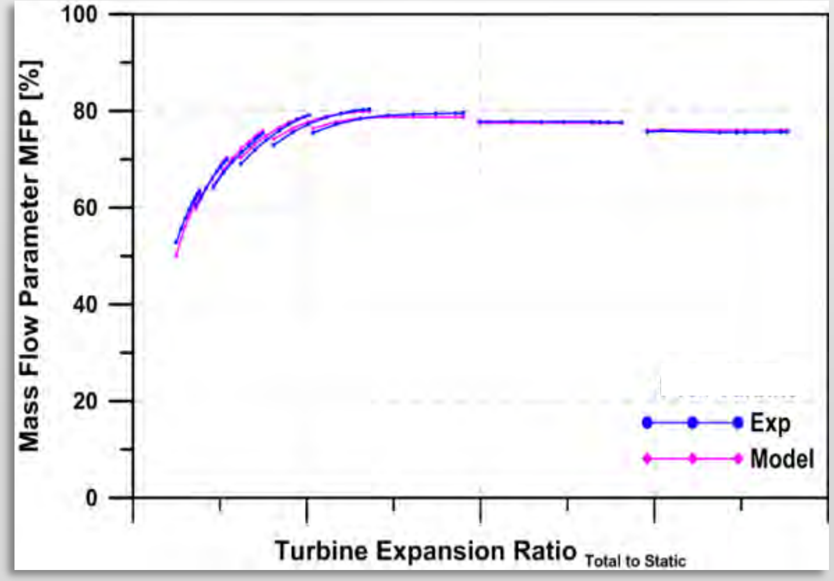


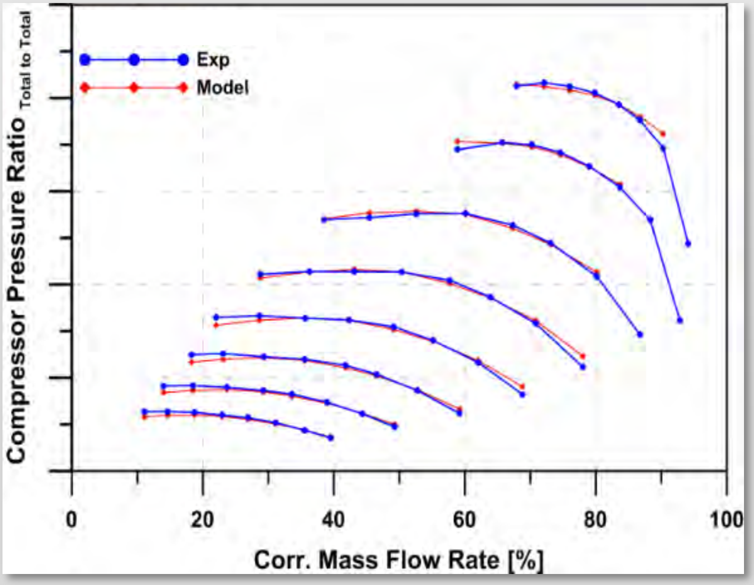
Vehicle modelling

- Computation of engine fuel maps
- Hybrid vehicle architectures
- Powertrain management

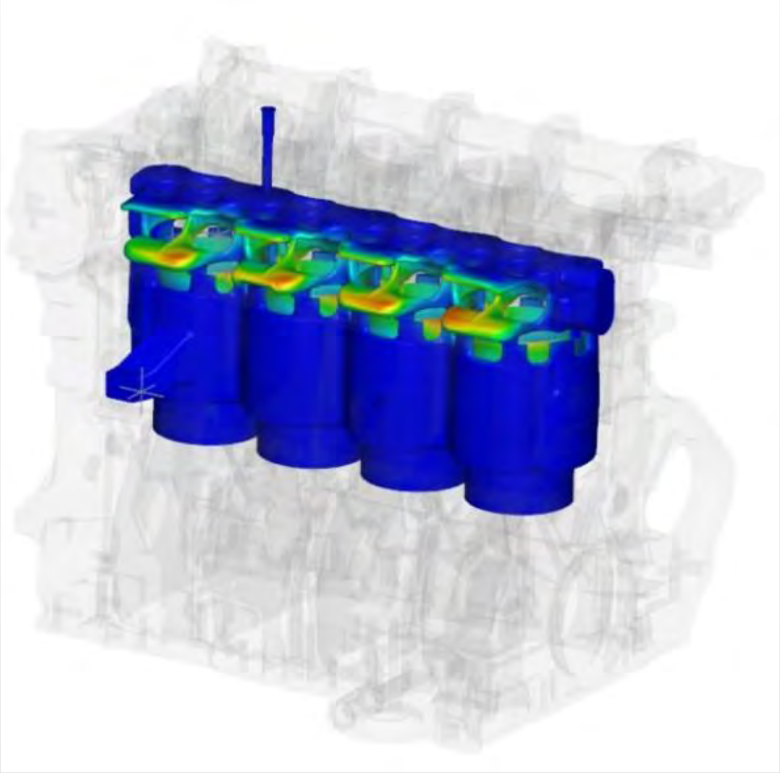
Turbocharging

- Compressor and turbine maps
- Advanced 1D modeling of compressor surge and WG turbine

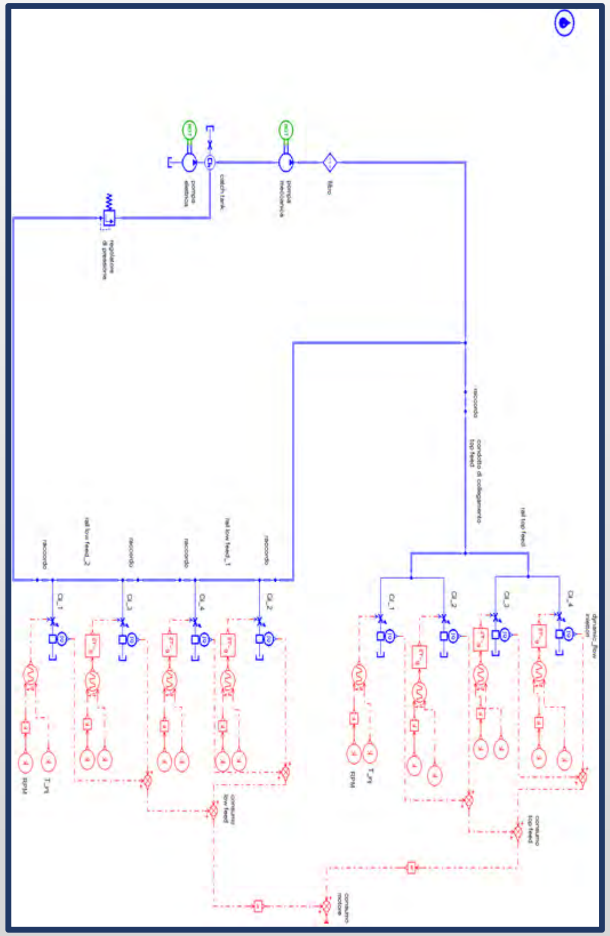




- Hydraulic Circuits: Lubrication, Cooling and Injection



- Piston Cooling Jets



- **Hydraulic Circuits: Lubrication, Cooling and Injection**
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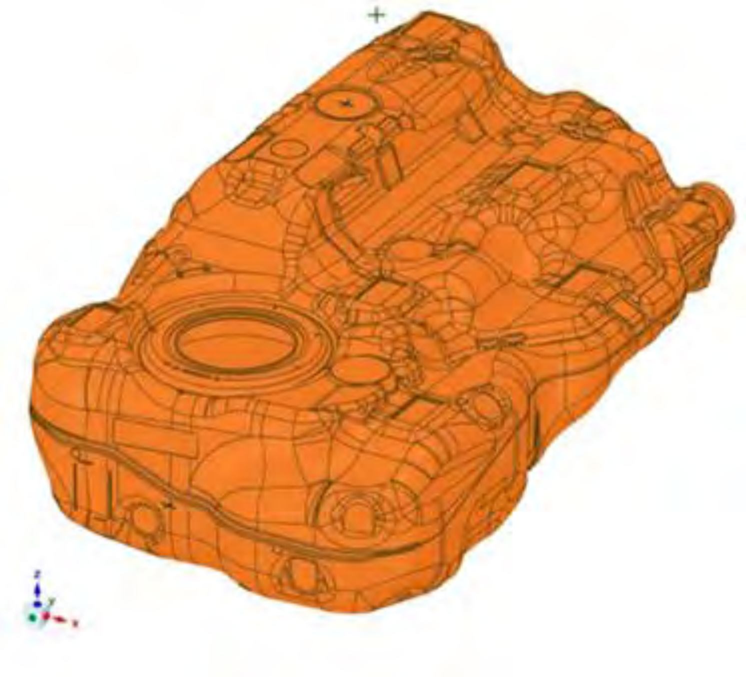
- Fuel tank evaporation



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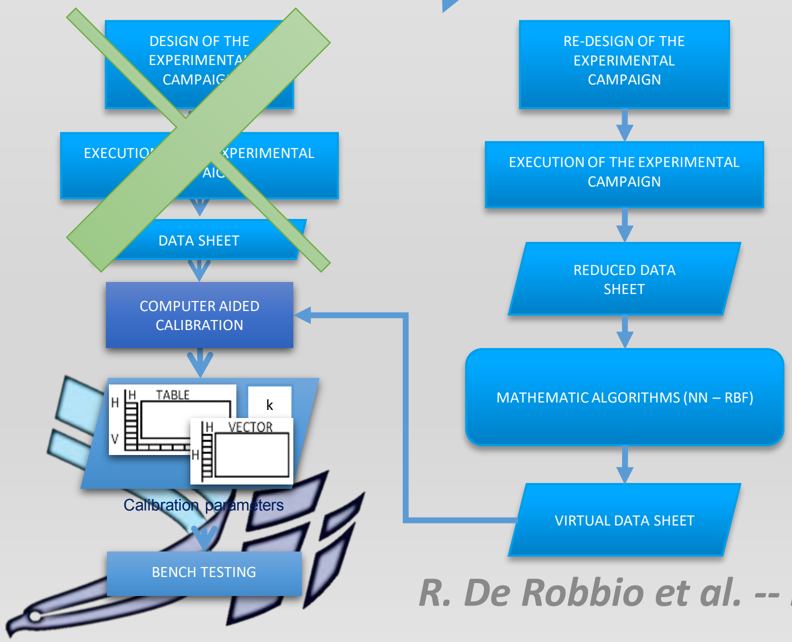
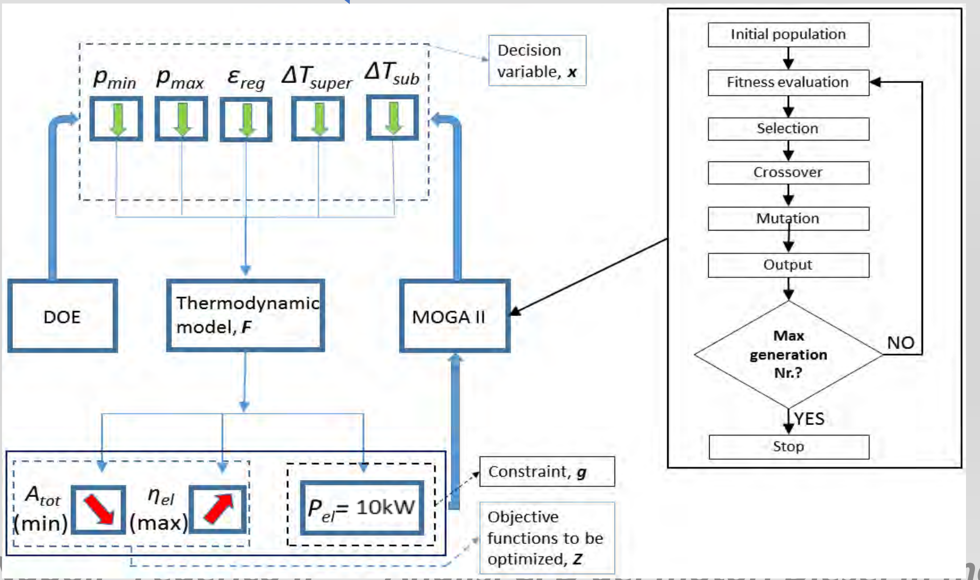
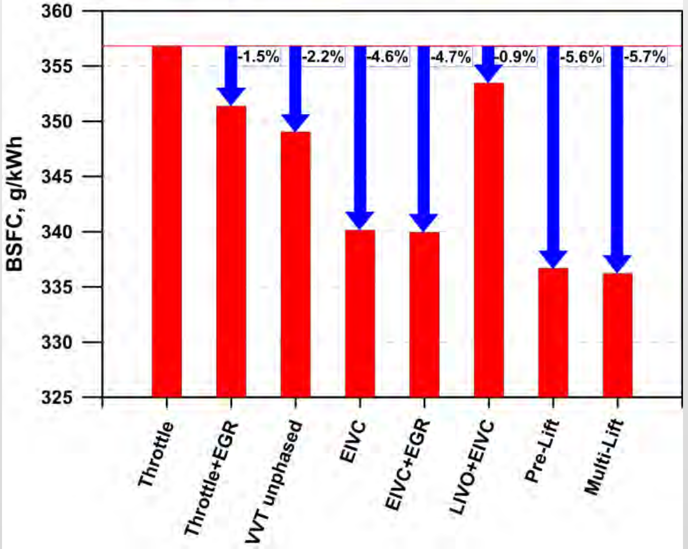
Dr. Maccario A.
Dr. Pennacchia O.

- Modelli motore e validazione automatica
- Calibrazione Automatica della Centralina Motore;
- Combustibili alternativi per Motori ad accensione comandata;
- HCCI;
- Powertrain ibride e valutazione efficienza energetica veicoli;
- Waste Heat Recovery Systems using Organic Rankine Cycle;

Attuali collaborazioni:
- ✓ *Teoresi SpA;*
- ✓ *FCA;*
- ✓ *Istituto Motori CNR*







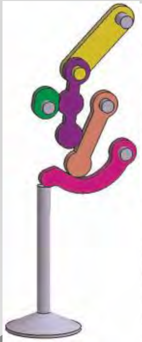


Table 8. Maximum increase in the dissipated power that is allowed in the studied VVA system to still obtain BSFC improvement.

Engine operating condition (r/min-bar of BMEP)	Maximum lift (mm)	Limit value of increase in power dissipated, ΔPL_{th} (kW)
5000 r/min-3 bar	3.8	0.523
5000 r/min-5 bar	3.8	0.590
5000 r/min-7 bar	5.1	0.522
5000 r/min-9 bar	5.1	0.638

BMEP: brake mean effective pressure.

Turbocharger Test Rig Layout

Experimental activities on Turbochargers

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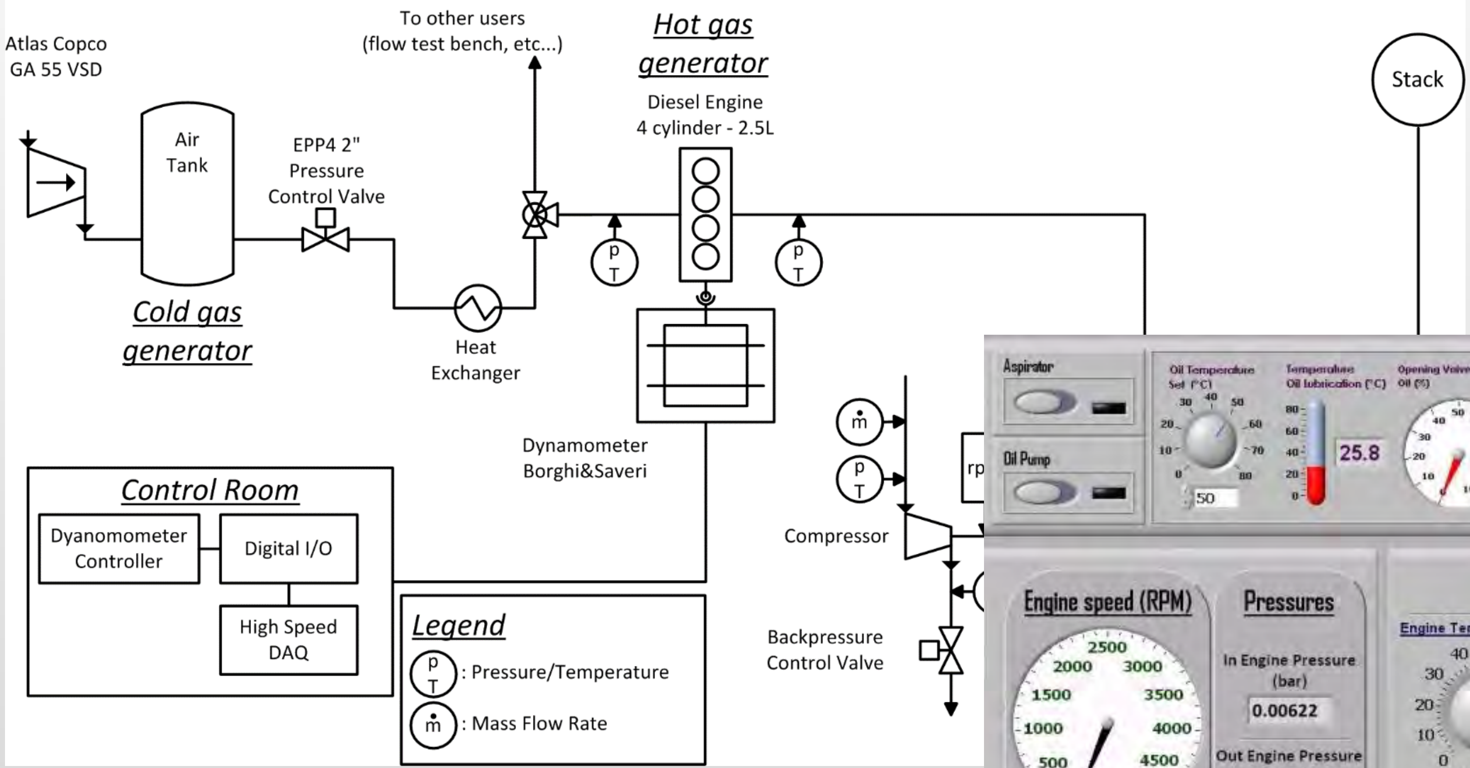
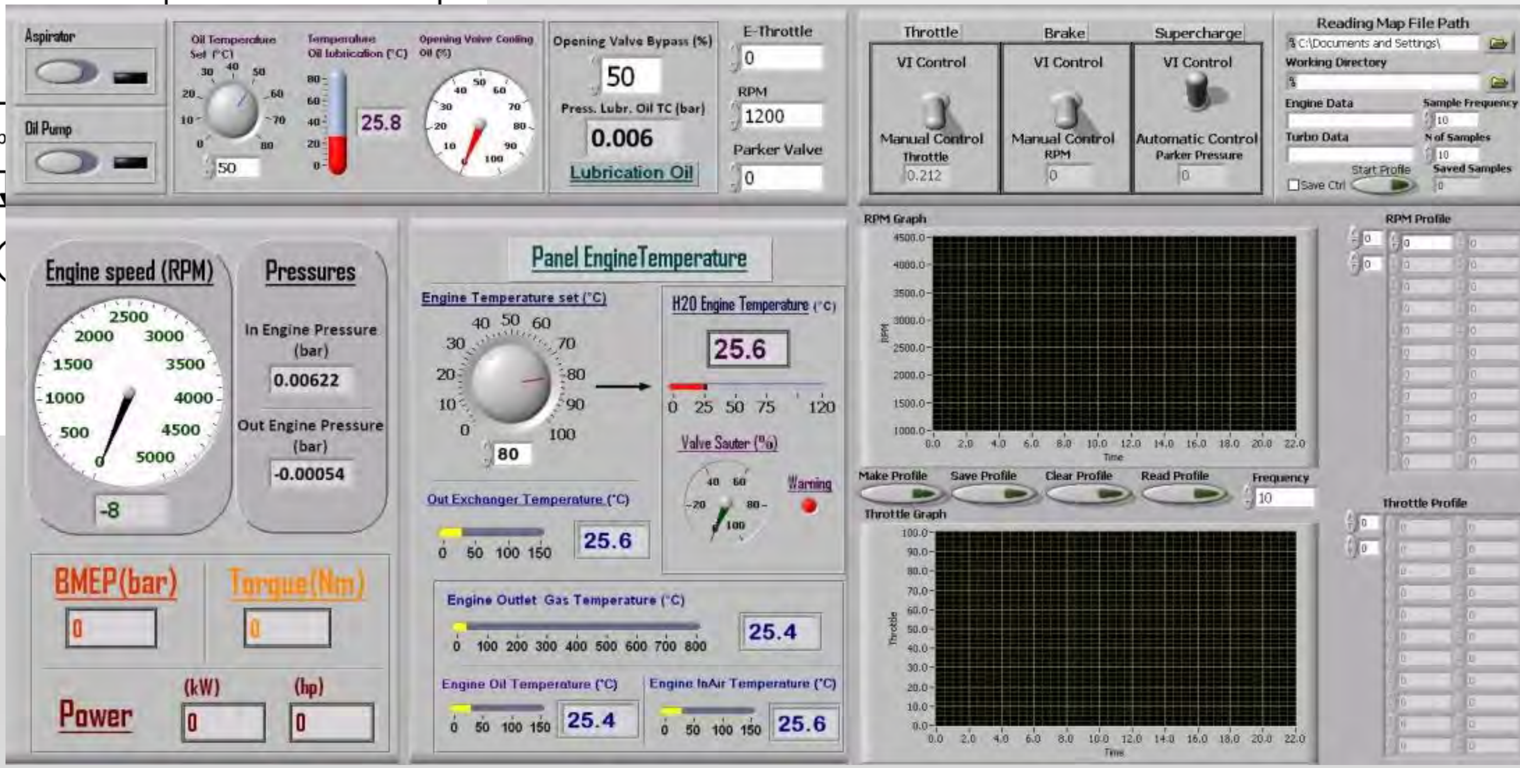
Prof. Marcello Manna
Dr. Rodolfo Bontempo



Virtual Instruments



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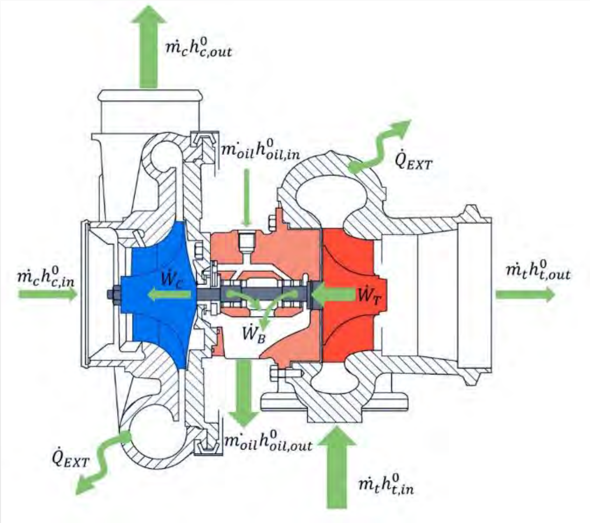
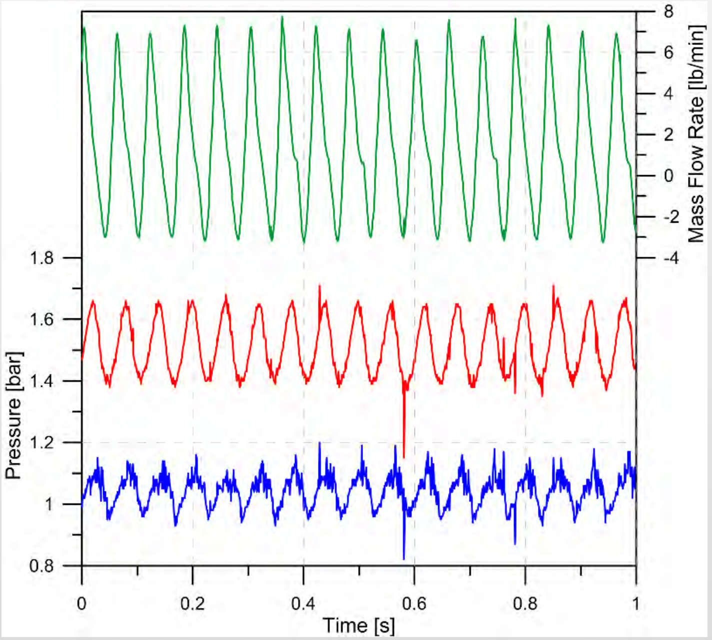
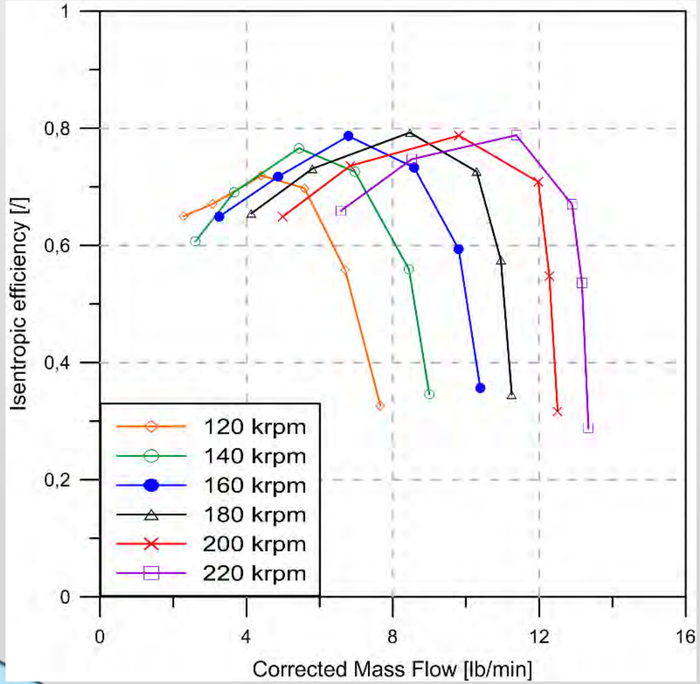
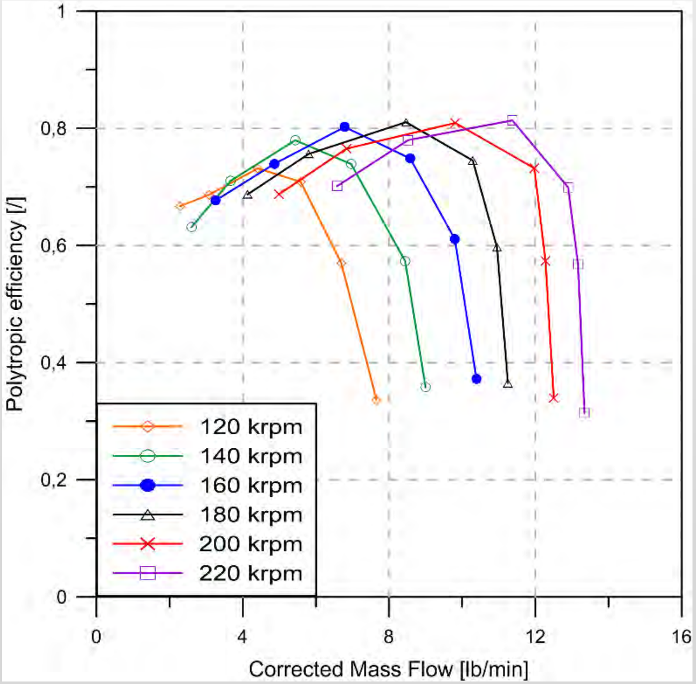


- Unsteady Experimental Results







- Experimental Steady-State Results

Dual Fuel Diesel Engine

Attuali collaborazioni:
- ✓ *Istituto Motori CNR*
- ✓ *Brunel University*

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Energetic and Environmental issues (like problems related to the air quality and to the use of fossil fuels)

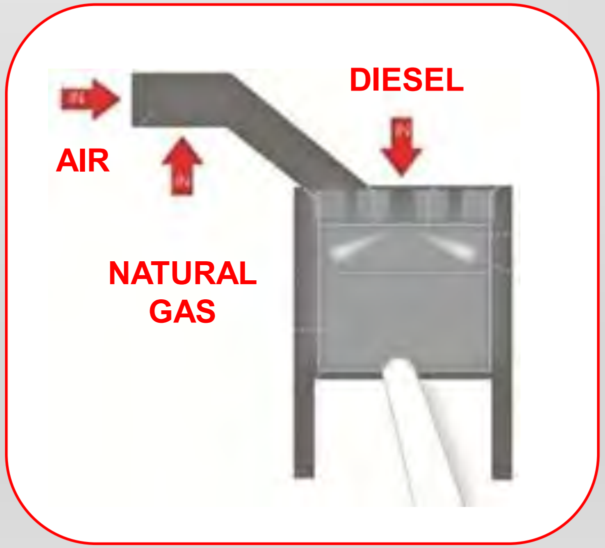
↓

Need of different energy sources in the field of internal combustion engines for vehicles.

↓

The dual-fuel represents a viable solution to reduce emissions from diesel engines by using natural gas as an alternative fuel

DUAL FUEL MODE



Advantages of NG
- ✓low NOx
- ✓low particulate matter
- ✓low CO_2

↕

Advantages of diesel:
- ✓high efficiency
- ✓high brake mean effective pressure

Background

[1] Abagnale C., Cameretti M.C., De Simio L., Gambino M., Iannaccone S., Tuccillo R. (2014). "Numerical Simulation And Experimental Test Of Dual Fuel Operated Diesel Engines". Applied Thermal Engineering, vol. 65, 2014, p. 403-417

[2] Abagnale C., Cameretti M.C., Ciaravola U., Tuccillo R., Iannaccone S., (2015). “Dual Fuel Diesel Engine at Variable Operating Conditions: A Numerical and Experimental Study”. SAE Technical paper 2015-24-2411. SAE TECHNICAL PAPER, vol. 1

[3] Cameretti M.C., Ciaravola U., Tuccillo R., Iannaccone S., L. DeSimio, (2016). “A numerical and experimental study of dual fuel diesel engine for different injection timings.” Applied Thermal Engineering Journal.

[4] Cameretti, M., De Robbio, R., and Tuccillo, R., "Performance Improvement and Emission Control of a Dual Fuel Operated Diesel Engine," SAE Technical Paper 2017-24-0066, 2017, doi:10.4271/2017-24-0066.

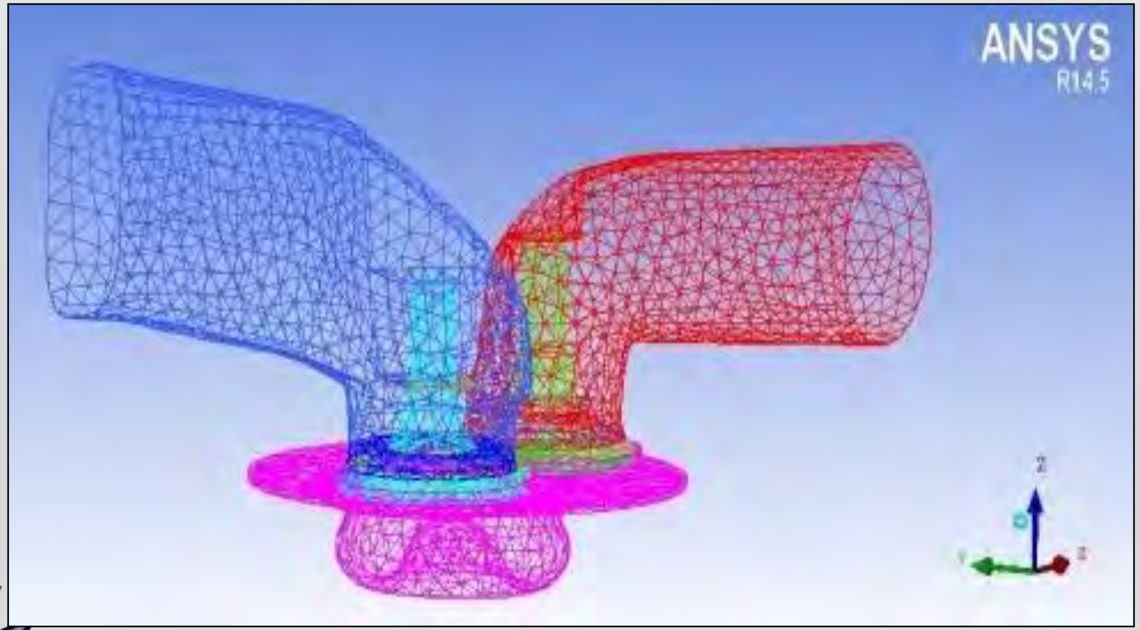
RP up to 80%

[5] Cameretti, M.C., De Robbio, R., Mancaruso, E., Sequino, L., Tuccillo, R., and Vaglieco, B., "CFD Analysis of the Combustion Process in Dual Fuel Diesel Engine," SAE Technical Paper 2018-01-0257, 2018

RP up to 22%

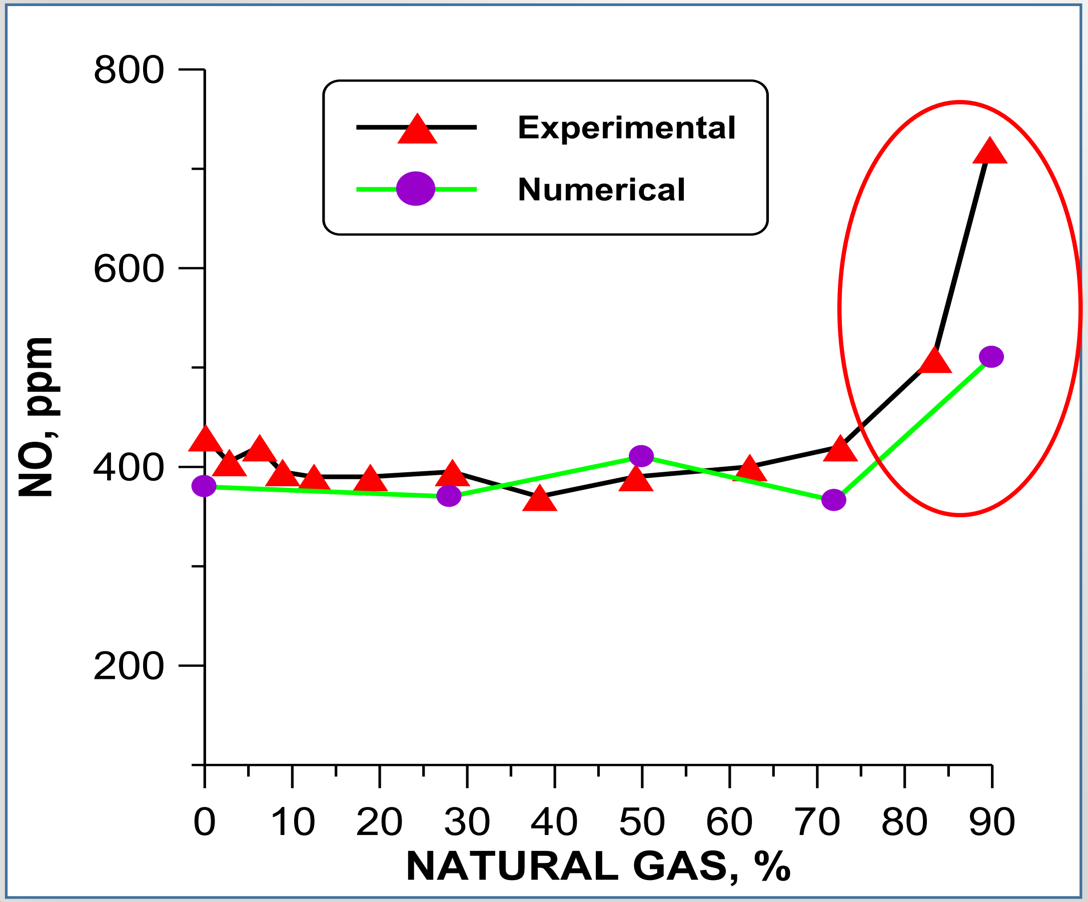
CFD Analisys

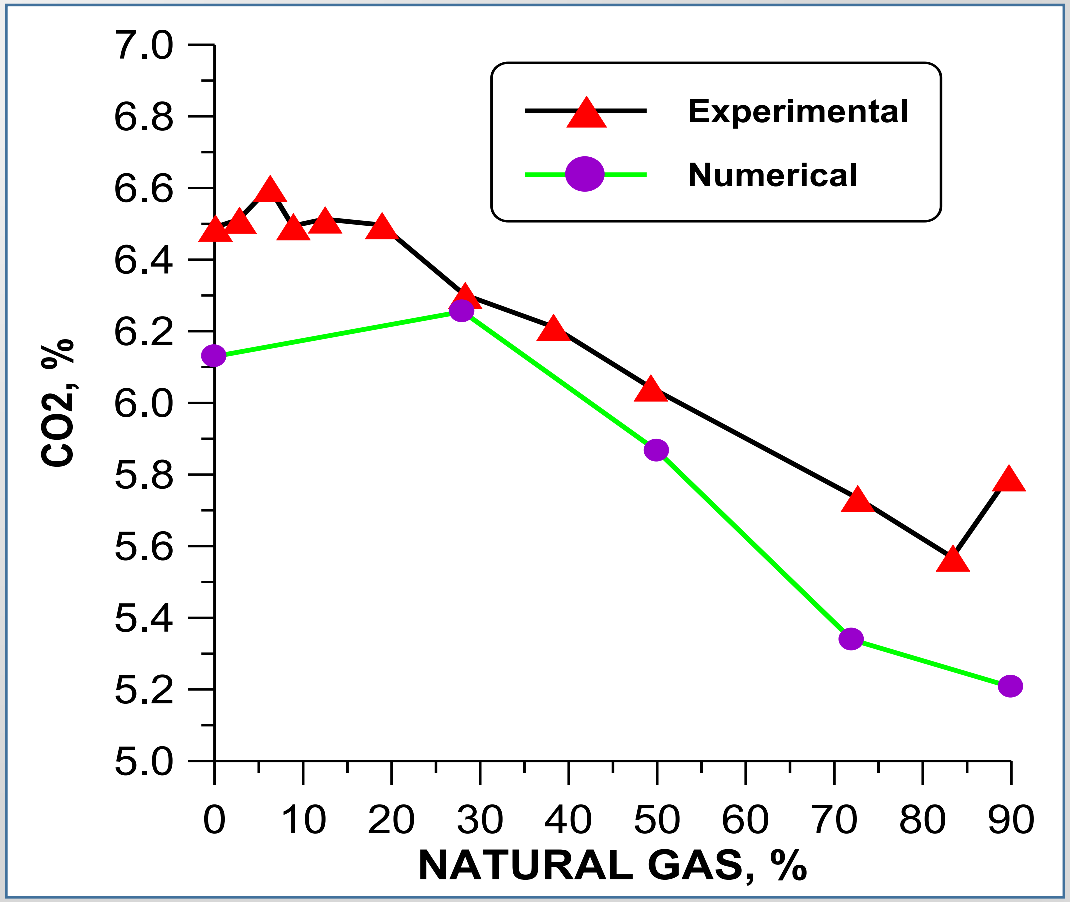
- *Several numerical tests have been performed by using a 3D Ansys-Fluent Code and Kiva-3V solver*
- *The full computational domain includes inlet and exhaust ducts and valves geometry*
- *A reliable estimation of the cylinder filling and of the interaction between flow field and spray is possible*



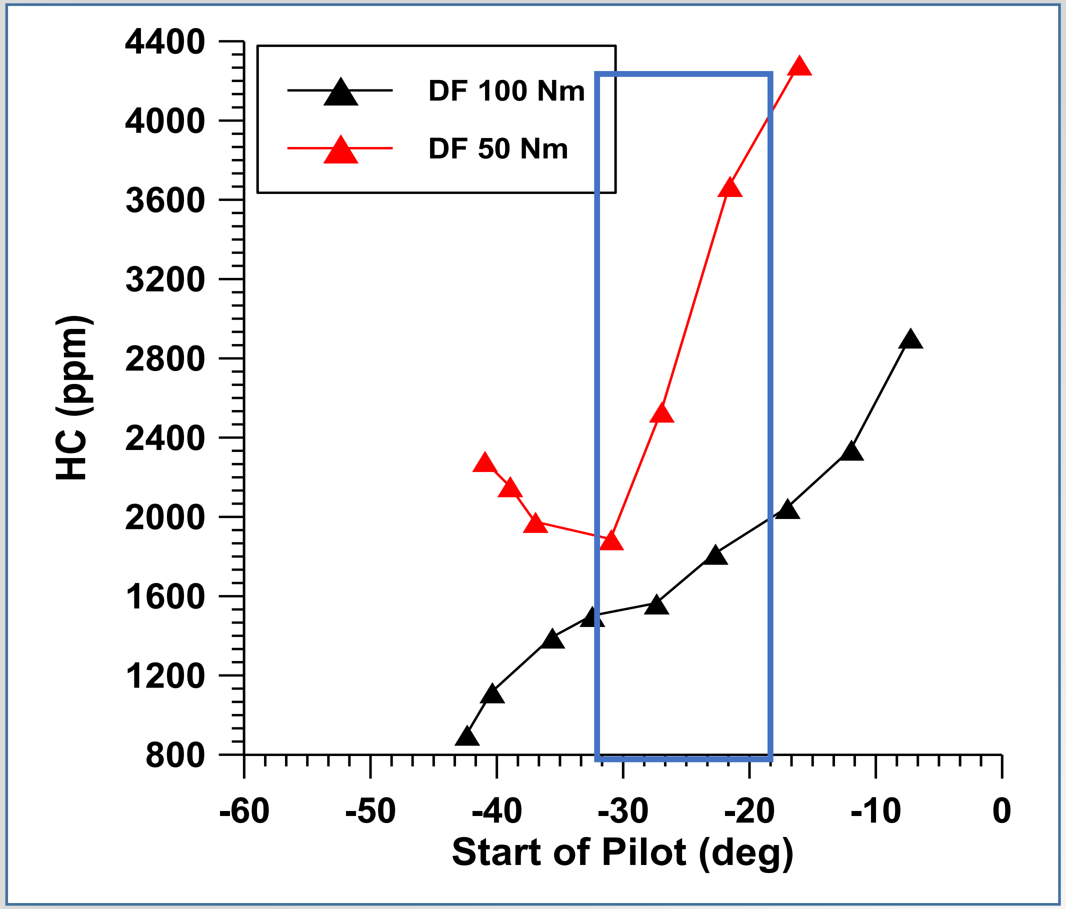
Engine	**1.9 Multijet**
Type	**4 stroke**
Number of cylinders	**4**
Cylinder vol. displ. (cm^3)	**483**
Valves /cylinder	**2**
Stroke (mm)	**90.4**
Bore (mm)	**82**
Compression Ratio	**18:1**

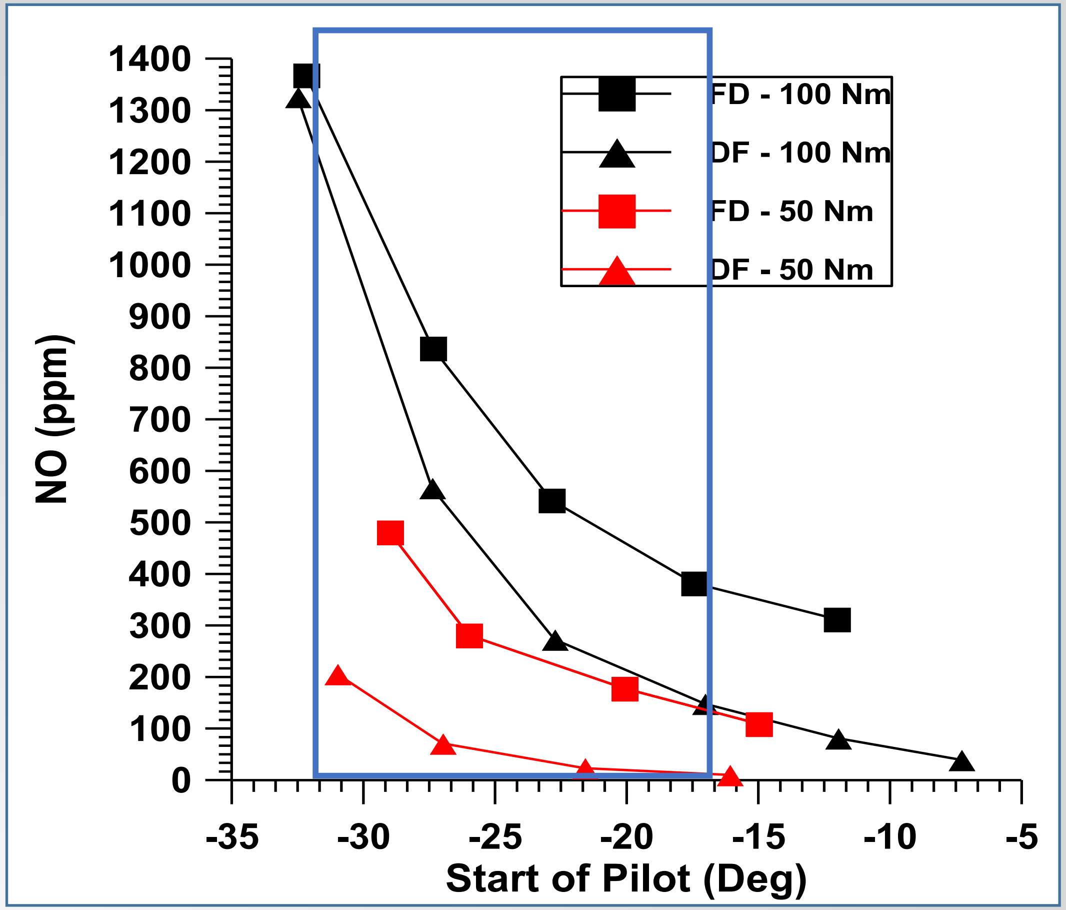
Sensitivity Analysis: different NG levels





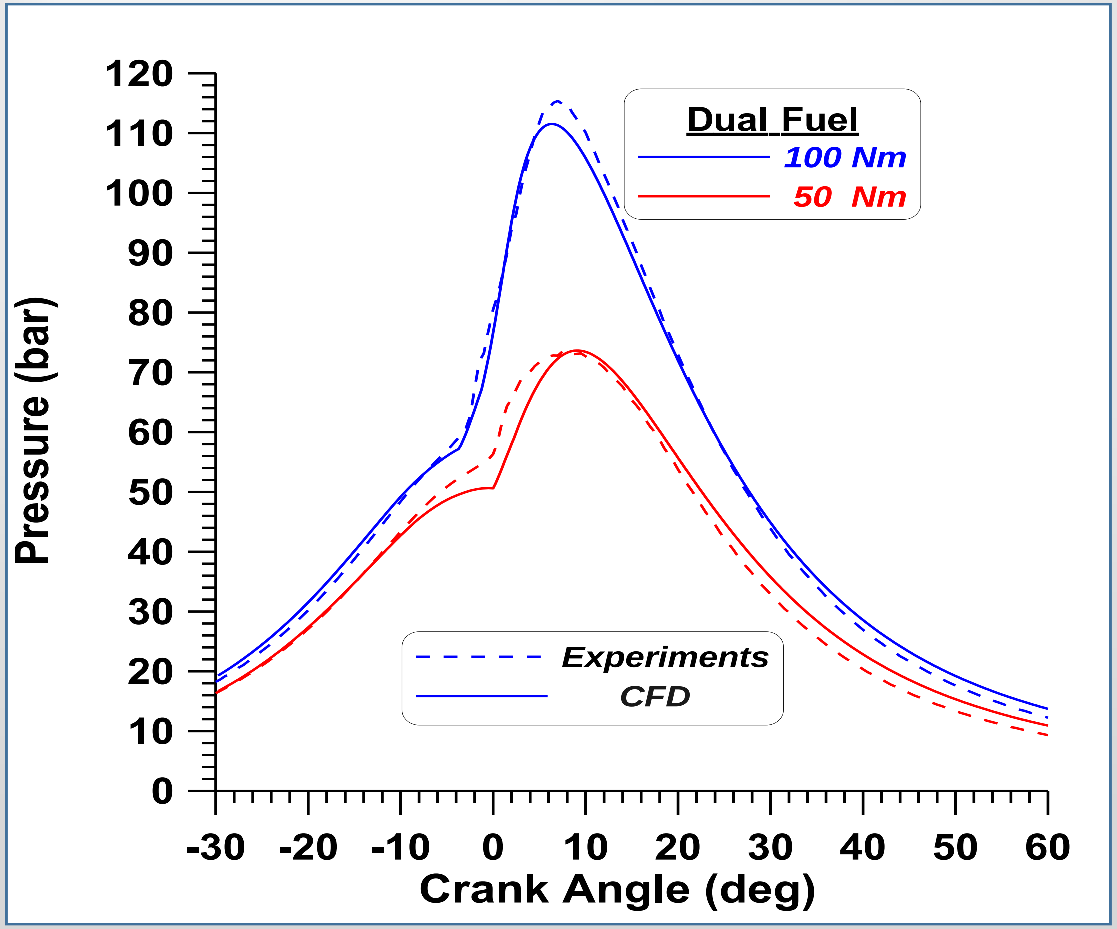
Sensitivity Analysis: different SOP

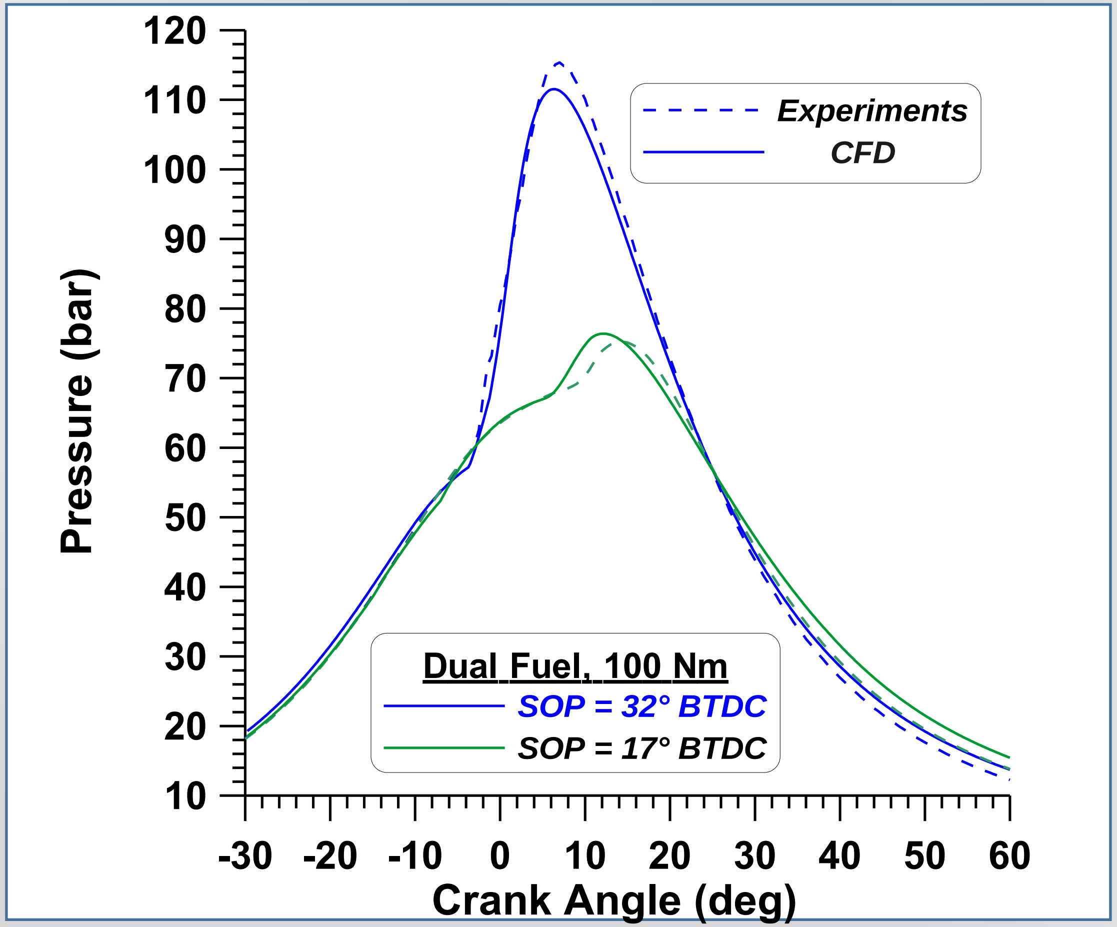




Methane Percentace is fixed at 80%

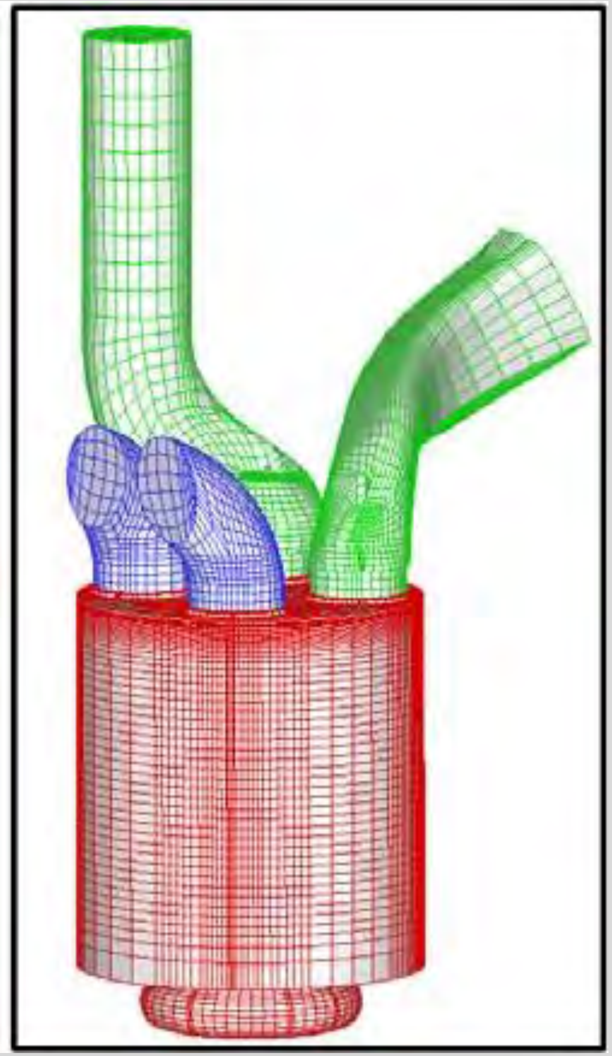
Pressure Cycle





New Engine

Engine	Turbo-charged Multijet
Type	**4 stroke**
Number of cylinders	**4**
Cylinder vol. displ. (cm^3)	**499**
Valves /cylinder	**4**
Stroke (mm)	**90**
Bore (mm)	**84**
Compression Ratio	**16:1**



One-dimensional model

The one-dimensional code is used to describe the behavior of the overall engine to provide the boundary conditions

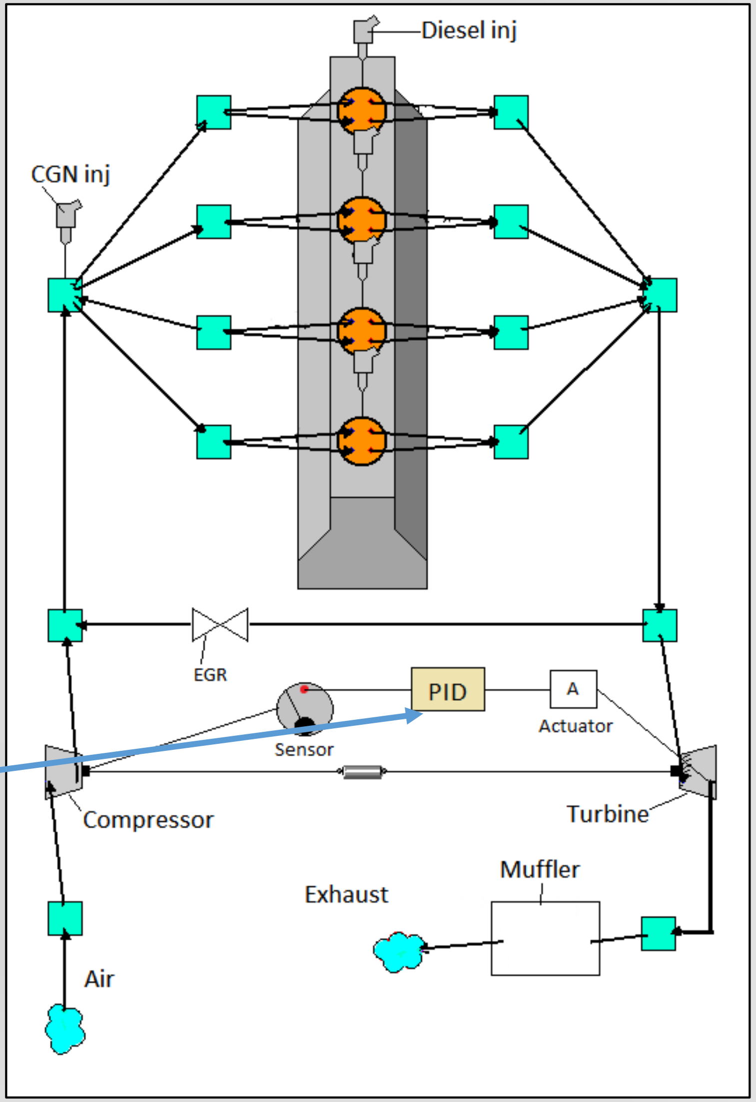
The HEAT TRANSFER MODEL considered in the simulation is the one by Annand:

$$Q = h_c A(T_g - T_w) + C\sigma(T_g^4 - T_w^4)$$

The first term represents the heat exchange by convection

The second term calculates the radiation

The PID controller acts on the vane opening of the Variable Geometry Turbine in order to obtain a fixed boost pressure at all the injection timings



Combustion Model: Multi-Component Wiebe

Allows reproduction of burning rate laws in split-injection and multi-Fuel cases

$$x_b = \left\{1 - exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right]\right\}$$

- a = is a parameter related to the combustion duration Δθ
- m = form factor

From 3D combustion results in terms of fuel oxidation rates, the Wiebe heat release model of the 1-D code has been calibrated, introducing some combustion parameters:

1) START COMBUSTION
2) PEAK POSITION
3) END OF COMBUSTION

These parameters allow the determination of a , m , Δθ in the Multi-Wiebe heat release law

After checking a quite good fitting of the combustion results respect to the CFD result, three correlations have been obtained

Test cases

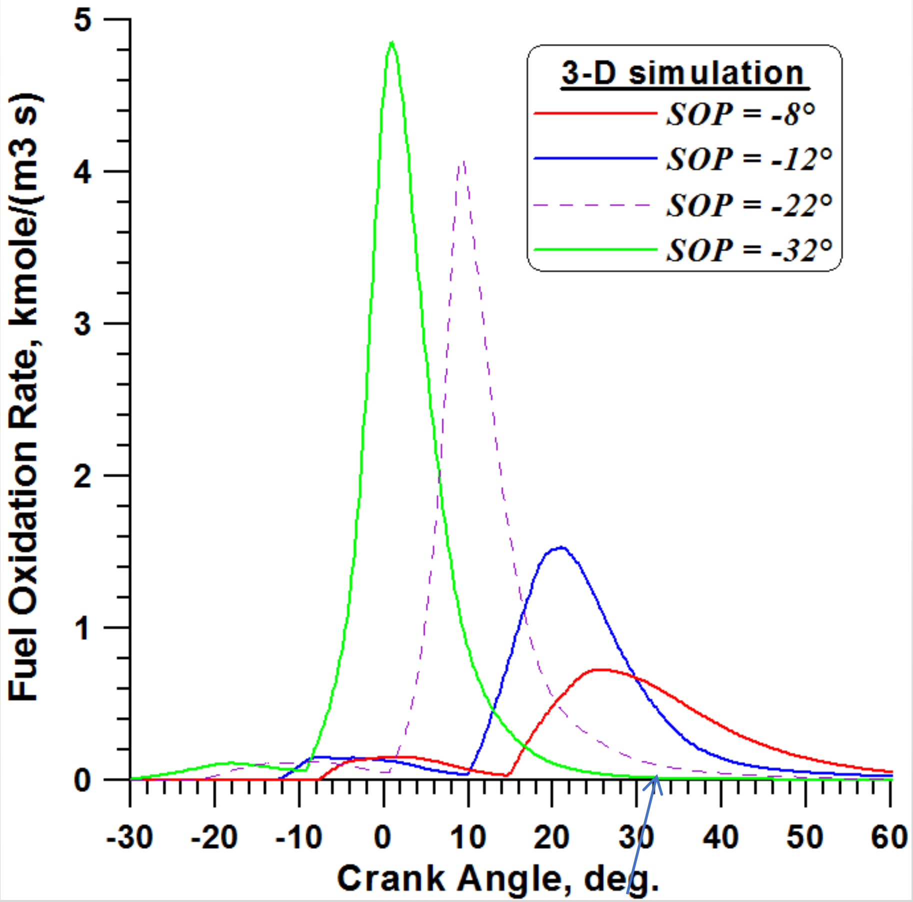
Same boundary conditions in all cases

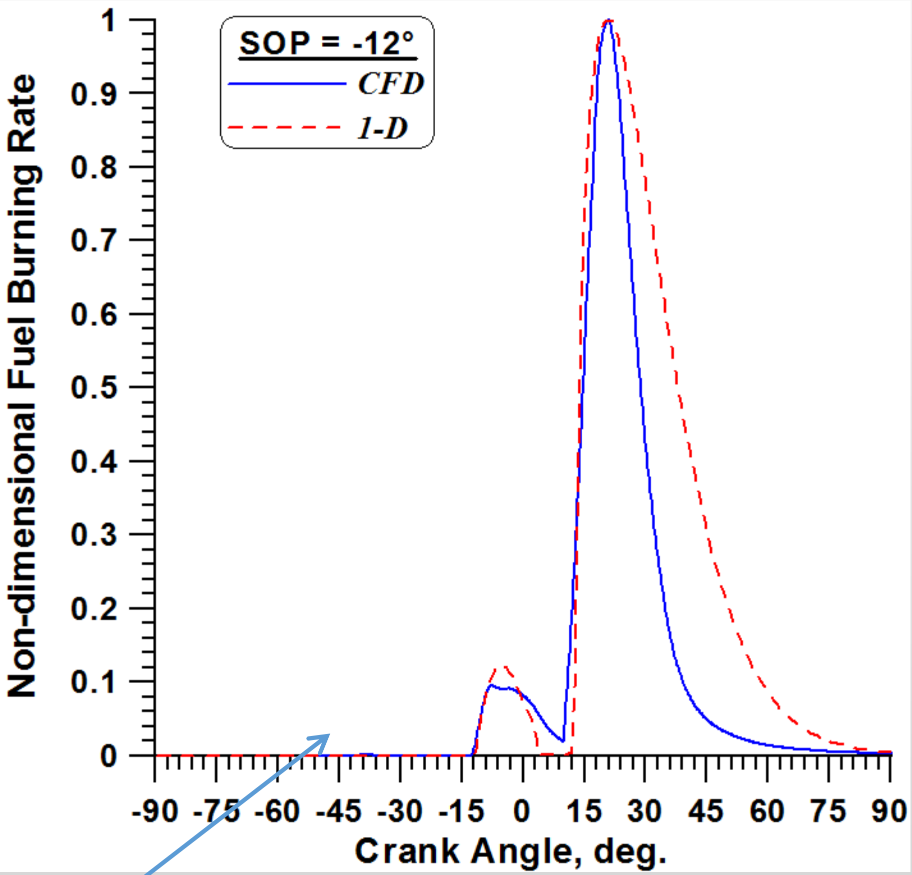
FULL DIESEL	Pilot Injection		Main Injection	
Injected Mass	1 mg		20.7 mg	
Test Cases	Start	Duration	Start	Duration
CASE 1	8° BTDC	3°	14 ATDC	8°
CASE 2	12° BTDC	3°	10° ATDC	8°
CASE 3	17° BTDC	3°	5° ATDC	8°
CASE 4	22° BTDC	3°	TDC	8°
CASE 5	27° BTDC	3°	5°BTDC	8°
CASE 6	32° BTDC	3°	10° BTDC	8°

DUAL FUEL	Pilot Injection		Main Injection	
Injected Mass	1 mg		2.8 mg	
Test Cases	Start	Duration	Start	Duration
CASE 4	12° BTDC	3°	10° ATDC	5°
CASE 5	17° BTDC	3°	5°ATDC	5°
CASE 6	27° BTDC	3°	5° BTDC	5°

Several test-cases were carried out for setting up the integrated procedure between 1-D and 3-D based approaches

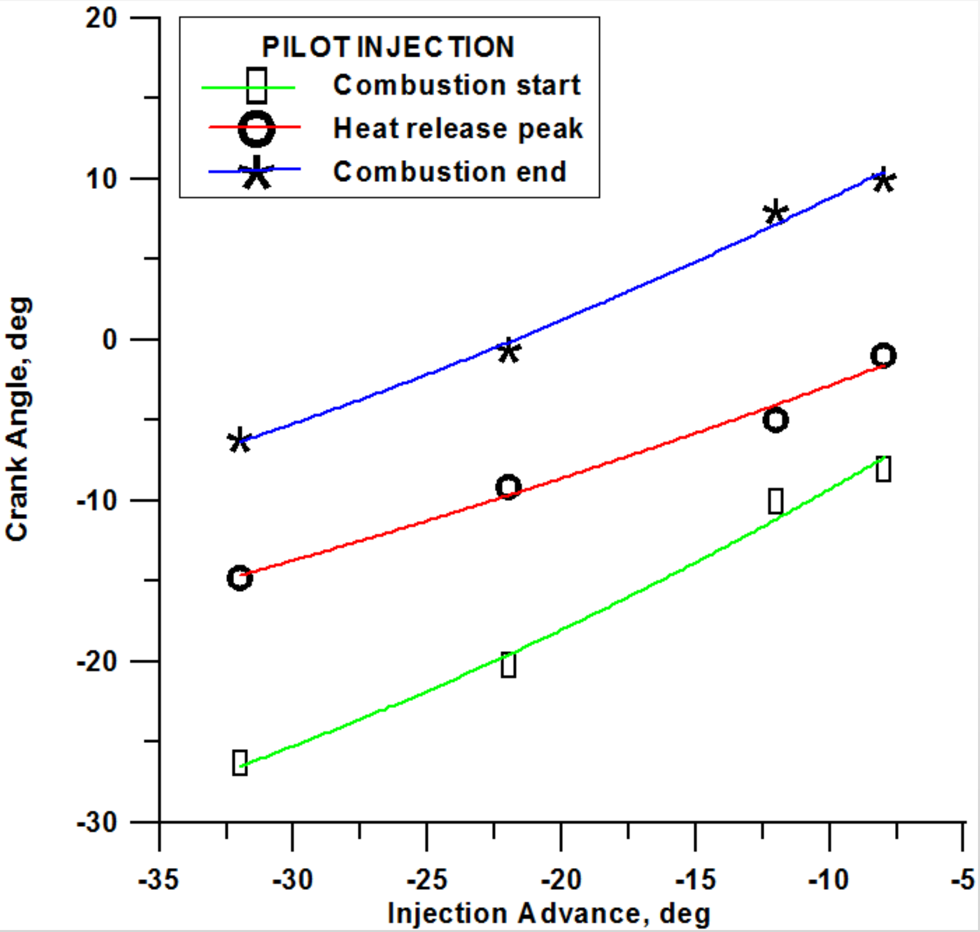
Full Diesel: Fuel Burning rate

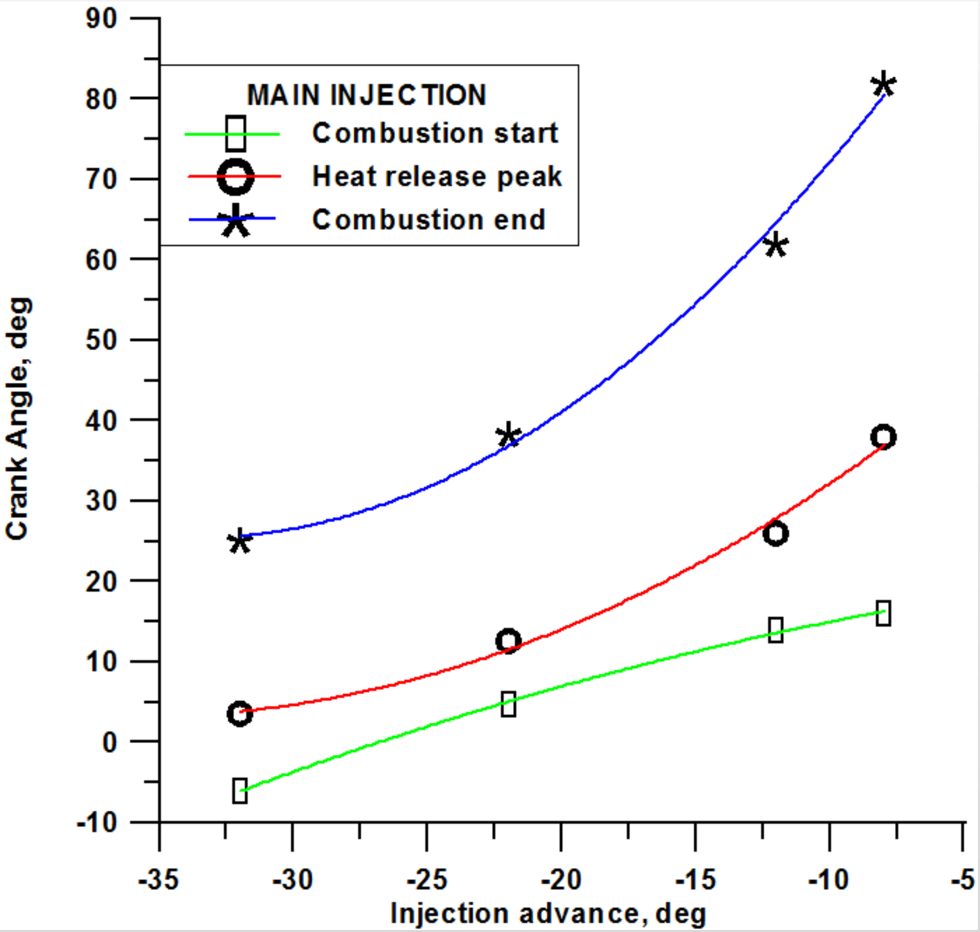




From 3D combustion results in terms of fuel oxidation rates the Wiebe heat release model of the 1-D code has been calibrated.

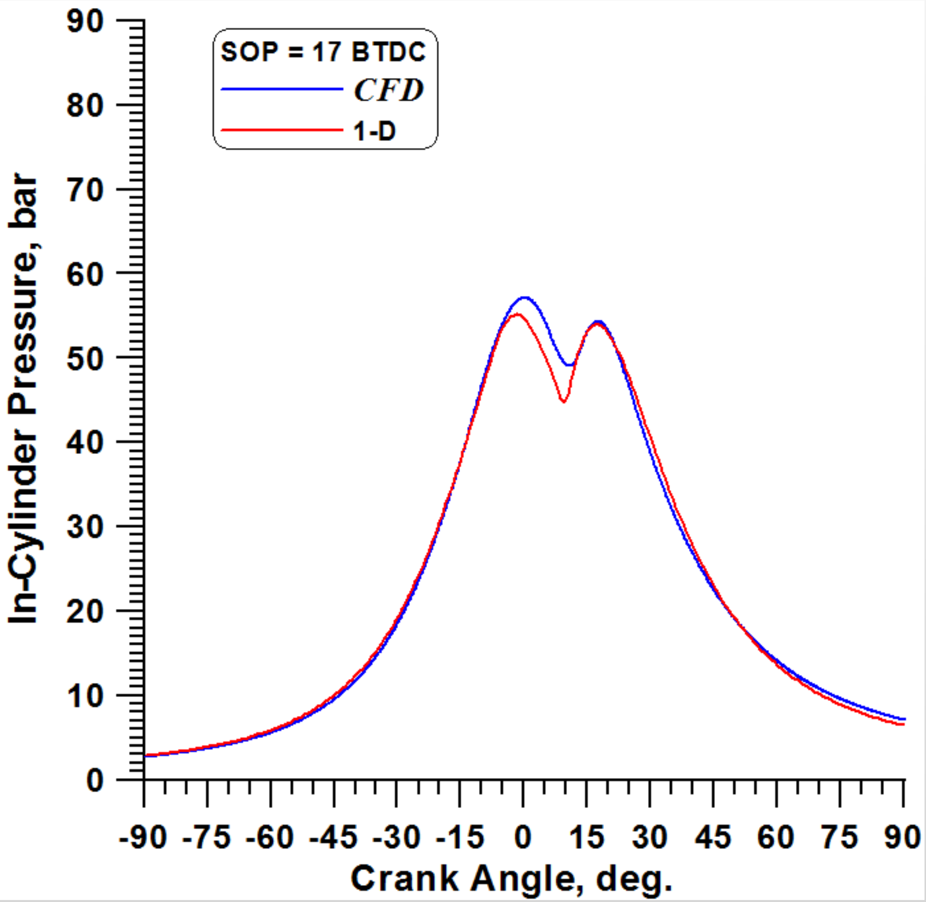
Full Diesel: Correlations

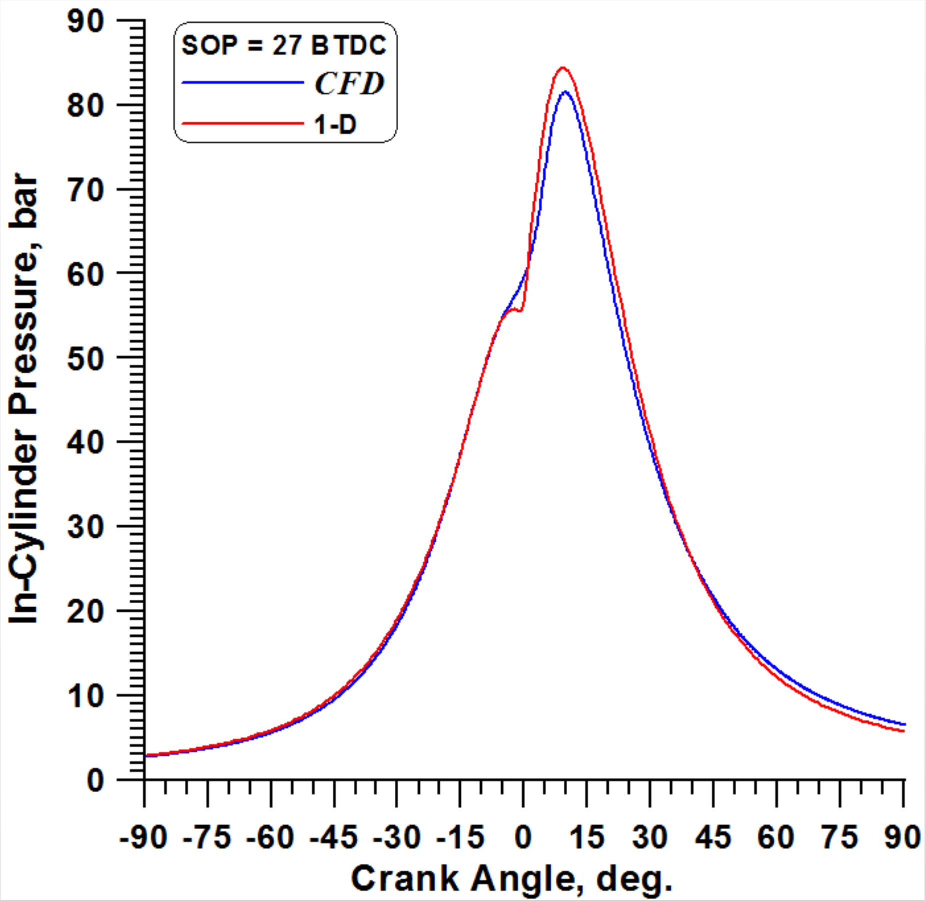




Correlations obtained by using second order polynomial curves

Full Diesel: Validation

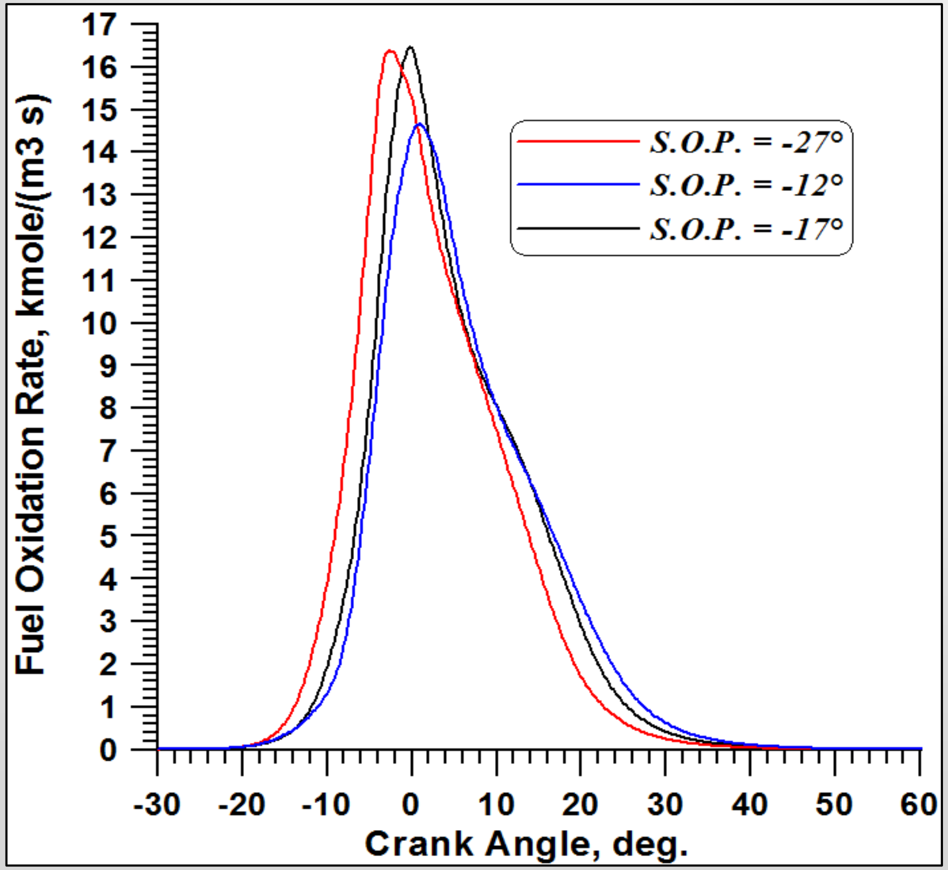




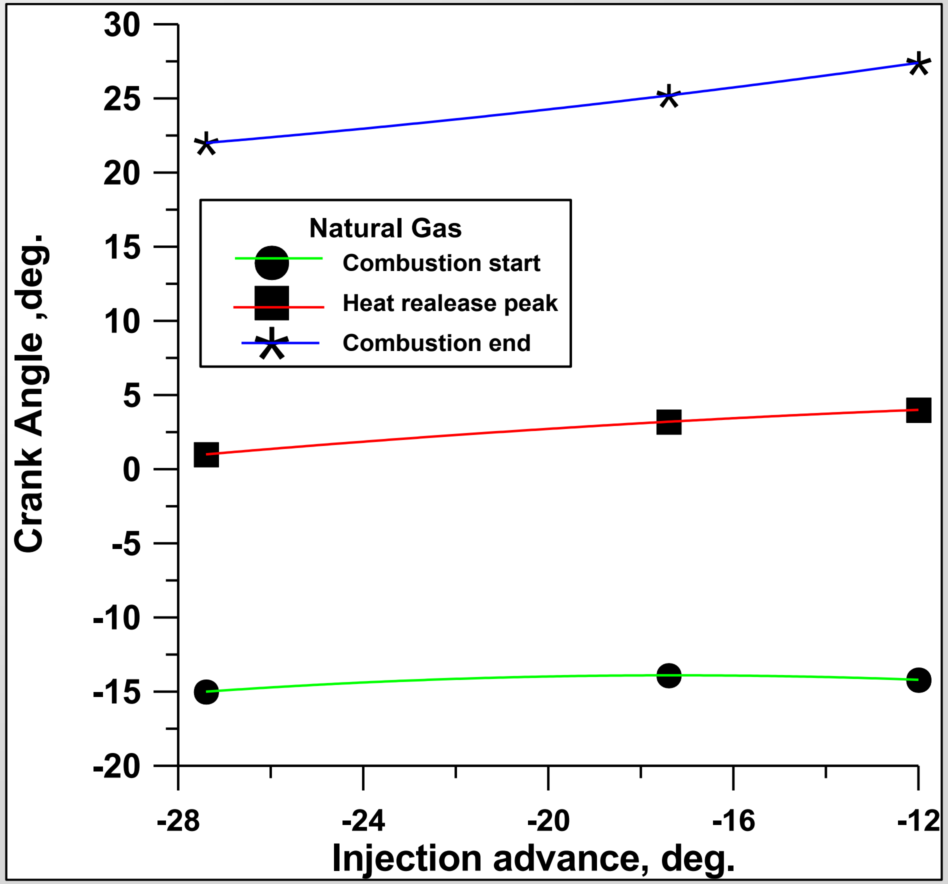
The two test cases (#3 and #5) with SOP equal to 17°and 27° BTDC have been performed by the two codes

The result comparison validates the heat release correlations

Dual Fuel: FBR and Correlations

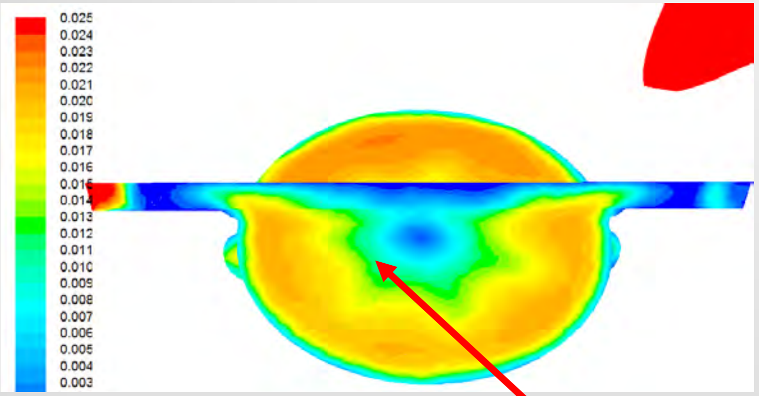


Methane oxidation rates

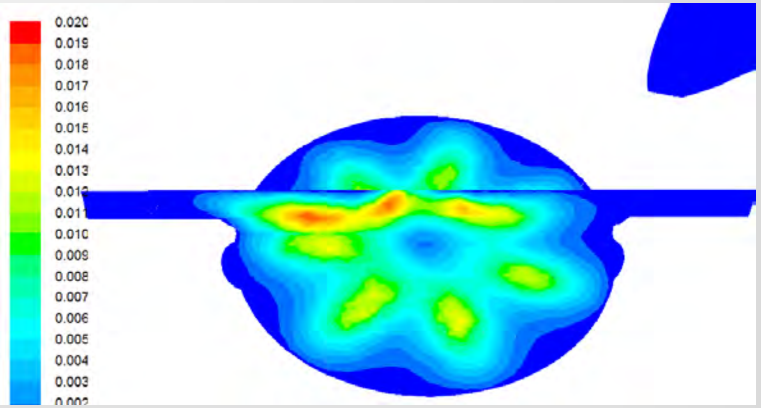


Correlation curves of Wiebe model for natural gas

Methane, Diesel Fuel Vapour mass fractions and reaction rates distributions

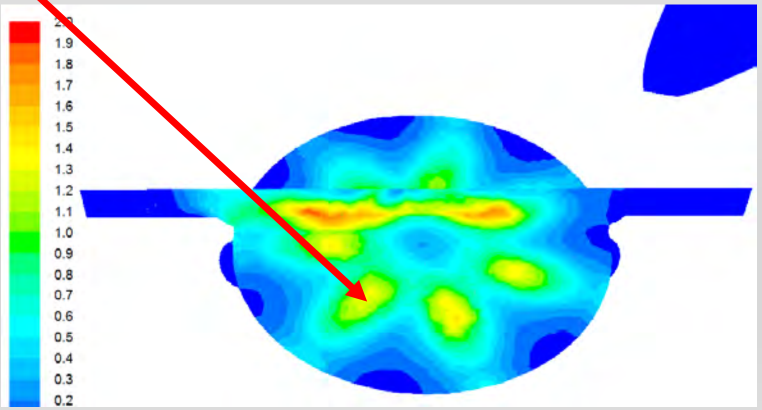


Methane



Diesel Vapour

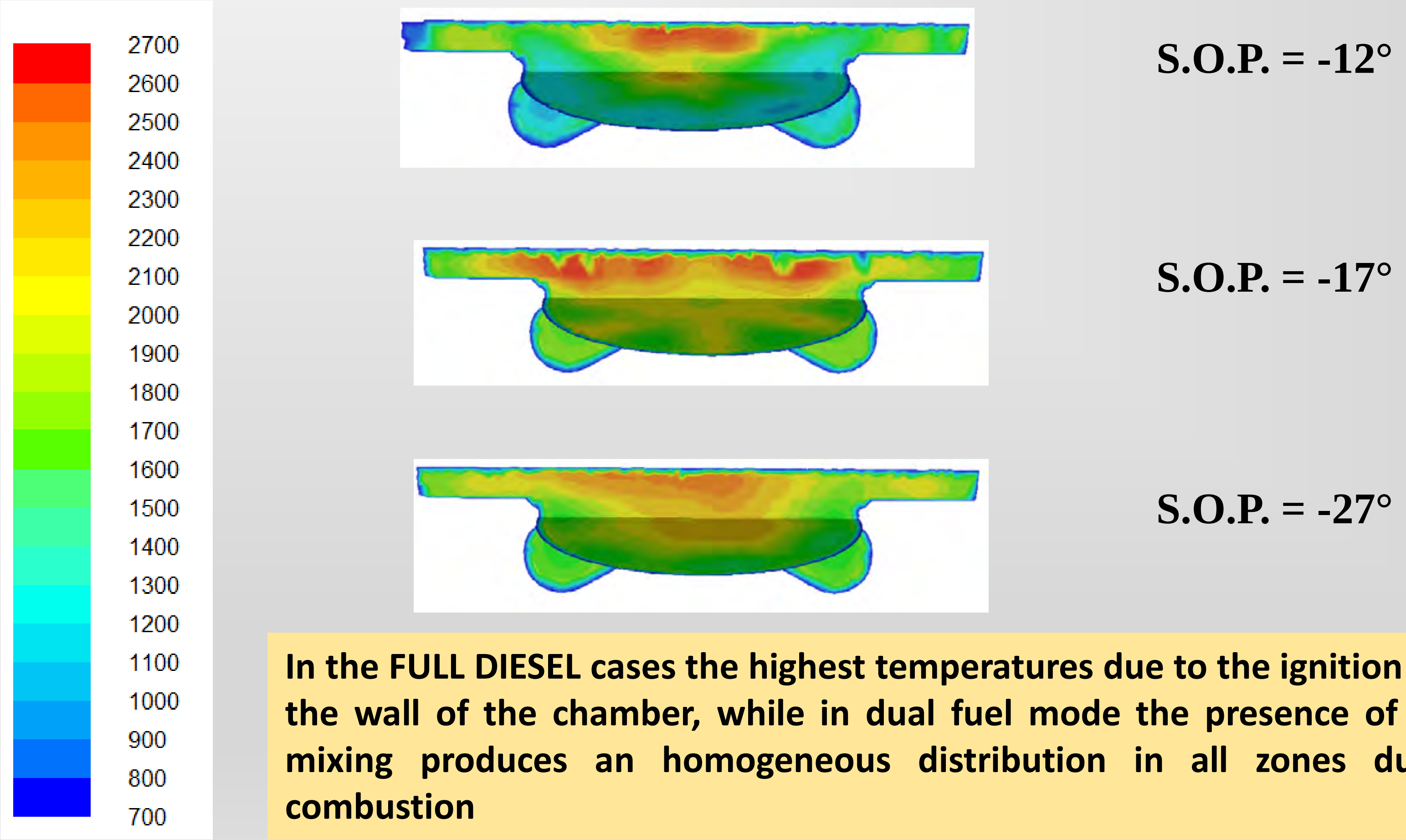
The interaction between the two fuels confirms that the role of the diesel fuel combustion is to allow a faster consumption of the gaseous fuel.



Diesel Fuel Oxidation Rate [kmole/(m3 s)]

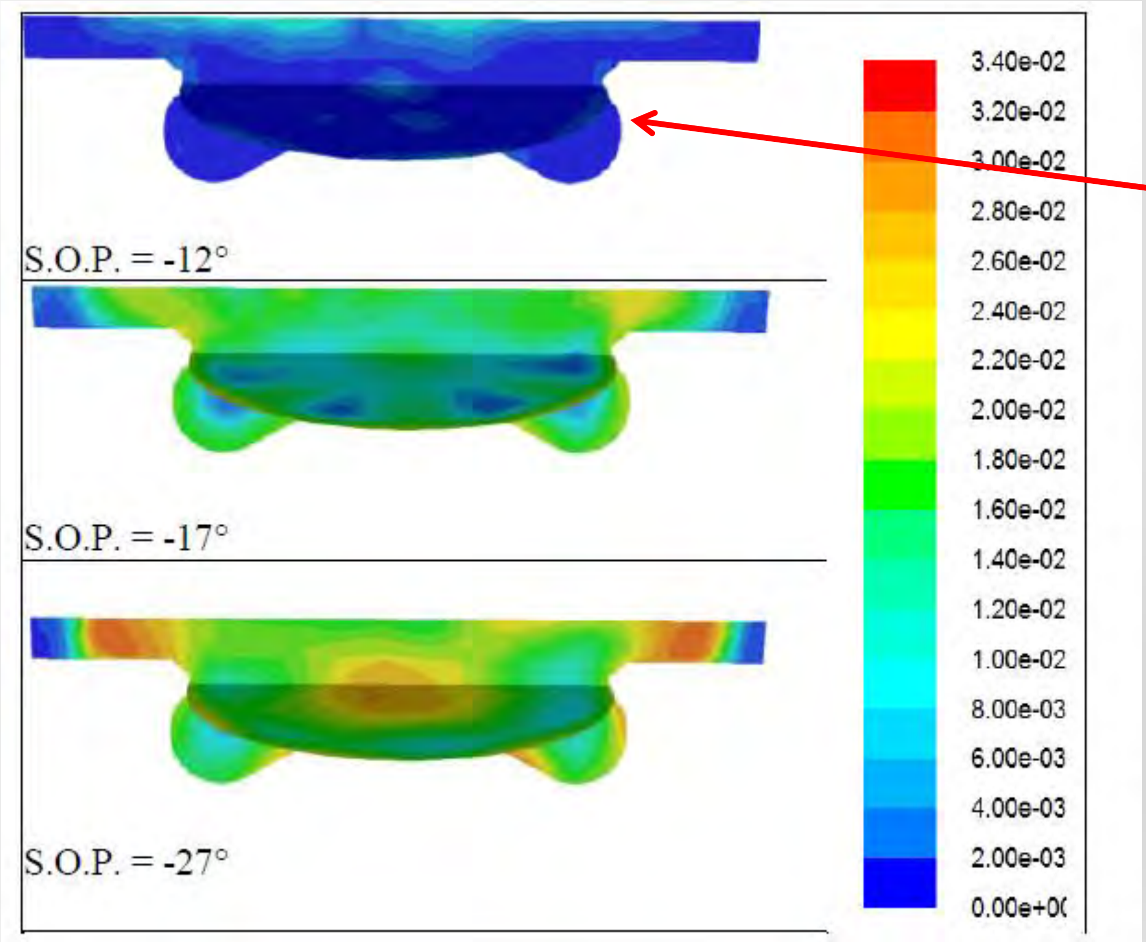
SOP = 12° BTDC
At 20° ATDC

Temperature distributions for different injection timings - At 20° ATDC



In the FULL DIESEL cases the highest temperatures due to the ignition appear near to the wall of the chamber, while in dual fuel mode the presence of NaturalGas/air mixing produces an homogeneous distribution in all zones due to a rapid combustion

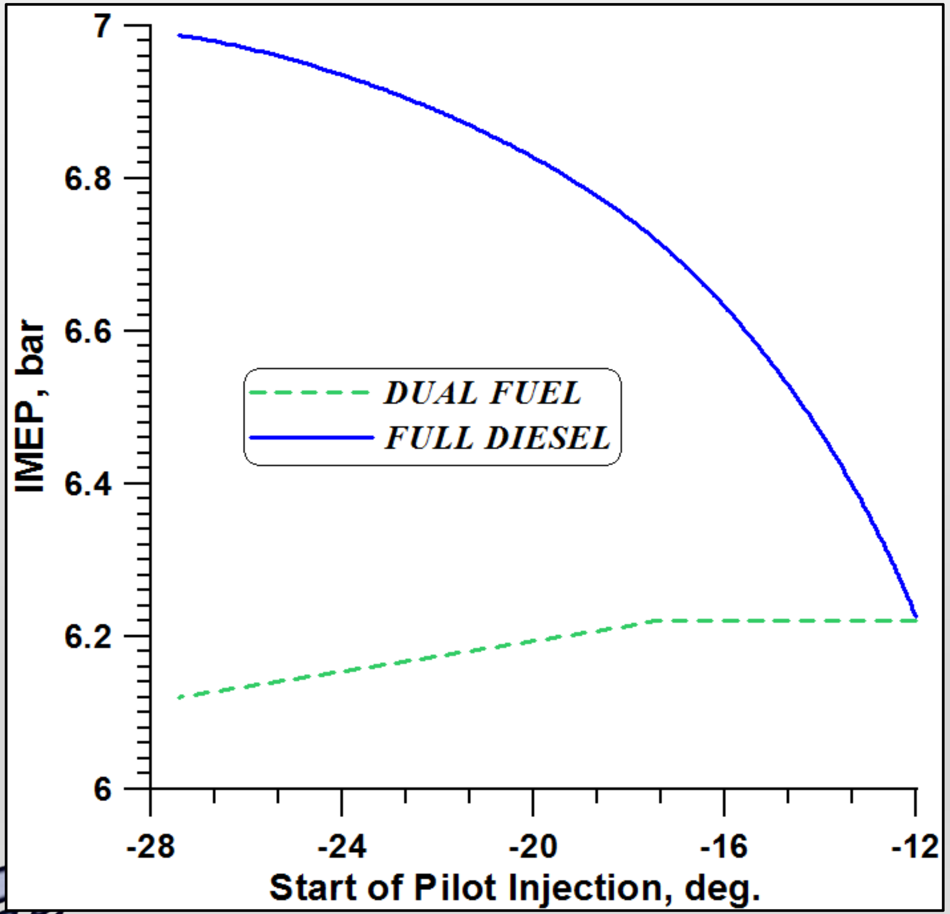
NO mass fraction distributions for different injection timings - At 20° ATDC



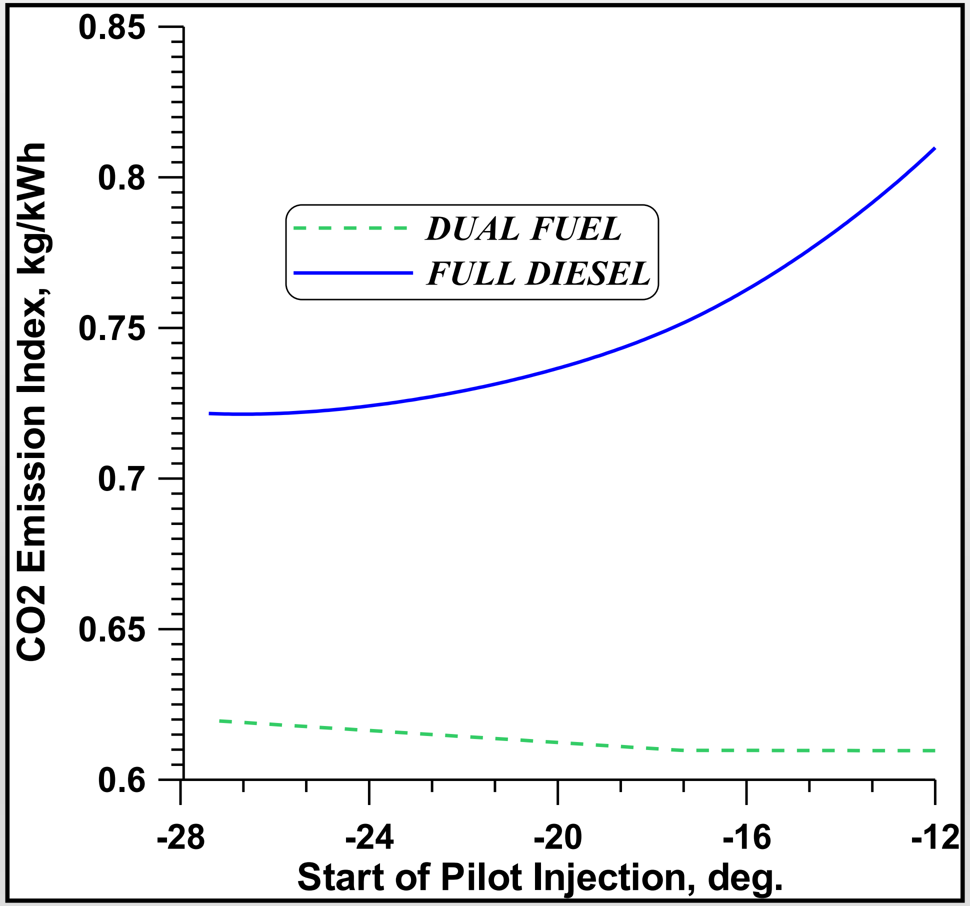
low NO amount

A Comparison Dual Fuel- Full Diesel

IMEP calculated by 1-D model for full diesel and dual fuel modes



CO_2 emissions calculated by 1-D model for full diesel and dual fuel modes



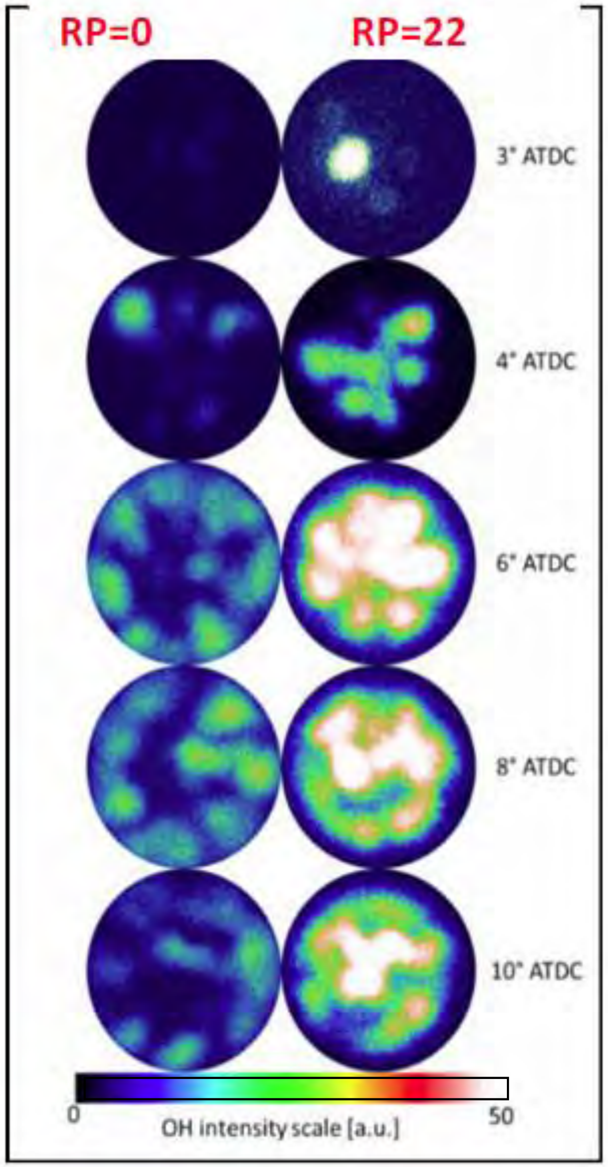
A detailed visualization of combustion

- Research, optically accessible engine
- Pressure cycle
- Pollutant emissions
- OH Chemiluminescence

OH radical represents a significant signal of the combustion rate intensity

$$CH_4+O_2 \rightleftharpoons HO_2+CH_3$$
$$CH_4+HO_2 \rightleftharpoons H_2O_2+CH_3$$
$$CH_4+OH \rightleftharpoons H_2O+CH_3$$
$$CH_3+O_2 \rightleftharpoons CH_2O+OH$$
$$CH_2O+OH \rightleftharpoons CHO+H_2O$$
$$CHO+O_2 \rightleftharpoons CO+HO_2$$
$$CHO+M \rightleftharpoons CO+H+M$$
$$H+O_2+M \rightleftharpoons HO_2+M$$
$$H_2O_2+M \rightleftharpoons 2OH+M$$

**Li & Williams model*



Operating conditions

RP, %	0	15	22
Thermal Power (fuel LHV), kW	4.78	5.6	6.125
IMEP. Bar	1.4	2.3	3.75
Trapped Air Mass, mg/cycle	786	764	742
Injected diesel oil, mg/cycle	8.9	8.9	8.9
Trapped Methane Mass, mg/cycle	0	1.32	2.14
Methane mass fraction	0	0.00172	0.00287
Air / Fuel Ratio	88.39	74.80	67.23
Fuel / Air Equivalence Ratio	0.164	0.199	0.24
Methane / Air Equivalence Ratio	0	0.0298	0.0497

$$f_{st,CH_4} = \frac{m_{CH_4}}{m_{air}} = 0.0578$$

and

$$\varphi = \frac{f}{f_{st,CH_4}}$$

Far below the flammability limits

Diesel combustion is necessary to ignite the methane/air mixture

Models

Diesel oil

- Atomization: **WAVE**

$$\tau_{BU} = 3.726 \frac{B_1 a}{\Lambda \Omega}$$

Where $B_1 = 1.73$

- Self-ignition delay by **Assanis**:

$$\tau_{ID} = 2.4\phi^{-0.2}\overline{p}^{-1.02}\exp\left(\frac{E_a}{R_u \overline{T}}\right)$$

Natural gas

- Self-ignition delay by **Li&Williams** (T<1300K):

$$\tau = (2.6 \times 10^{-15}[O_2]^{-4/3}[CH_4]^{1/3})/T^{-0.92}exp(-13180/T)$$

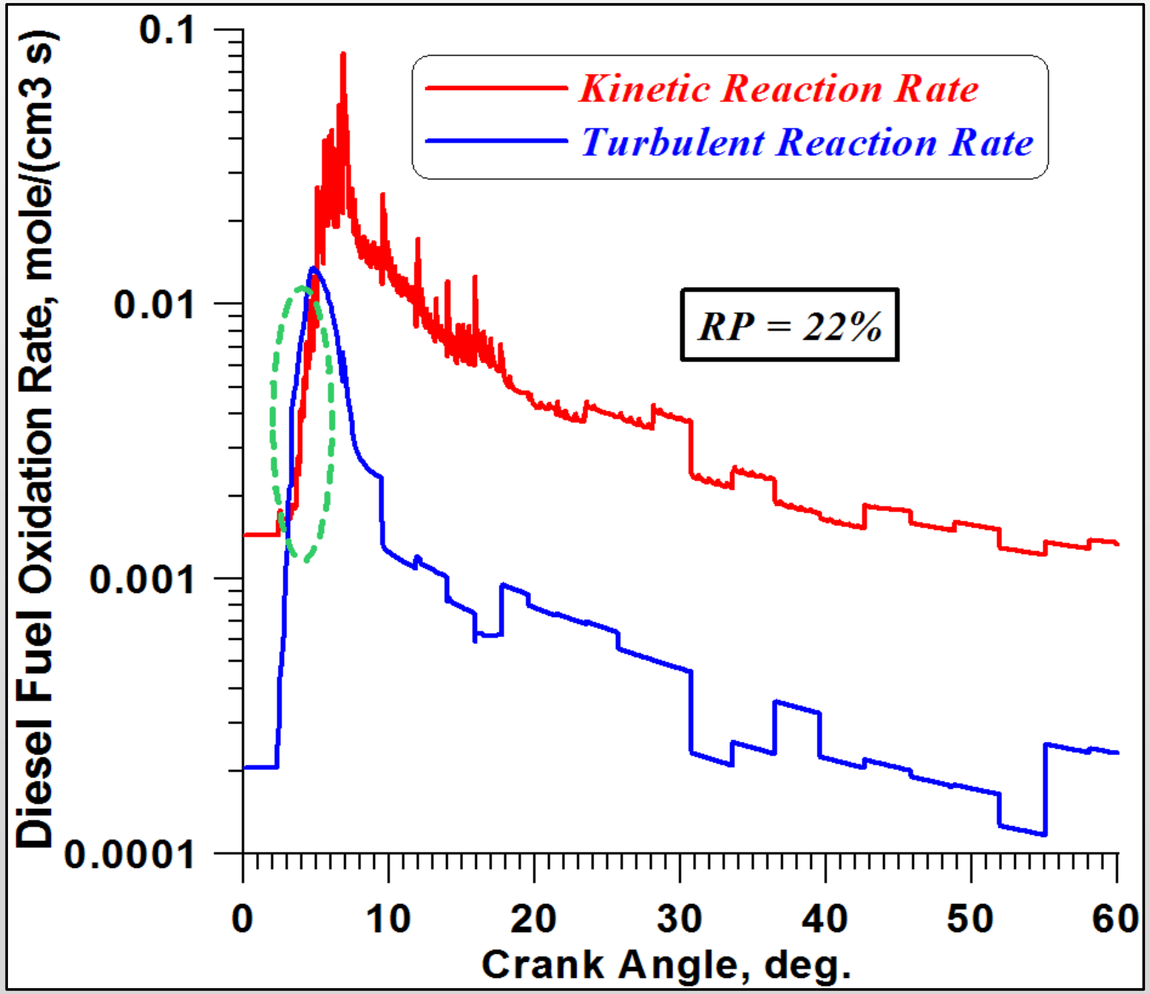
- Reaction:
  - Diesel oil: one-step
  - Natural gas: two- step (100% methane)

- Combustion: **Finite rate – Eddy dissipation**

Kinetic rate (Arrhenius) ⟷ Turbulent mixing-controlled rate (Magnussen and Hiertager)

→ Calibration

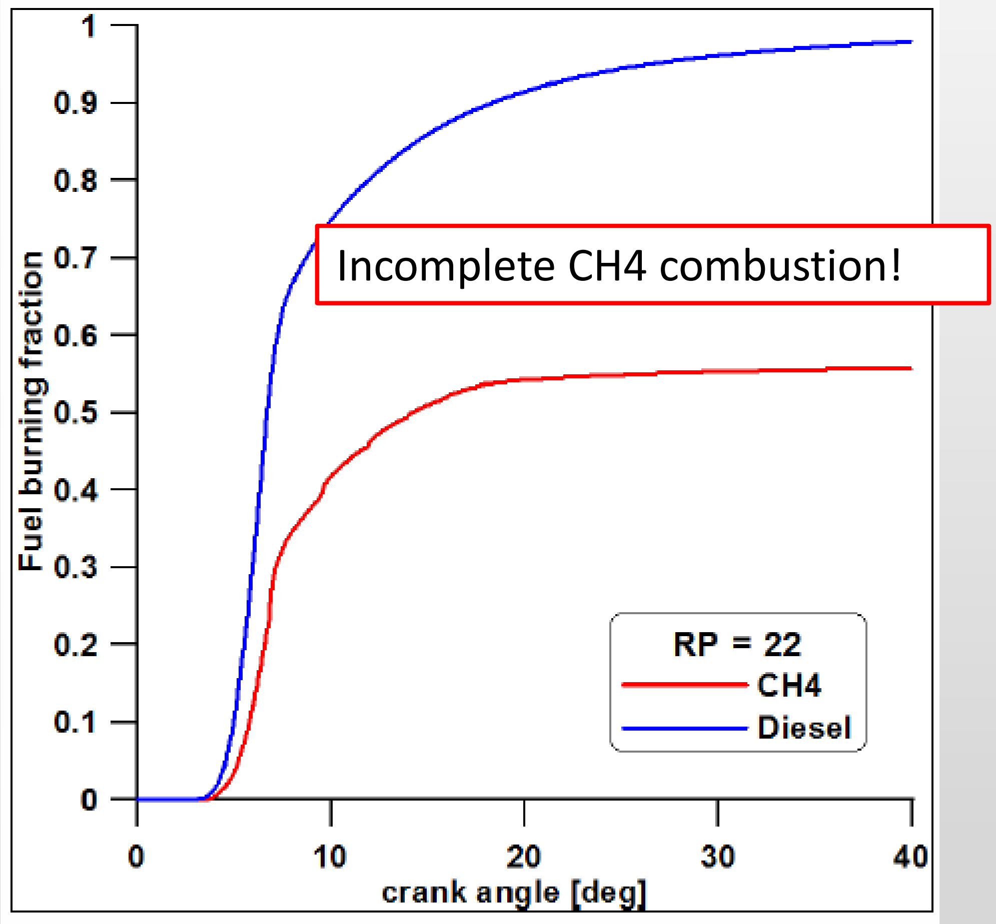
Results

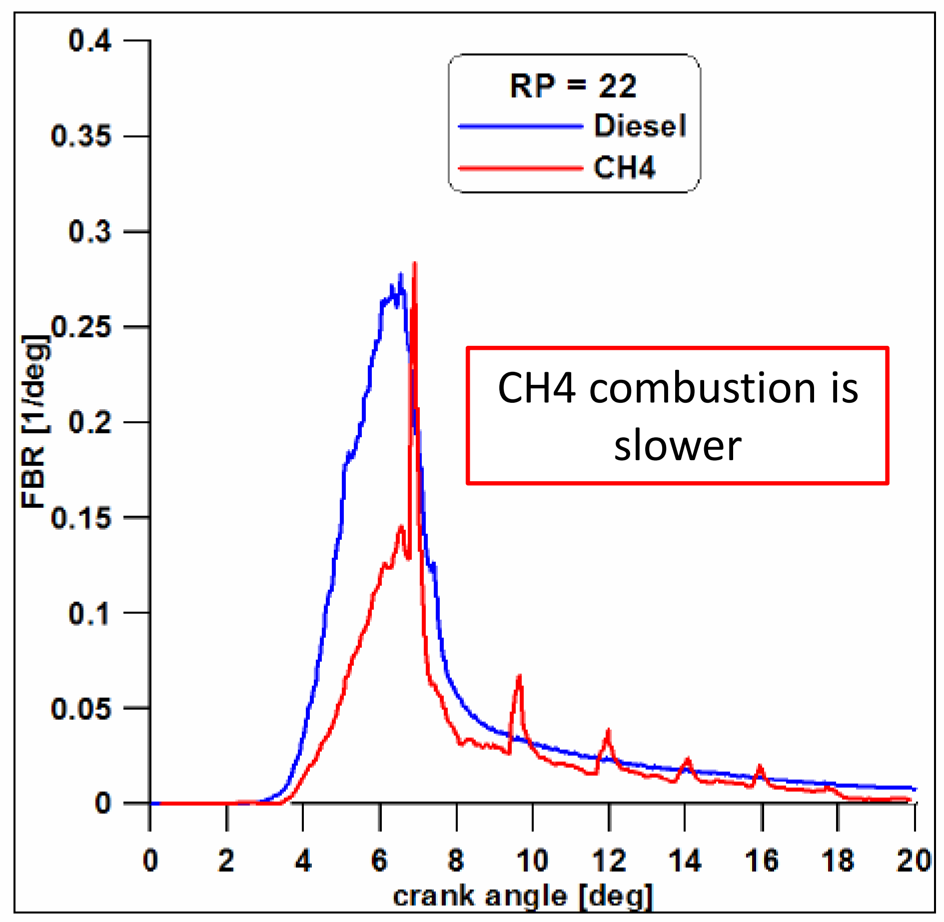


The Kinetic Reaction rate is lower than the Turbulent one in the early phase of combustion

The diesel fuel combustion is governed by the mixing-controlled mechanism in the next combustion development

Results

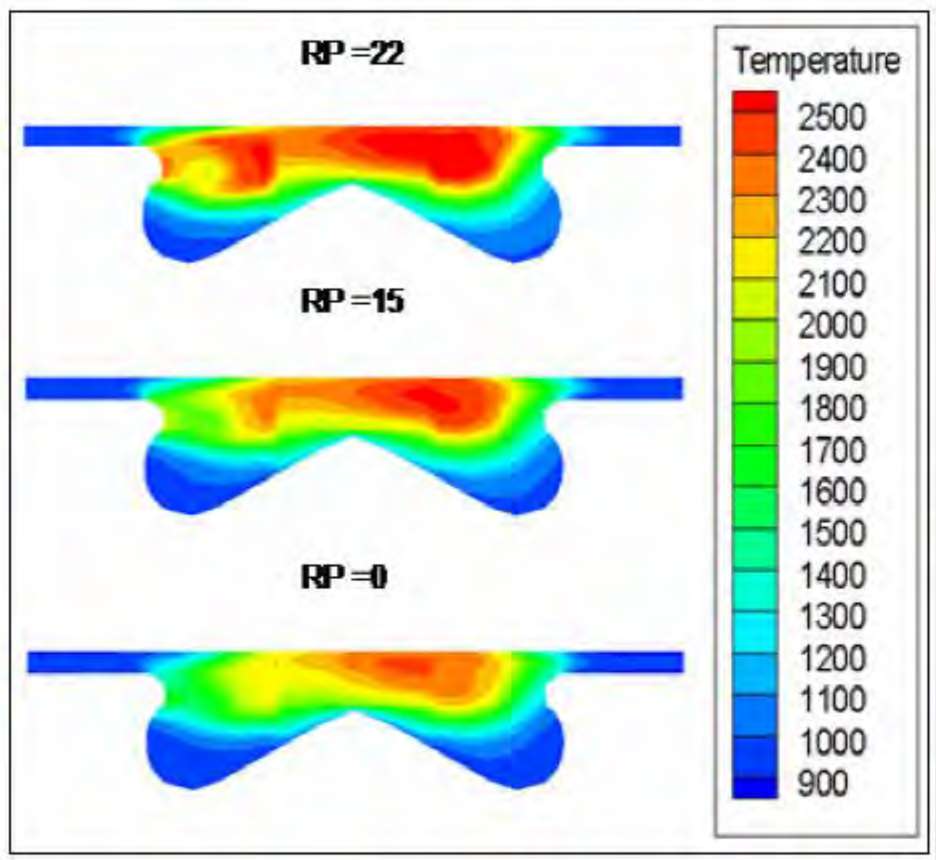


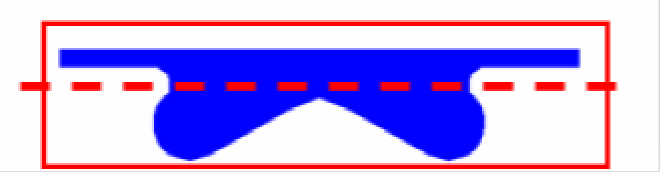


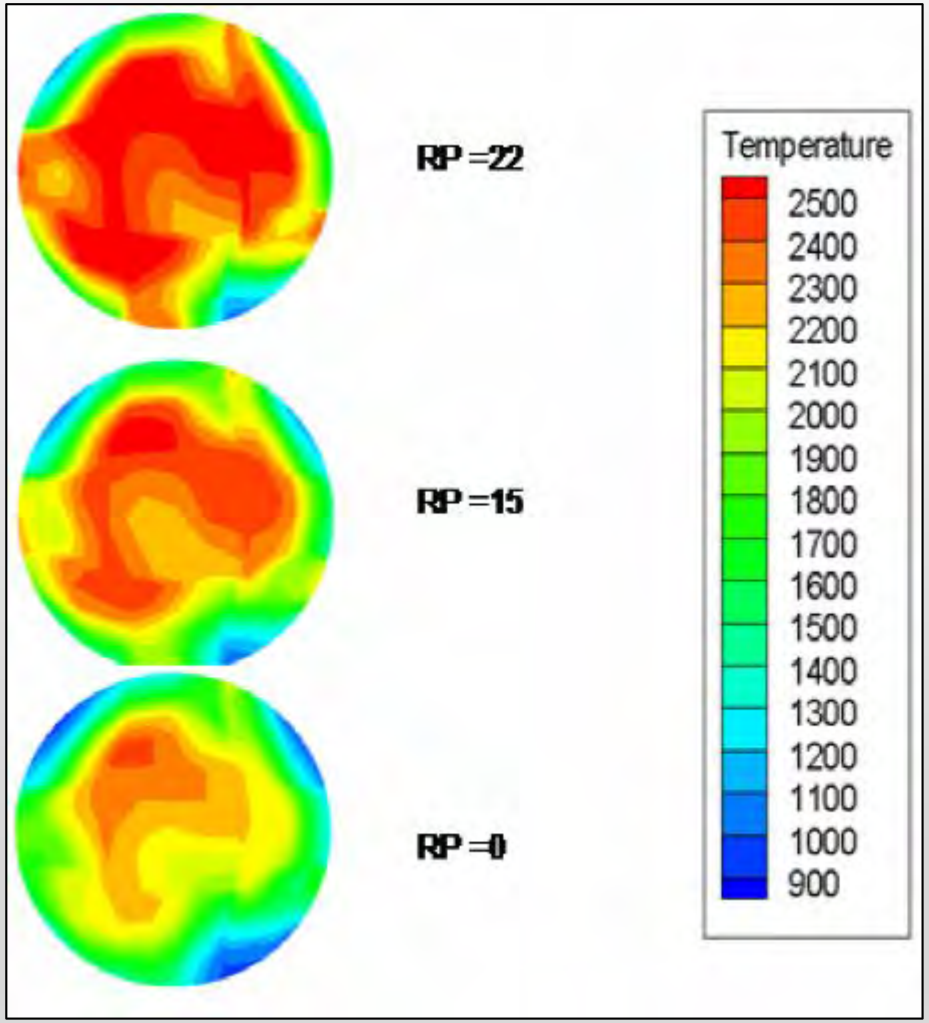
In accordance with experimental results

The FBR of the two distinct fuels can be seen only through numerical simulations

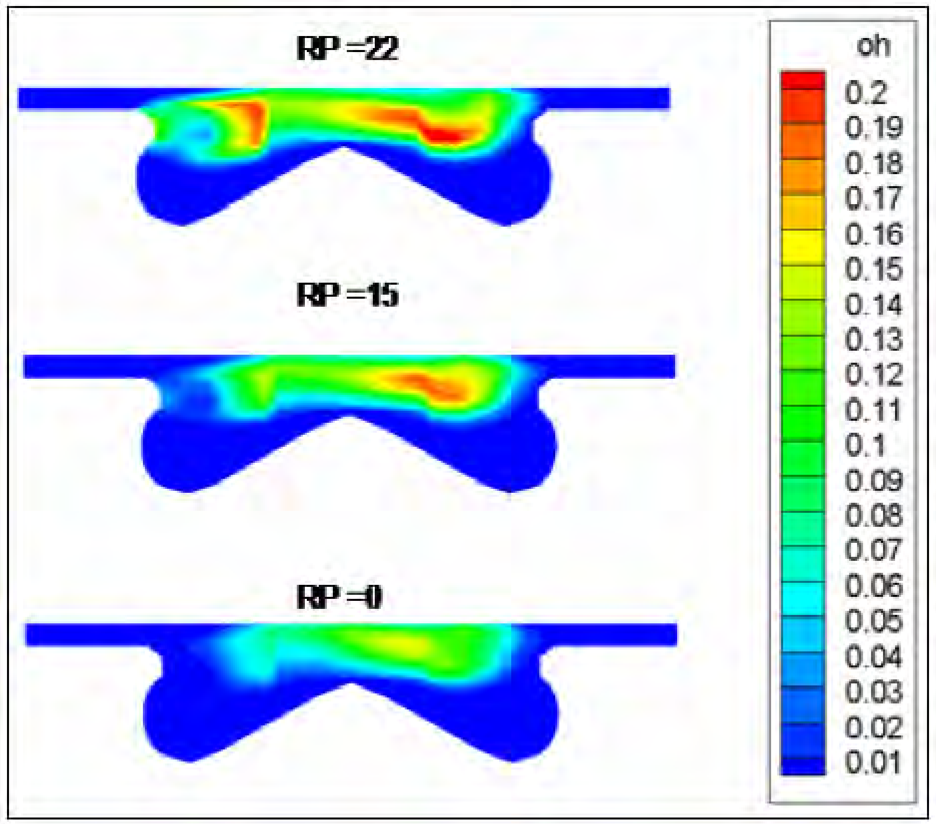
Temperature contours

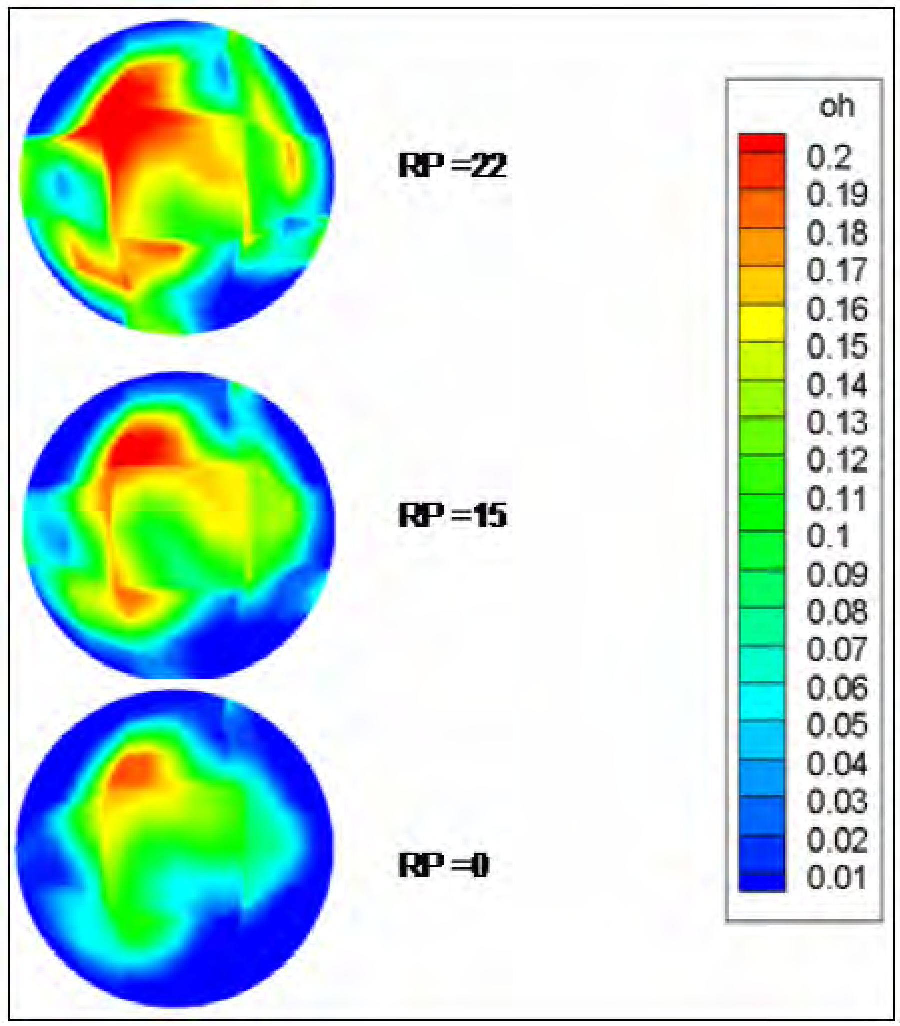






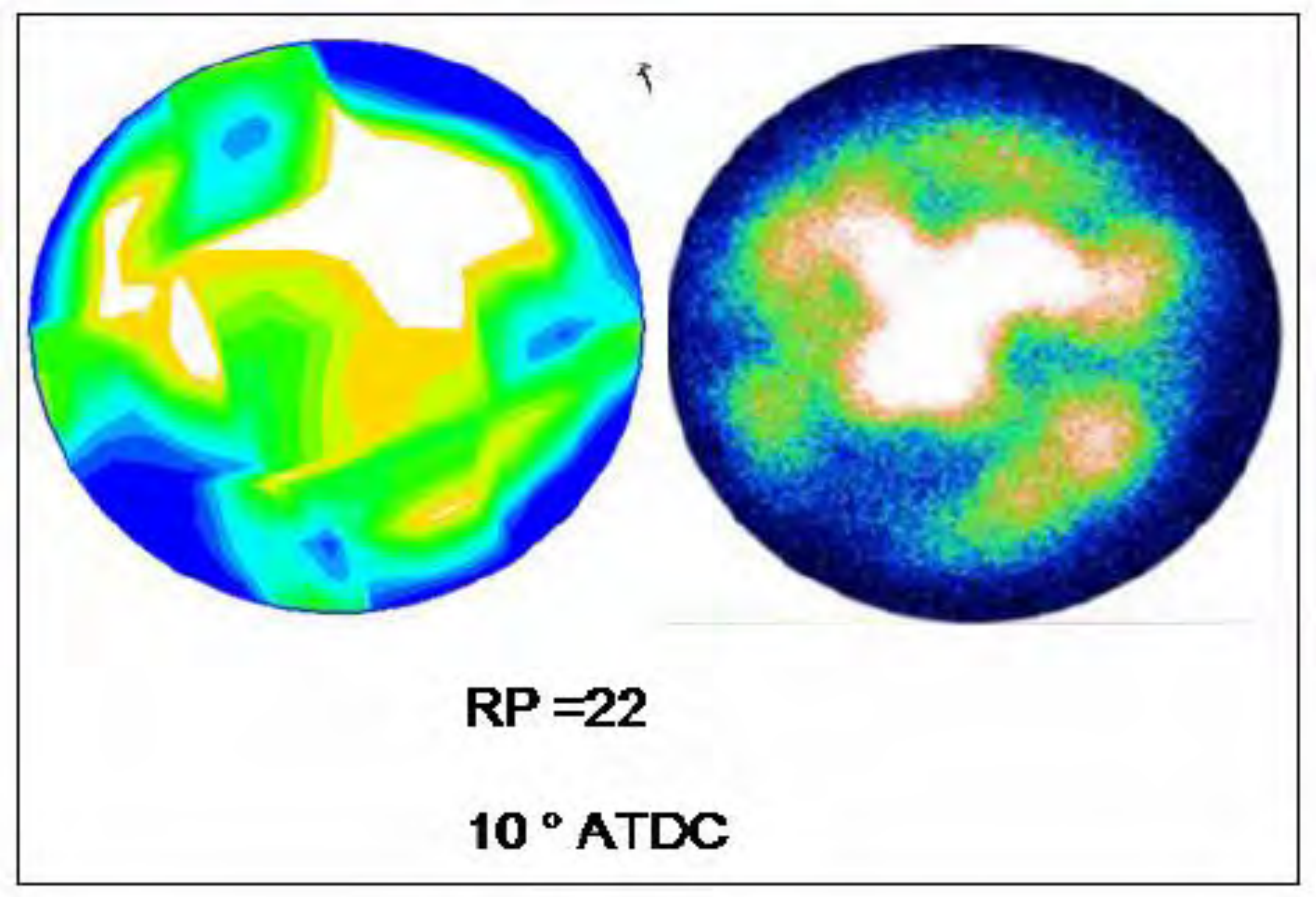
OH contours



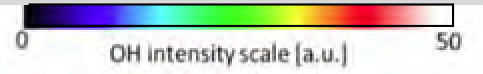


The zones with OH concentrations exhibit a fair compliance with those characterized by temperature peaks.

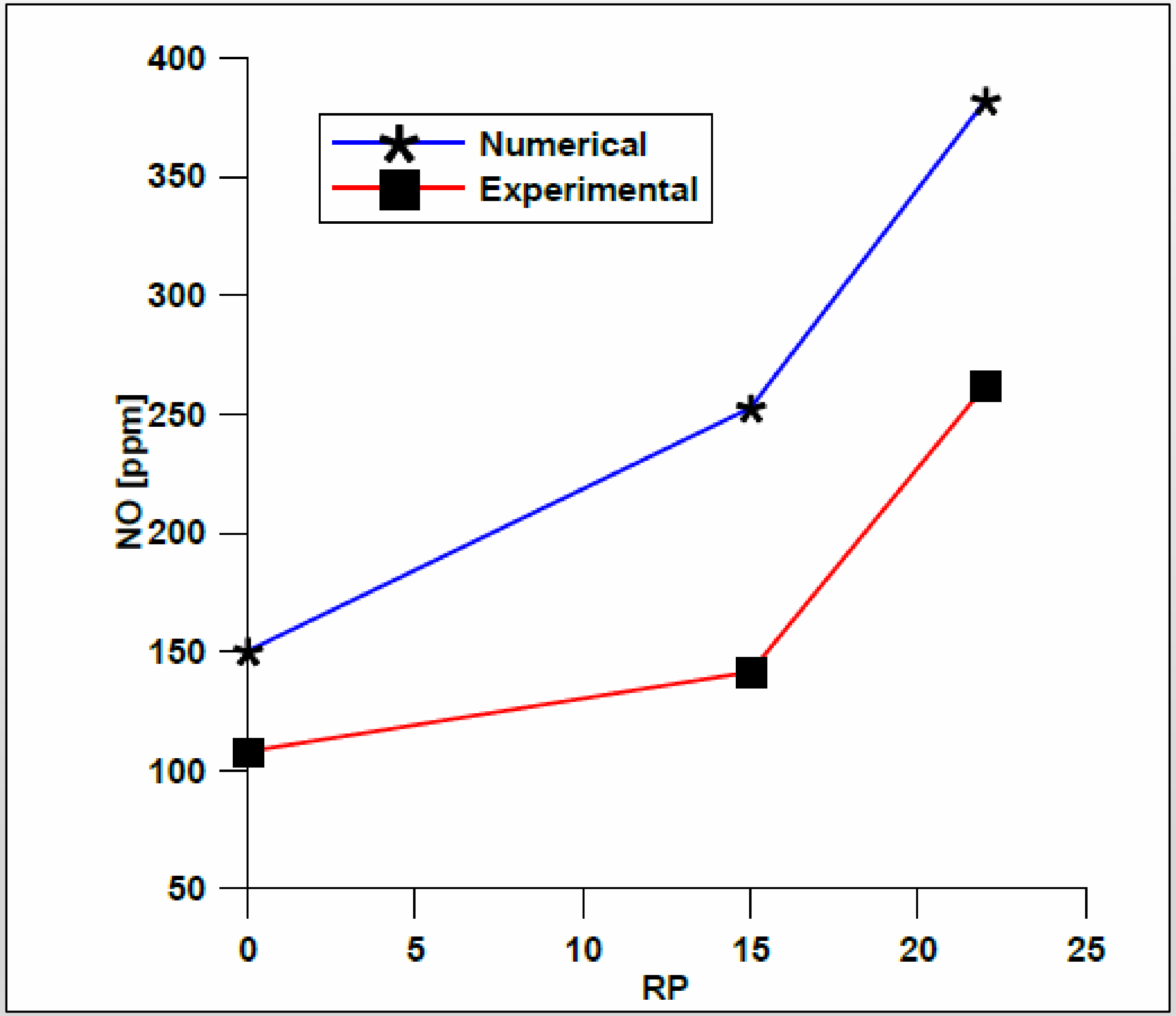
OH distributions comparison



The computed location of the regions with the highest OH contents is rather similar to the one detected by the chemiluminescence based analysis.



Nitric oxides



Increasing the natural gas amount the extent of high temperature regions becomes larger so promoting its formation, like predicted by the thermal NO formation mechanism.

A good accordance can be observed

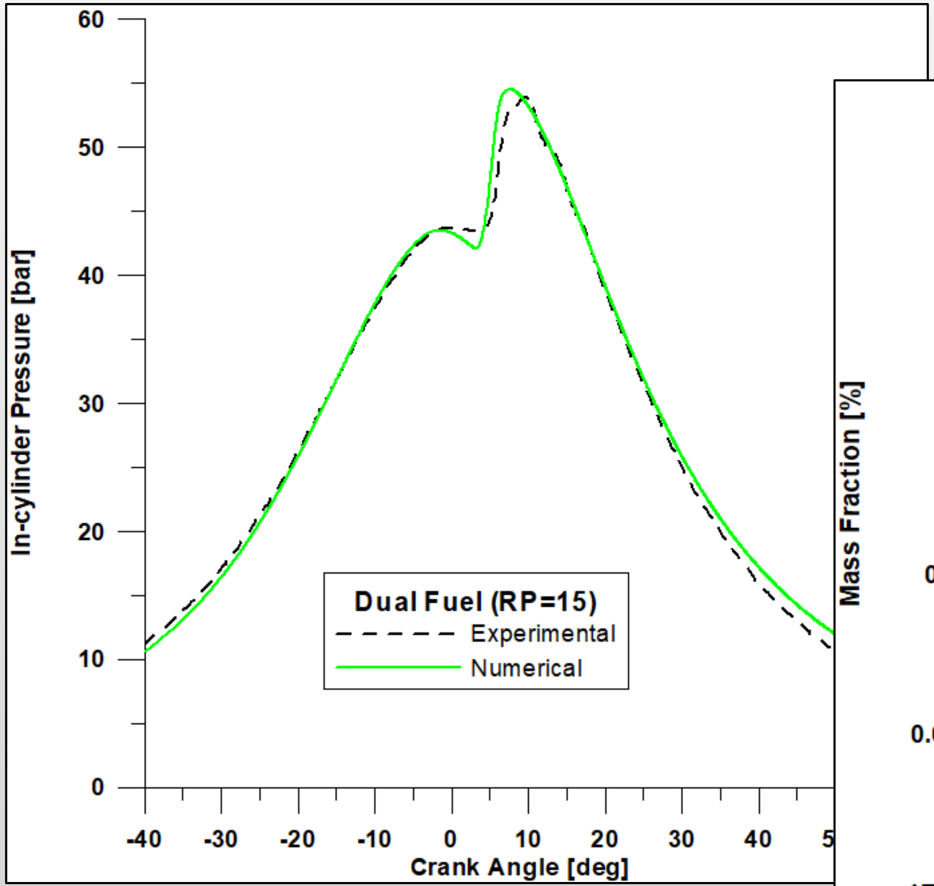
Conclusions

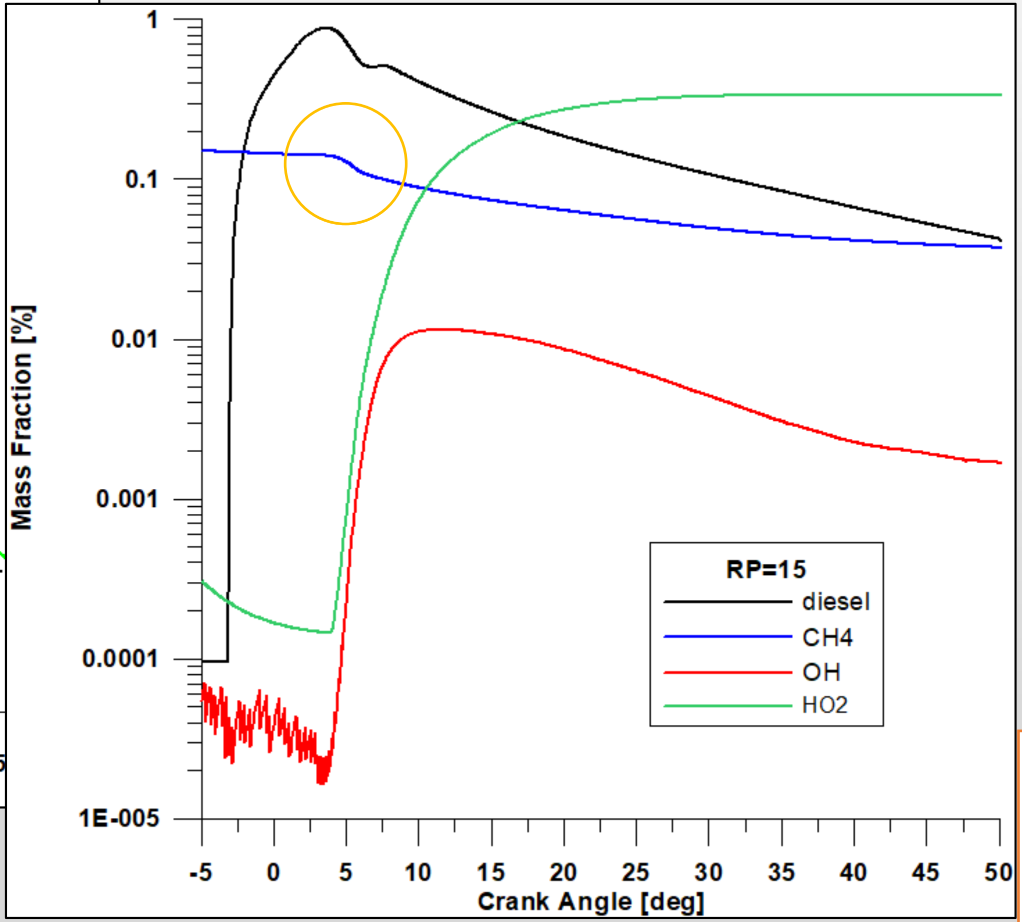
It should be reminded that the results discussed cannot represent a definitive response to the analysis of the dual fuel technology

- **Employment of more complex and realistic kinetic scheme**
- **Extension of the numerical and experimental activities to more usual engine operating conditions**
- **Extension of both experiments and computations to biodiesel-biogas dual fuel operation**

Future activities

New kinetic mechanism made of 9 reactions for the self- ignition delay of methane





$$CH_4 + O_2 \rightleftharpoons HO_2 + CH_3$$
$$CH_4 + HO_2 \rightleftharpoons H_2O_2 + CH_3$$
$$CH_4 + OH \rightleftharpoons H_2O + CH_3$$
$$CH_3 + O_2 \rightleftharpoons CH_2O + OH$$
$$CH_2O + OH \rightleftharpoons CHO + H_2O$$
$$CHO + O_2 \rightleftharpoons CO + HO_2$$
$$CHO + M \rightleftharpoons CO + H + M$$
$$H + O_2 + M \rightleftharpoons HO_2 + M$$
$$H_2O_2 + M \rightleftharpoons 2OH + M$$

It is possible to release from the empirical self-ignition delay correlation for methane

Thank you for your kind attention!

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