LES of stratified combustion in spray-guided direct injection engine

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Large-eddy simulation

- Favre-filtered Navier-Stokes equations

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial (\bar{\rho} \tilde{u}_j)}{\partial x_j} = 0$$

$$\frac{\partial \bar{\rho} \tilde{u}_i}{\partial t} + \frac{\partial (\bar{\rho} \tilde{u}_i \tilde{u}_j)}{\partial x_j} = \frac{\partial \bar{\tau}_{ij}}{\partial x_j} + \frac{\partial \tau_{ij}^{sgs}}{\partial x_j} - \frac{\partial \bar{p}}{\partial x_i} + \bar{\rho} g_i$$

$$\frac{\partial \bar{\rho} \tilde{h}}{\partial t} + \frac{\partial (\bar{\rho} \tilde{h} \tilde{u}_j)}{\partial x_j} + \frac{\partial \bar{\rho} k}{\partial t} + \frac{\partial \bar{\rho} k \tilde{u}_j}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\alpha_{eff} \frac{\partial \tilde{h}}{x_j} \right) + \frac{\partial (\bar{p} \tilde{h})}{\partial x_i}$$

- Smagorinsky model

$$\tau_{ij}^{sgs} = -2 \bar{\rho} \nu_T \left(\tilde{S}_{ij} - \frac{1}{3} \delta_{ij} \tilde{S}_{kk} \right)$$

$$\nu_T = C_s^2 \Delta^2 \sqrt{2 \tilde{S}_{ij} \tilde{S}_{ij}}$$

$$\tilde{S}_{ij} = \frac{1}{2} \left(\frac{\partial \tilde{u}_i}{\partial \tilde{x}_j} + \frac{\partial \tilde{u}_j}{\partial \tilde{x}_i} \right)$$

FSC combustion modeling

→ Flame propagation is modeled by a transport equation for the progress variable.
→ Flame wrinkling is described by an algebraic model.

$$\frac{\partial \tilde{c}\bar{\rho}}{\partial t} + \frac{\partial(\bar{\rho}\tilde{u}_j\tilde{c})}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\bar{\rho} D_t \frac{\partial c}{\partial x_j}\right) + \rho_u U_t |\nabla \tilde{c}|$$

$$U_t = A u' D_a^{1/4}\left[1 - \frac{\tau_L}{t_{fd}} + \frac{\tau_L}{t_{fd}}\exp\left(-\frac{t_{fd}}{\tau_L}\right)\right]^{1/2}$$

τ_L : Lagrangian time scale

t_{fd} : flame development time after spark discharge

D_a : Damköhler number

→ Transport equation for mixture fraction

$$\frac{\partial(\bar{\rho}\tilde{f})}{\partial t} + \frac{\partial(\bar{\rho}\tilde{u}_j\tilde{f})}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\bar{\rho} D_t \frac{\partial \tilde{f}}{\partial x_j}\right) + \overline{\rho S}$$

→ Transport equation for mixture fraction variance

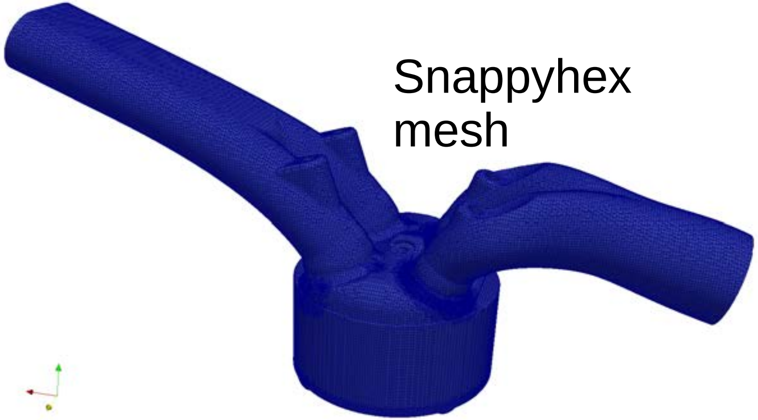
$$\frac{\partial(\bar{\rho}\widetilde{f''^2})}{\partial t} + \frac{\partial(\bar{\rho}\tilde{u}_j\widetilde{f''^2})}{\partial x_j} - \frac{\partial}{\partial x_j}\left(\bar{\rho} D_t \frac{\partial \widetilde{f''^2}}{\partial x_j}\right) = 2\bar{\rho} D_t \left|\frac{\partial^2 \tilde{f}}{\partial t^2}\right|^2 - \bar{\rho}\chi_f + (1-\tilde{c})\bar{\rho}\tilde{S}_v$$

Engine Modeling

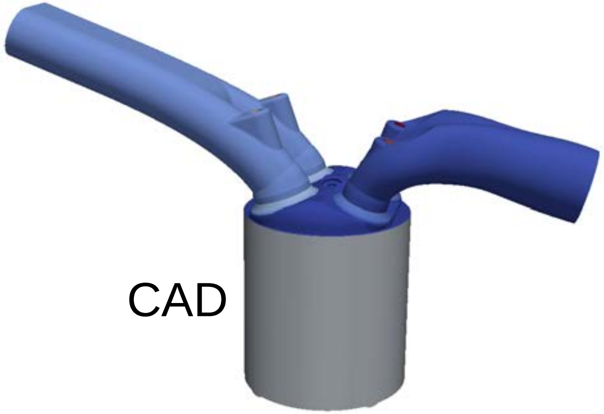
Moving mesh strategy

- Work flow

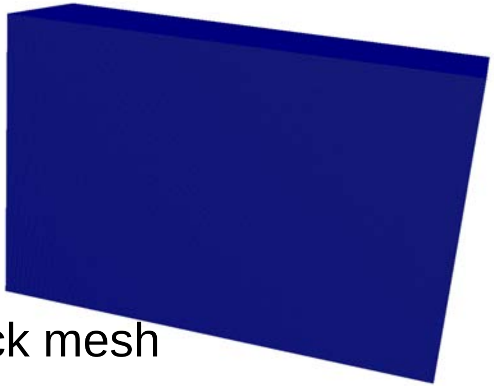
→ Whole process is automated with shell scripts
→ Works in parallel
→ Whole cycle (720 CAD)



Snappyhex mesh



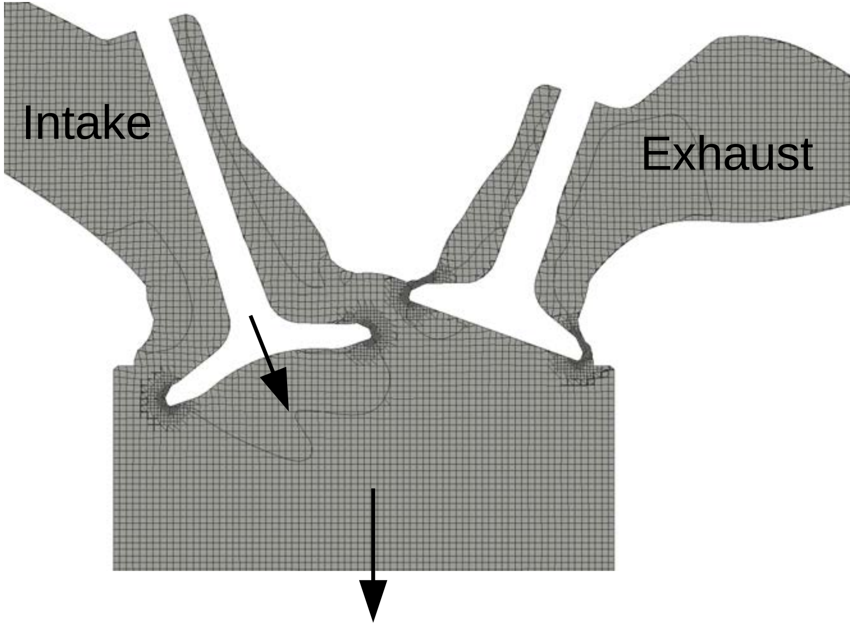
CAD



Block mesh

Moving mesh strategy

- OpenFOAM dynamic grid motion solver with topological changes
  - Keeps track of grid points positions
  - Calculates grid point velocities
  - Updates grid points positions
  - Mesh deformation dealt with mapping



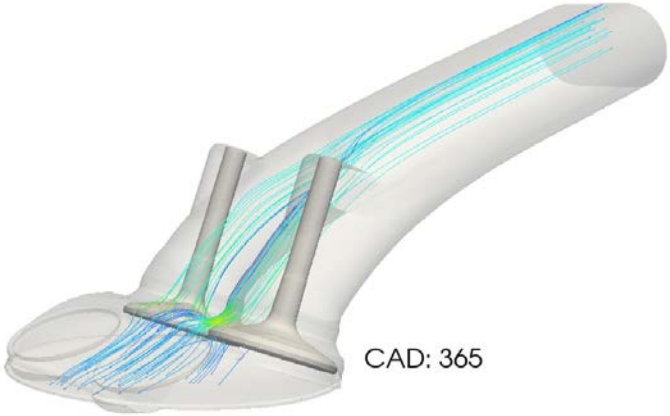
- Laplace equation for mesh motion (Jasak and Tukovic 2006)

$$\frac{\partial}{\partial x_j}\left(\gamma \frac{\partial \tilde{u}_{cell}}{\partial x_j}\right) = 0$$

Chalmers optical engine

Bore	83 mm
Stoke	90 mm
Compression ratio	10.2
Intake valve dia.	33 mm
Exhaust valve dia.	28 mm
IVO/IVC	340/600 CAD
EVO/EVC	105/365 CAD

Numerical setup



Grids: Unstructured grids with local mesh refinement

Meshing: Automated meshing + mapping

Mesh motion: Moving grids without topological changes

Grid resolution: ~1 mm, and < 0.05mm near valve

Flow solver: Compressible (pressure based)

Turbulence: Standard Smagorinsky model (Cs=0.2)

Spatial discretization: 2nd order CDS + TVD

Temporal discretization: 2nd order implicit backward

Pressure BC: Time varying pressure Bcs

Temperature BC: Iso-thermal walls

Results

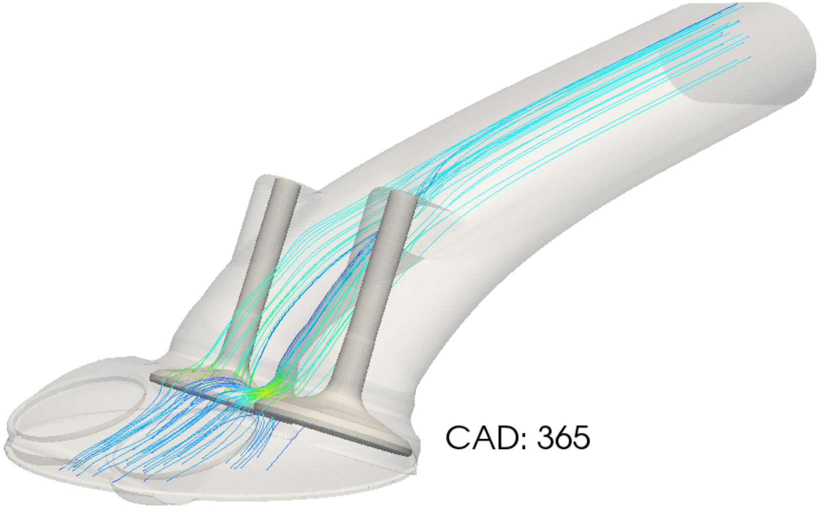
Work flow

1. Cold flow simulation
   to generate turbulent flow field

2. Reactive simulation
   stratified engine operation under different levels of stratification as specified in table

Engine point	Inj. time (°bTDC)	Ign. time (°bTDC)	Dur. (°CA)	λ	Imep (bar)
1	18	15	4	2.6	3.6
2	20	16	4	2.6	3.8
3	32	20	5	1.85	5

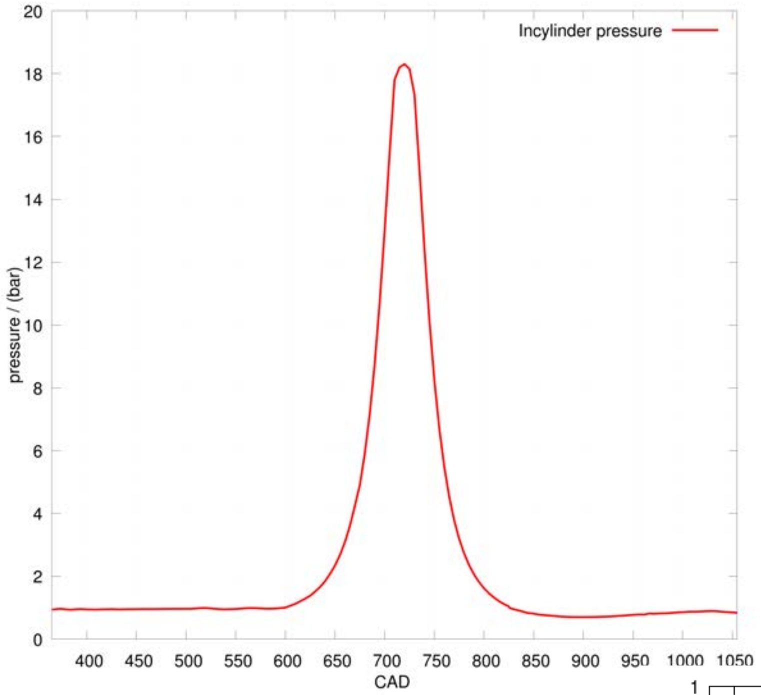
Cold flow
(streamlines)



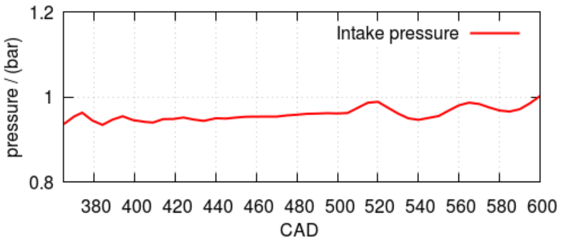
Results

Cold flow simulation

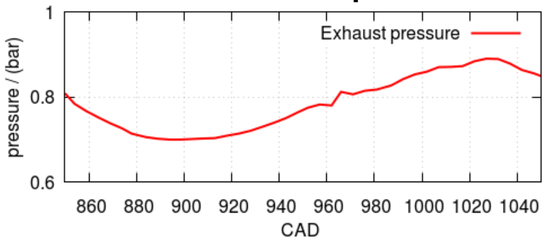
(pressure traces)



Intake pressure



Exhaust pressure



Results

Cold flow simulation

(flow fields)

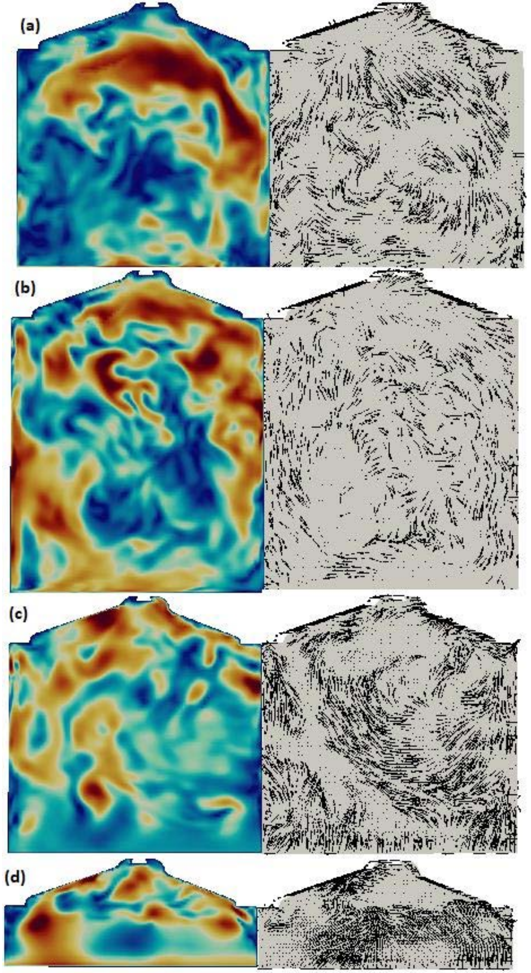
- The instantaneous flow fields are shown on cylinder symmetry plane during intake and compression stoke.
- Large turbulent structures formed during the intake stoke.
- Large turbulent structures are broken into small features.
- Smaller structures contribute more to flame wrinkling than large structure.

480 CAD

540 CAD

605 CAD

670 CAD



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Results

Cold flow simulation
(velocity)

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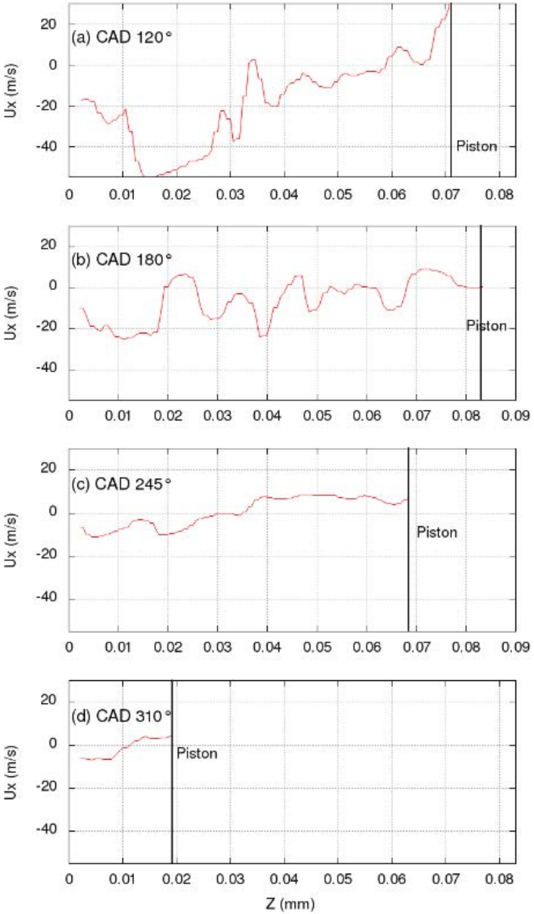

Results

Reactive simulation
(Injection and spark ignition)

Combustion mode: stratified

Injector: hollow cone

Flow solver: Compressible (pressure based)

Combustion model: FSC combustion model

Time discretization: 2^{nd} order backward

Numerical discretization: 2^{nd} order CDS + TVD

Heat loss: Isothermal wall

Results

Effect of turbulent mixture fraction fluctuations on pressure traces

Engine point	Inj. time (°bTDC)	Ign. time (°bTDC)	Dur. (°CA)	λ	Imep (bar)
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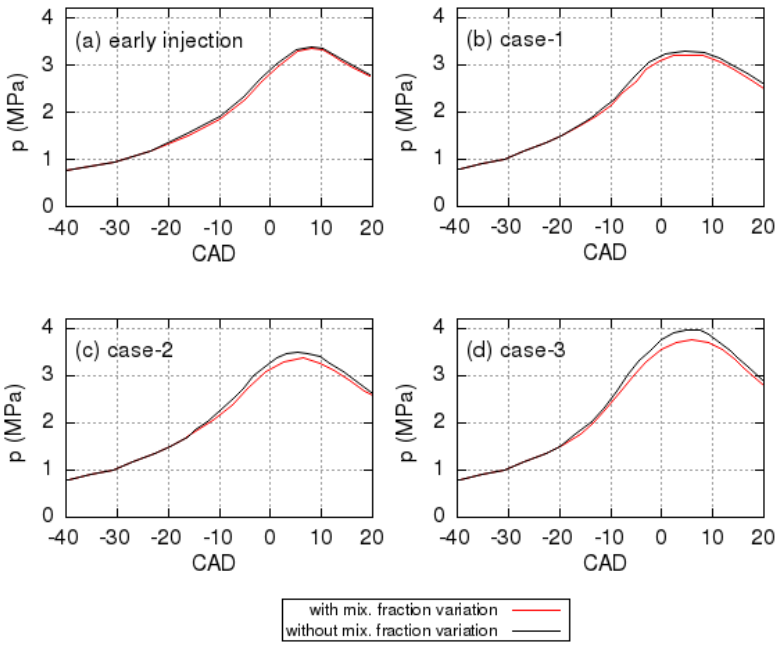

Early injection: inj.= 119°bTDC, ign=20°bTDC, λ=1.15

→ Early injection shows almost no dependency of mixture fraction variation because there is enough time for turbulent mixing to create a homogeneous mixture.

→ Late injection shows considerable variation in pressure traces.

→ Late injection shows different peak pressure values.

Results

Effect of turbulent mixture fraction fluctuations on pressure traces

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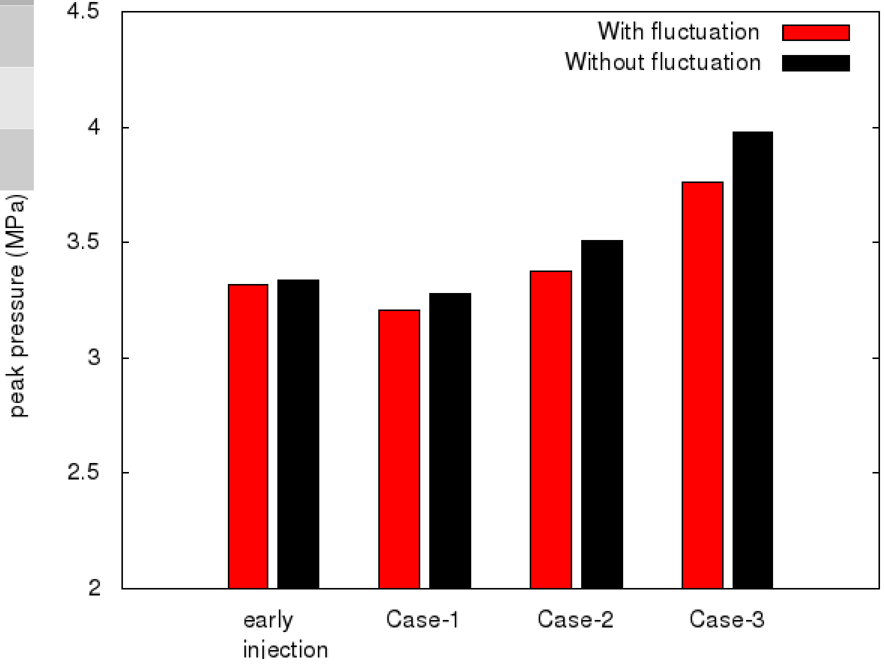

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Results

Comparison with measurements

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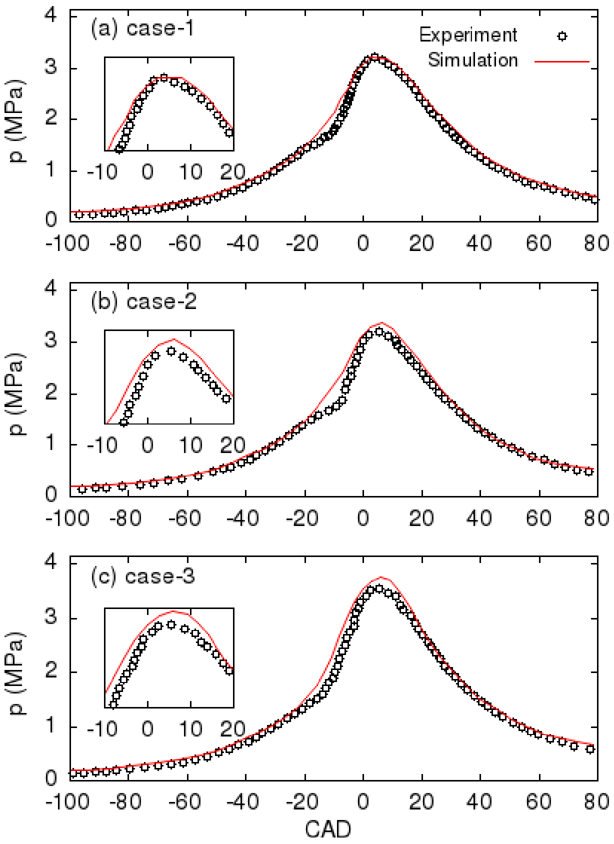
→ Case-1 show good agreement with measurement.

→ Case-2 and 3 over predict the peak pressure.

→ The ignition model is based on approximation of the initial flame kernel diameter.

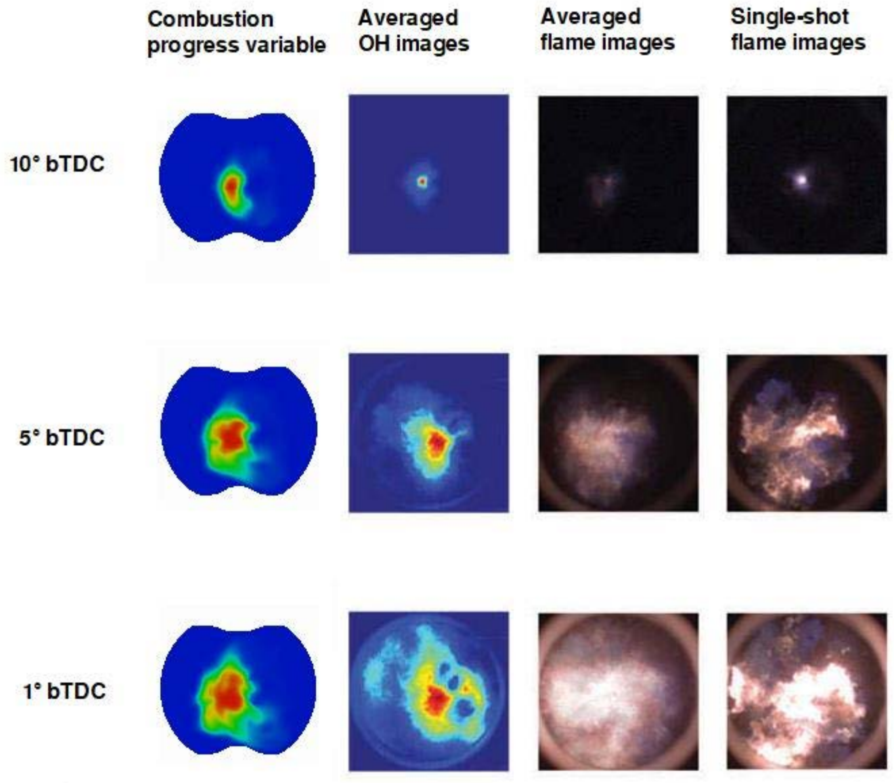
→ Wrong estimation of in-cylinder trapped mass which can only be measured through the flow-meter of test bench, which unfortunately does not exist.



Results

Flame propagation
(stratified combustion case-2)

- → At initial stage, flame zone is over-predicted.
- → Qualitatively good agreement with average OH images.
- → Strong dependence on turbulence level.
- → Need more cycles for the accurate prediction.



Summary

- Pressure variation for three diverse case of different loads were validated against experiment.

- Pressure rise with and without considering mixture fraction fluctuations shows considerable variation.

- Results revail that the mixture fraction variance for early fuel injection does not affect the burning rate.

- For late fuel injection, mixture fraction variance is relevant and significantly affects the burning rate.

- Computed pressure traces agree well for low load and late injection, but for late injection with high load pressure is slightly over-predicted.

Thank you
for your attention!

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