Internal Combustion Engines
I: Fundamentals and Performance Metrics

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Summer School on Combustion
Course Length: 9 hrs

(Mon.- Wed., June 25-27)

Short course outline:

Internal Combustion (IC) engine fundamentals and performance metrics, computer modeling supported by in-depth understanding of fundamental engine processes and detailed experiments in engine design optimization.

Day 1 (Engine fundamentals)

Hour 1: IC Engine Review, Thermodynamics and 0-D modeling
Hour 2: 1-D modeling, Charge Preparation
Hour 3: Engine Performance Metrics, 3-D flow modeling

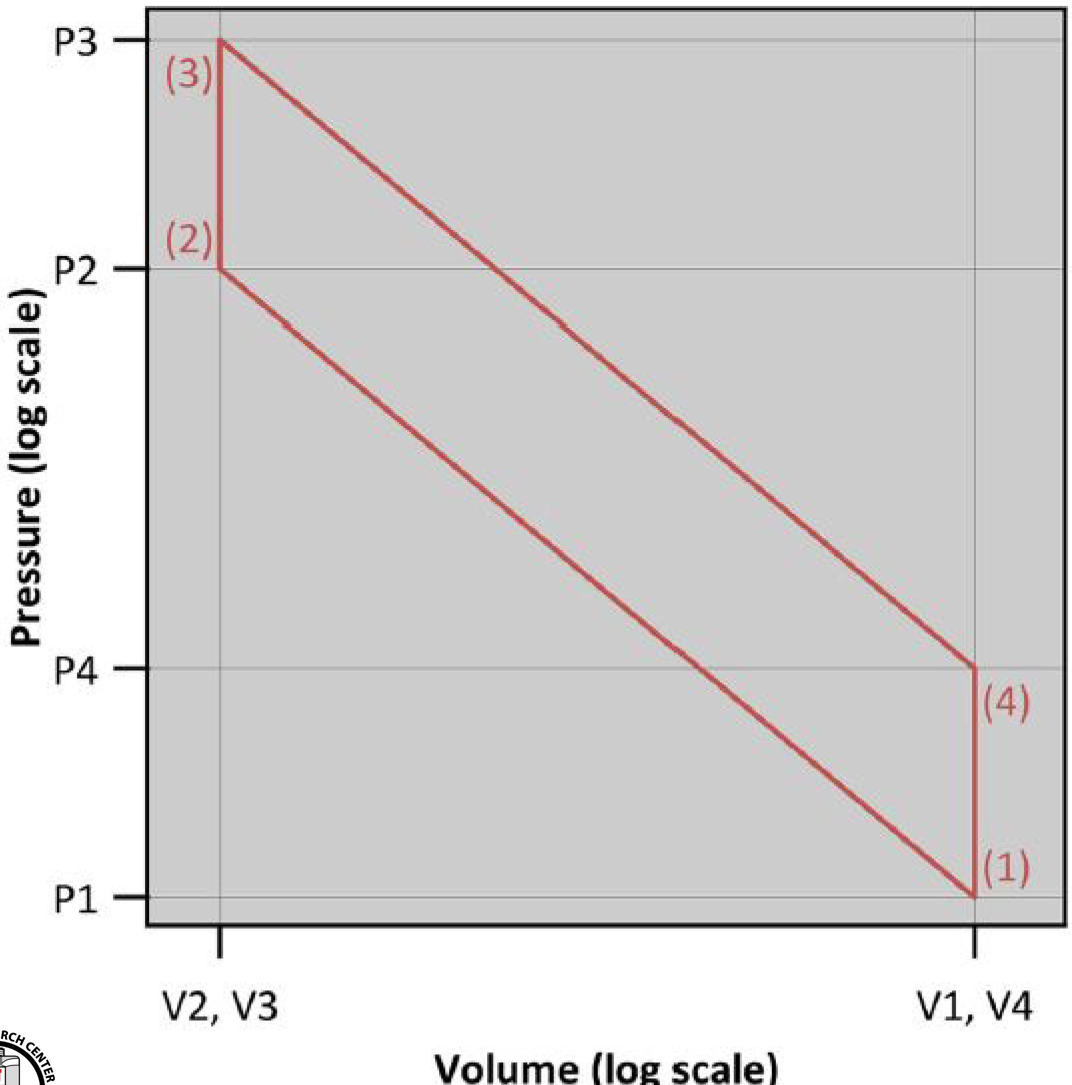
Day 2 (Computer modeling/engine processes)

Hour 4: Engine combustion physics and chemistry
Hour 5: Premixed Charge Spark-ignited engines
Hour 6: Spray modeling

Day 3 (Engine Applications and Optimization)

Hour 7: Heat transfer and Spray Combustion Research
Hour 8: Diesel Combustion modeling
Hour 9: Optimization and Low Temperature Combustion

Performance metrics: Ideal engine efficiency – Otto cycle



Maximum possible closed-cycle efficiency ("ideal efficiency")

State (1) to (2) isentropic (i.e., adiabatic and reversible) compression from max (V1) to min cylinder volume (V2)
Compression ratio rc = V1/V2.

State (2) to (3) adiabatic and isochoric (constant volume) combustion,
State (3) to (4) isentropic expansion.

State (4) to (1) exhaust process
- available energy is rejected
- can be converted to mechanical or electrical work:

Ideal engine efficiency – Otto cycle

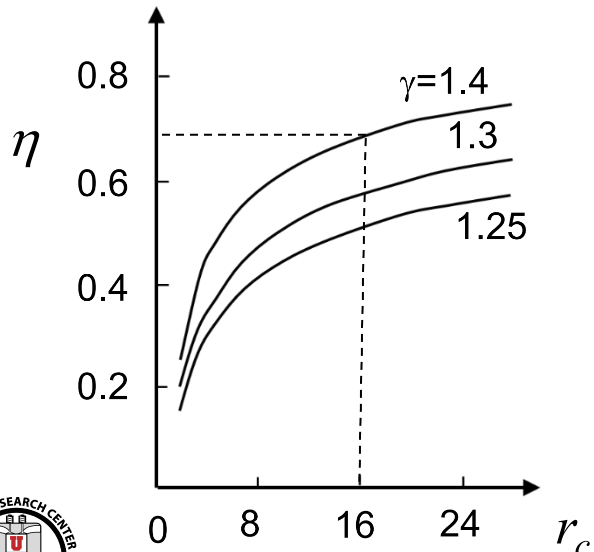
Efficiency = net work / energy supplied

$$\eta = [(T_3 - T_4) - (T_2 - T_1)]/(T_3 - T_2)$$

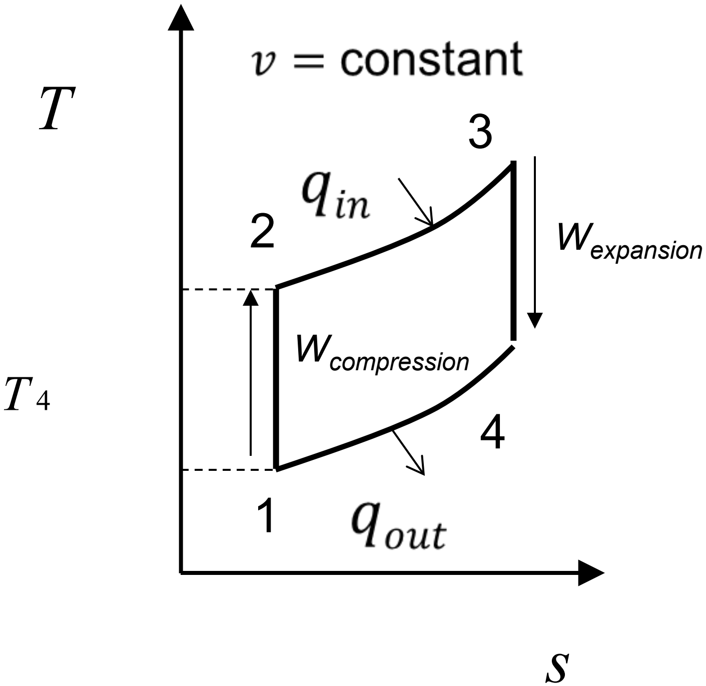
$$= 1 - (T_4 - T_1)/(T_3 - T_2)$$

However,

$$T_2/T_1 = (V_1/V_2)^{\gamma-1} = r_c^{\gamma-1} = (V_4/V_3)^{\gamma-1} = T_3/T_4$$



Otto



$$\boxed{\eta = 1 - 1/r_c^{\gamma-1}}$$

$$\eta_{ideal} = 1 - \frac{1}{r_c^{\gamma-1}}$$

η_{ideal} Function of only two variables, compression ratio (r_c) and ratio of specific heats (γ)

Increasing r_c increases operating volume for compression and expansion
Increasing γ increases pressure rise during combustion and increases work extraction during expansion stroke.

Both effects result in an increase in net system work for a given energy release and thereby increase engine efficiency.

<u>Actual closed-cycle efficiencies to deviate from ideal:</u>

1.) Assumption of isochoric (constant volume) combustion:
Finite duration combustion in realistic engines.
Kinetically controlled combustion has shorter combustion duration than diesel or SI
- duration limited by mechanical constraints, high pressure rise rates with audible engine noise and high mechanical stresses

2.) Assumption of calorically perfect fluid:
Specific heats decrease with increasing gas temperature; species conversion during combustion causes γ to decrease

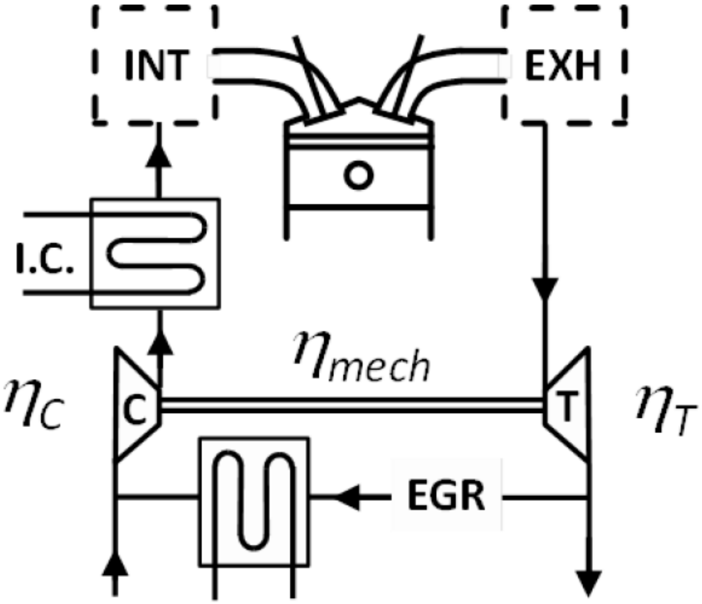
3.) Adiabatic assumption:
Large temperature gradient near walls results in energy being lost to heat transfer rather than being converted to crank work

Other assumptions:

In engine system models, compressors, supercharger, turbines modeled with constant isentropic efficiency instead of using performance map.

- typically, compressors, superchargers, and fixed geometry turbines have isentropic efficiencies of 0.7. VGT has isentropic efficiency of 0.65.

Charge coolers - intercooler, aftercooler, and EGR cooler modeled with zero pressure drop, a fixed effectiveness of 0.9, constant coolant temperature of 350 K.



Zero-dimensional closed-cycle analysis:

Combustion represented as energy addition to closed system

Fuel injection mass addition from user-specified start of injection crank angle (θ_{SOI}) and injection duration ($\Delta\theta_{\mathrm{inj}}$).

Pressure and mass integrated over the closed portion of cycle with specified initial conditions at IVC of pressure (p_0), temperature (T_0), and composition ($x_{n,0}$ for all species considered - N_2, O_2, Ar, CO_2, and H_2O) and initial trapped mass (m_0), including trapped residual mass

Post-combustion composition determined assuming complete combustion of delivered fuel mass.

Minor species resulting from dissociation during combustion not considered

Herold, 2011

First law energy balance: $de=dq - Pdv$

$$\left.\frac{dp}{d\theta}\right|_i = \left(\left.\frac{dQ_C}{d\theta}\right|_i - \left.\frac{dQ_{HT}}{d\theta}\right|_i - \frac{\gamma_i}{\gamma_i - 1} p_i \left.\frac{dV}{d\theta}\right|_i \right) \frac{\gamma_i - 1}{V_i}$$

Combustion:

$$\left.\frac{dQ_C}{d\theta}\right|_i = \frac{x_{b,i+1} - x_{b,i-1}}{\theta_{i+1} - \theta_{i-1}} \left(m_f LHV_f \right)$$

Combustion model - Wiebe function

$$x_{b,i} = 1 - \exp\left\{ - \left[\left(2.302^{\frac{1}{m_c+1}} - 0.105^{\frac{1}{m_c+1}} \right) \left(\frac{\theta_i - \theta_{SOC}}{\Delta\theta_{10-90}} \right) \right]^{m_c+1} \right\}$$

Wall heat transfer:

$$\left.\frac{dQ_{HT}}{d\theta}\right|_i = h_{c,i} \left[A_{IP,i} \left(T_i - T_{m,IP} \right) + A_{EP,i} \left(T_i - T_{m,EP} \right) + A_{l,i} \left(T_i - T_{m,l} \right) \right]$$

Heat transfer model - Woschni

$$h_{c,i} = 5 b^{m_{ht}-1} p_i^{m_{ht}} w_i^{m_{ht}} T_i^{0.75-1.62 m_{ht}}$$

$$w_i = 2.28 v_p + (3.25 \times 10^{-3}) \frac{V_d T_0}{p_0 V_{tr}} \left(p_i - p_{mot,i} \right)$$

Engine brake thermal efficiency BTE

BTE*LHV=IMEPg-PMEP-FMEP

DOE goal BTE=55%

Friction model

Chen-Flynn model (SAE 650733).

$$FMEP = C + (PF*P_{max}) + (MPSF*Speed_{mp}) + (MPSSF*Speed_{mp}^2)$$

where: C = constant part of FMEP (0.25 bar)

PF = Peak Cylinder Pressure Factor (0.005)

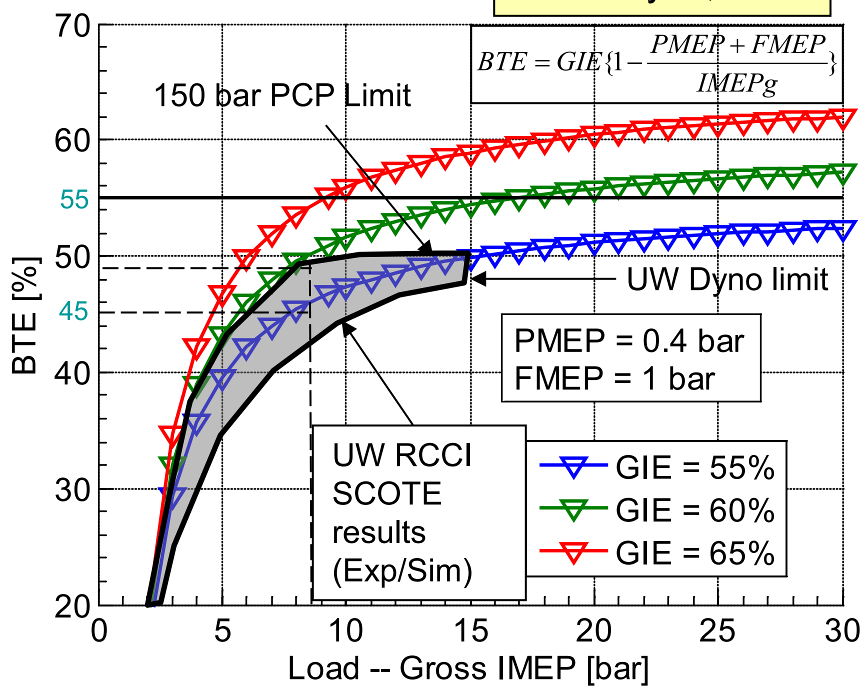
P_{max} = Maximum Cylinder Pressure

MPSF = Mean Piston Speed Factor (0.1)

MPSSF = Mean Piston Speed Squared Factor (0)

$Speed_{mp}$ = Mean Piston Speed

Chen-Flynn, 1965



1-D modeling for engine performance analysis

Table 1. Engine Specifications

Bore/Stroke	90 mm/100 mm
CR	12
Intake valves (2)	32.4 mm Diam/ 10.7 mm Lift
IVO (at 0 lift) IVC (at 0 lift)	-12°ATC gas exch. 60 to 224°ATC gas exch.
Exhaust valves (2)	26.1 mm Diam/ 10.7 mm Lift
EVO (at 0 lift) EVC (at 0 lift)	135°ATC firing 371°ATC firing

Mid load

Table 3. Operating conditions and parameters

RPM	2400 (U_P = 8 m/s)
Φ	0.2 – 1.2
EGR	0 – 80%
P_{EX}	1-3 (bar)
T_{IN}	333 K (60°C)
T_{ATM}	298 K (25°C)
T_{WALL} *(K)*	460 (head), 510 (pist), 390 (cyl)
T/C Eff (η_{OTC})	40, 50, 60%
Burn 10-90	25° CAD
CA50	10 ° ATC (~max eff.)

Lavoie, 2012

Table 2. Submodel specifications

Heat Transfer	Standard Woshni [27, 30]
Heat Release	Standard Wiebe [31]
Friction	Chen-Flynn [27, 32]
NOx model	2-zone Zeldovich [27, 31]

Woshni, 1967

Turbocharger equation

$$\left[1-\left(\frac{P_{ATM}}{P_{EX}}\right)^{\frac{\gamma_C-1}{\gamma_C}}\right]=\frac{\dot{m}_C C_{PC} T_{ATM}}{\dot{m}_T C_{PT} T_{EX}}\frac{1}{\eta_{OTC}}\left[\left(\frac{P_{IN}}{P_{ATM}}\right)^{\frac{\gamma_T-1}{\gamma_T}}-1\right]$$

$$\eta_{OTC}=\eta_T\eta_{MECH}\eta_C$$

Burn duration

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

Heat transfer

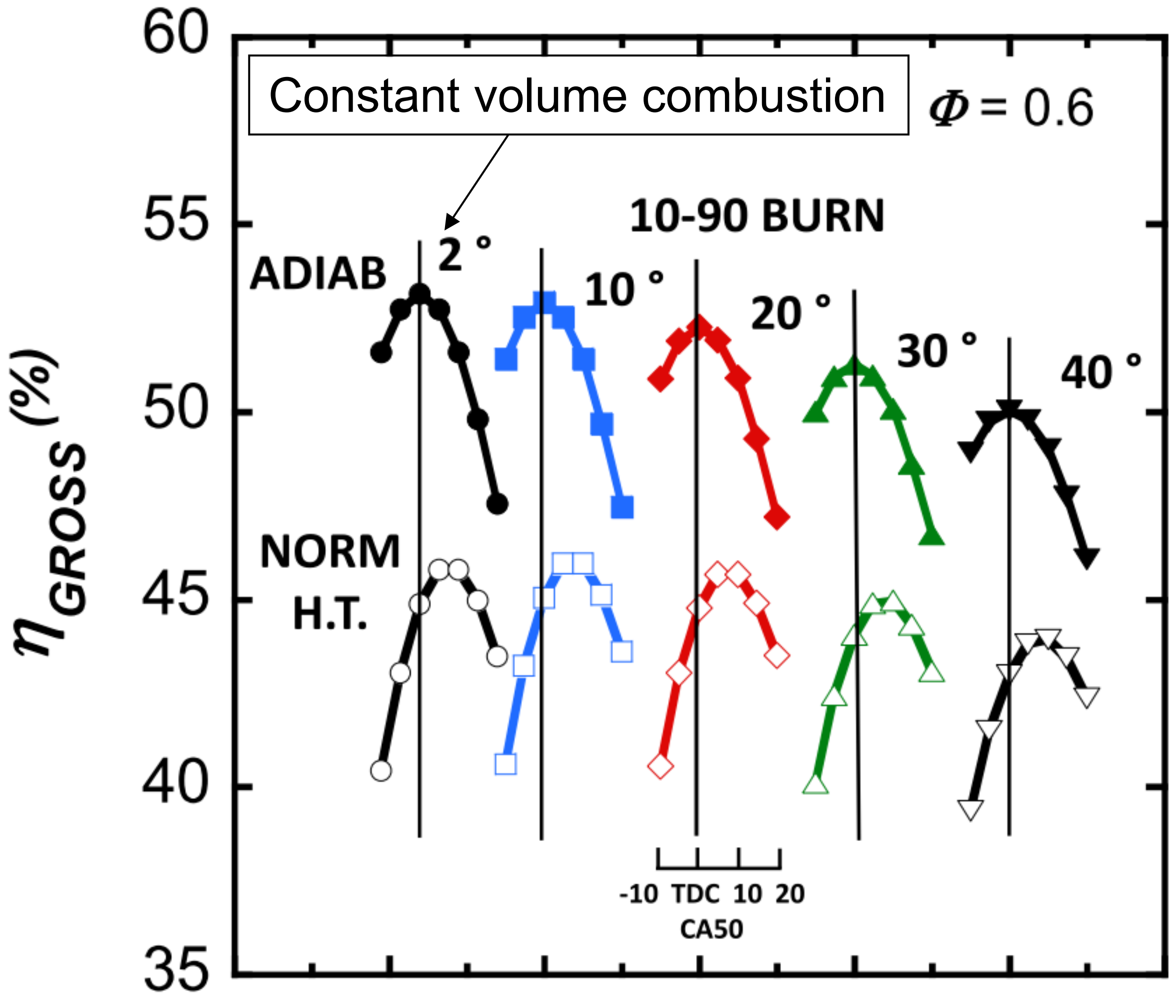
$$Nu\equiv\frac{hB}{k}\propto Re^m$$

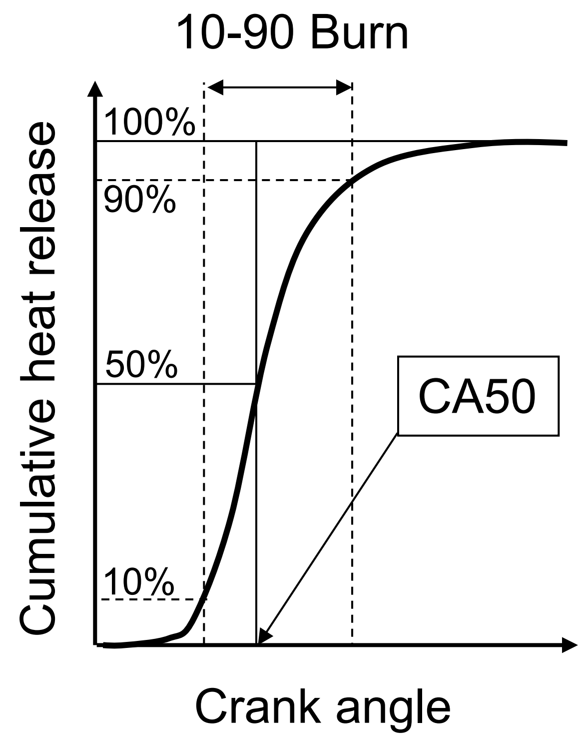
m~0.8, Re increases with Bore and ρ (boost)

Friction

$$FMEP\ (bar)=\ 0.4+0.005P_{MAX}+0.09U_P+\ 0.0009U_P^2$$

Effect of combustion phasing on efficiency

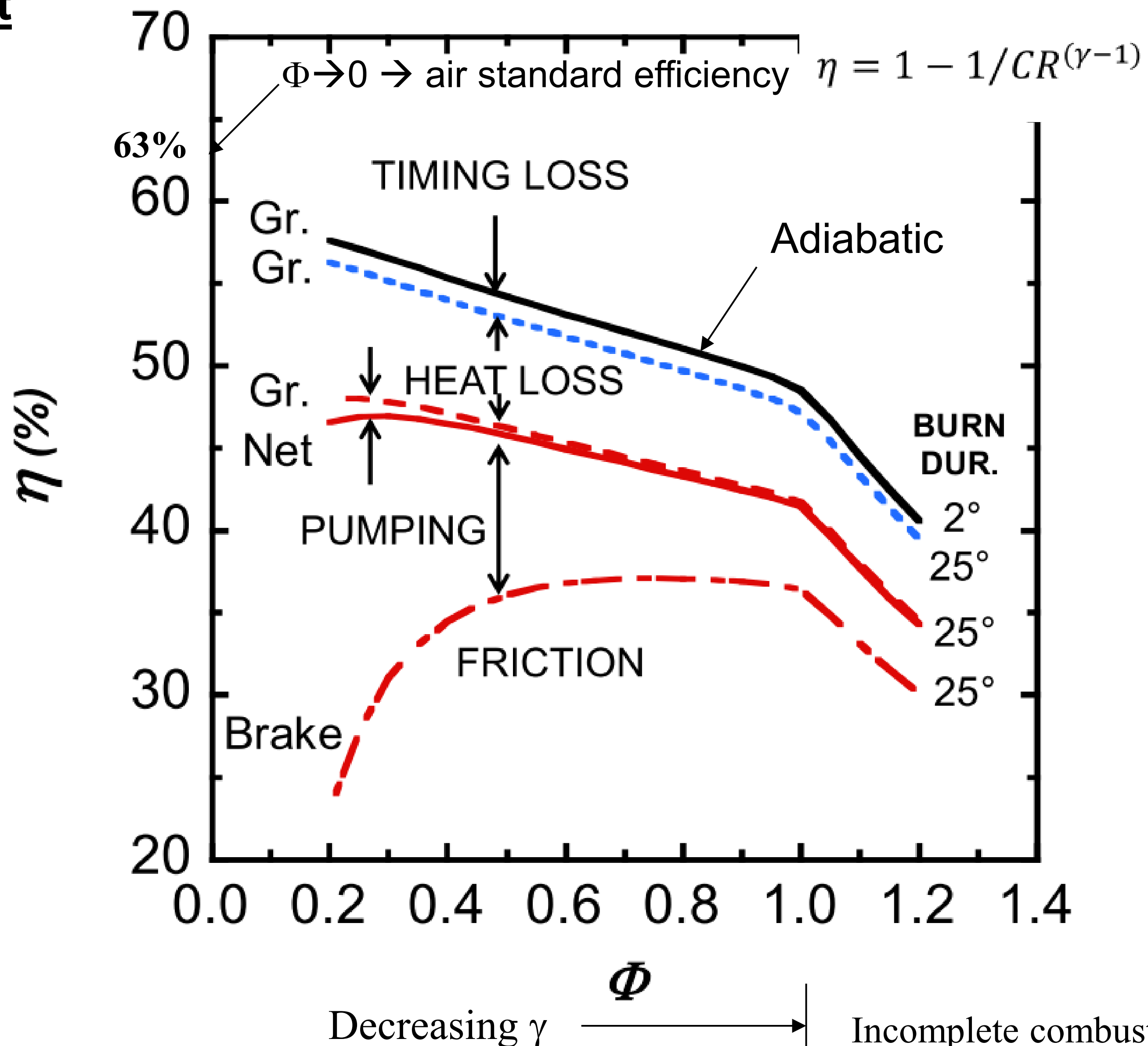




Without HT: Best efficiency CA50~TDC

With HT: best efficiency with CA50~10 deg – tradeoff between heat loss/late expansion

Energy budget

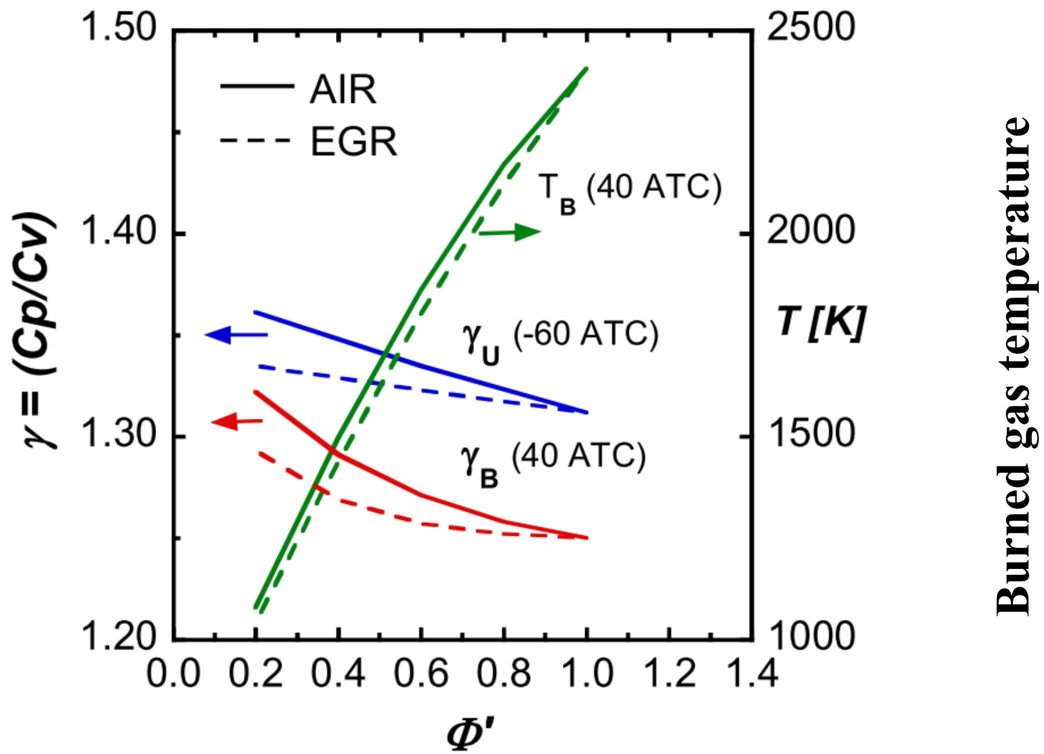


Effect of dilution

Fuel-to-charge equivalence ratio, ϕ'

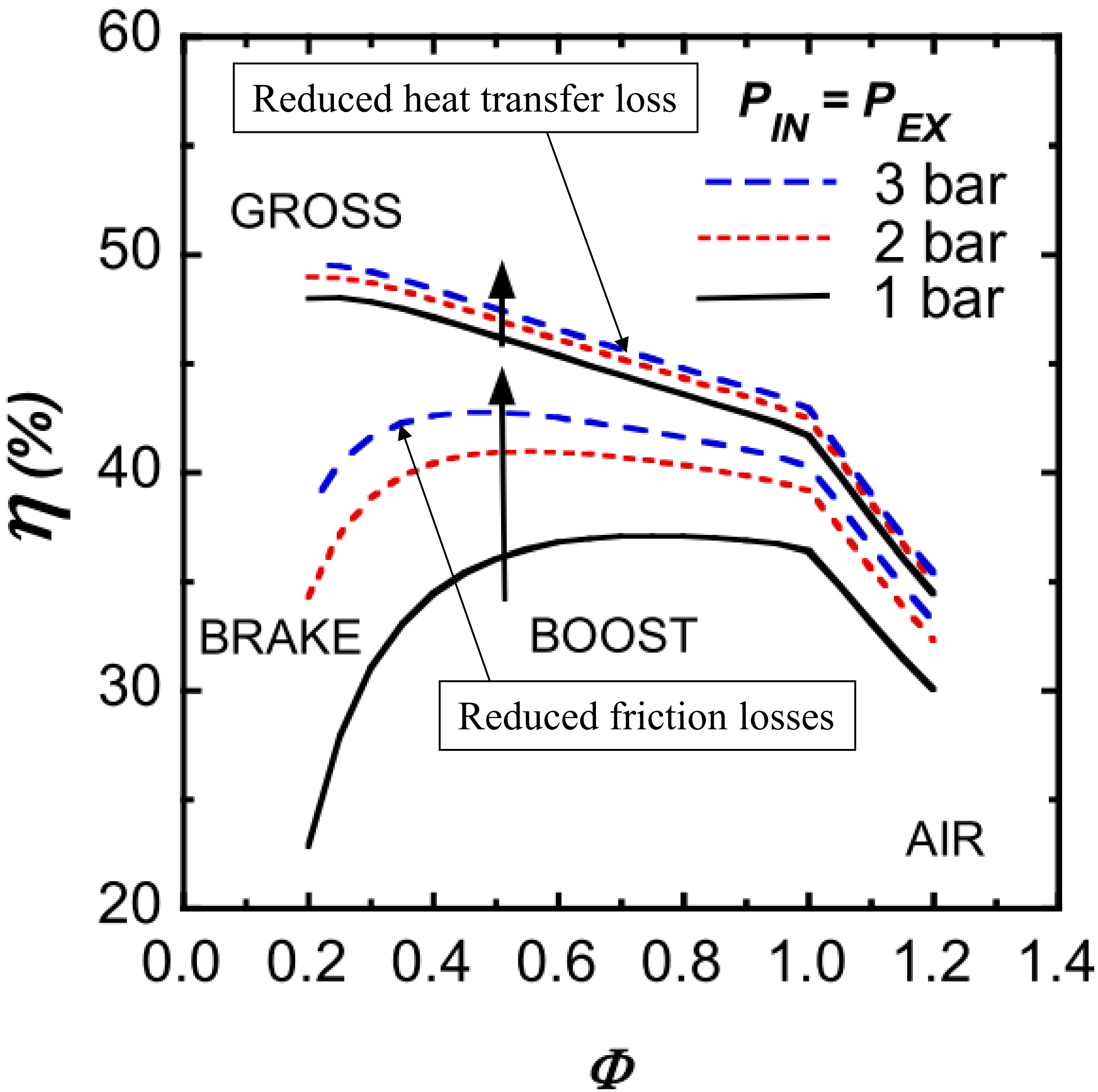
$$\Phi' \equiv \frac{F/(A+R)}{(F/A)_{ST}} = \frac{\Phi(1-RGF)}{[1+\Phi \cdot RGF \cdot (F/A)_{ST}]} \cong \Phi(1-RGF)$$

where F, A, and R denote mass of fuel, air, and residual gas, RGF is the total residual gas fraction

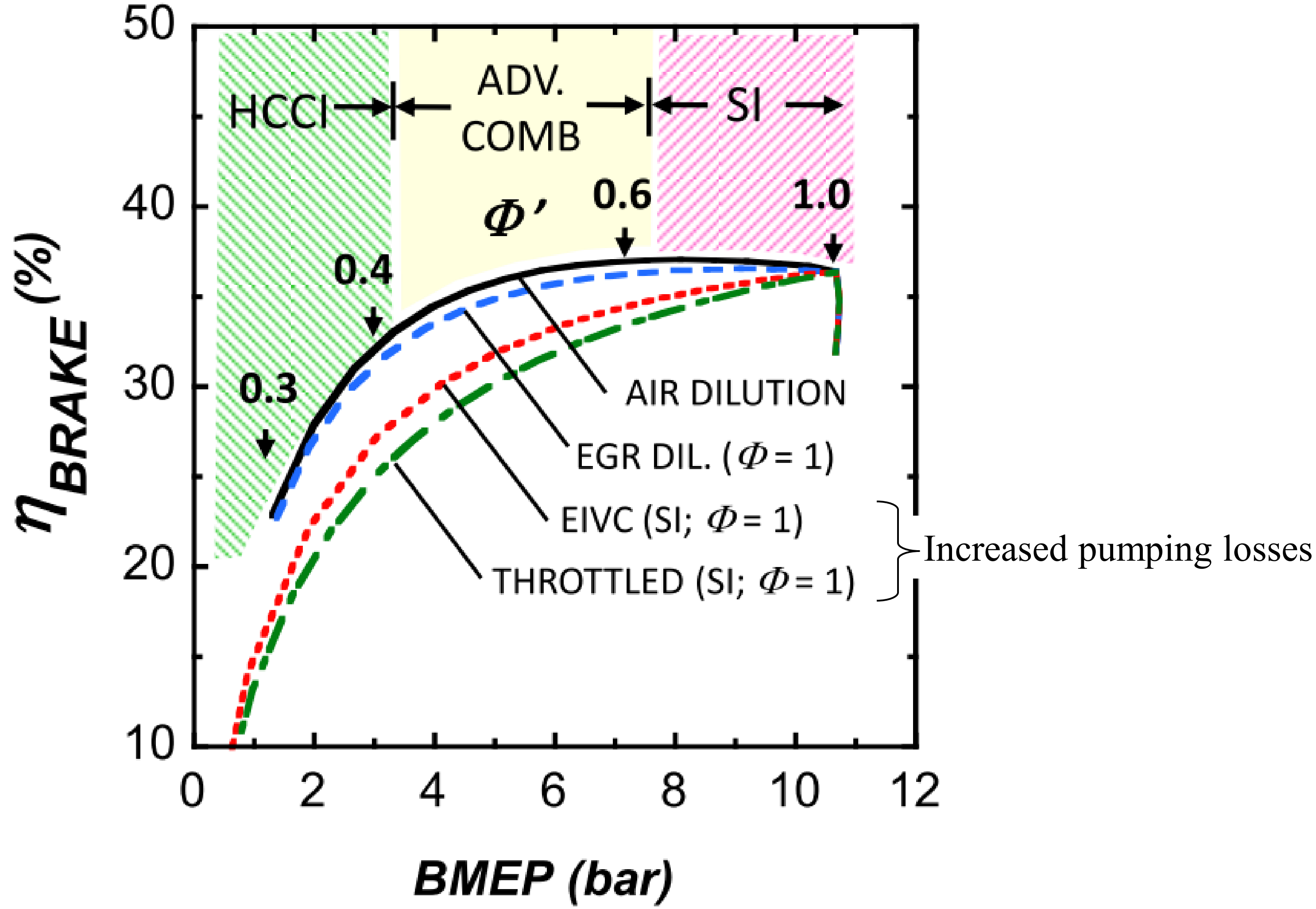


ϕ ranges from 0.2 to 1 with air, EGR ranges from 0 to 80% with ϕ=1

Effect of boost pressure on efficiency

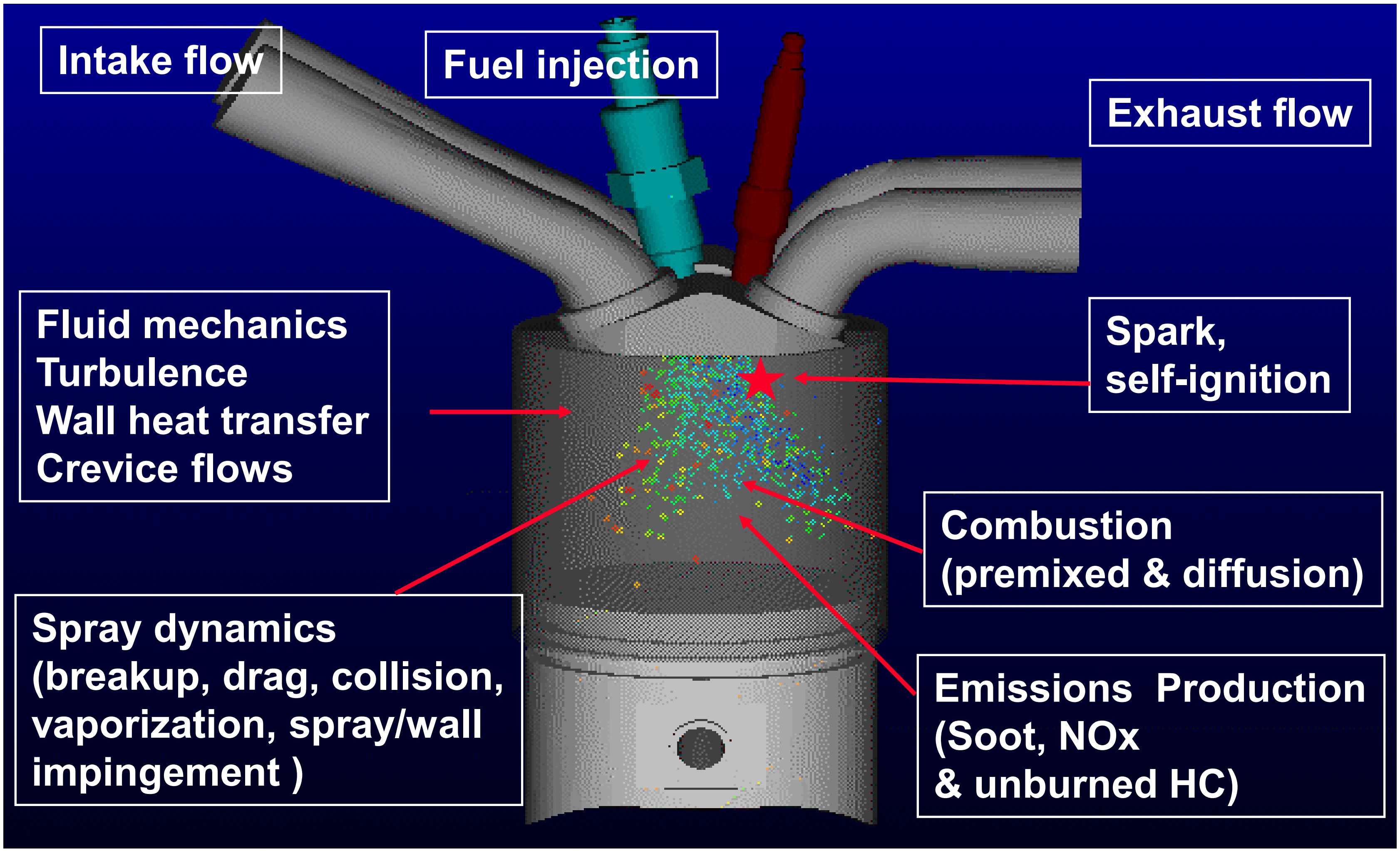


Potential brake efficiencies of naturally aspirated engines



3-D Modeling of Engine processes

Unsteady, turbulent, 2-phase reactive flows



Brief history of engine CFD

Arab oil crisis ~ 1973: US DOE

- Open source codes
  - Los Alamos National Lab, Princeton Univ., UW-ERC
  - 1970's – RICE → REC → APACHE → CONCHAS
  - 1980's – CONCHAS-SPRAY → KIVA family
  - 1985 – KIVA ;1989 – KIVA-II; 1993 – KIVA-3;
  - 1997 – KIVA-3V; 1999 – KIVA-3V Release 2; 2006 - KIVA-4
  - 2004 – OpenFOAM (2011 SGI)
- Commercial codes
  - 1980's Imperial College & others
  - Computational Dynamics, Ltd. → commercialize: STAR-CD
  - 1990's—other commercial codes: AVL FIRE, Ricardo VECTIS
  - 2005– FLUENT (with moving piston and in-cylinder models)
  - 2010 – CONVERGE (CSI), FORTE (ANSYS)
  - 2018 – FRESCO (WERC)…..

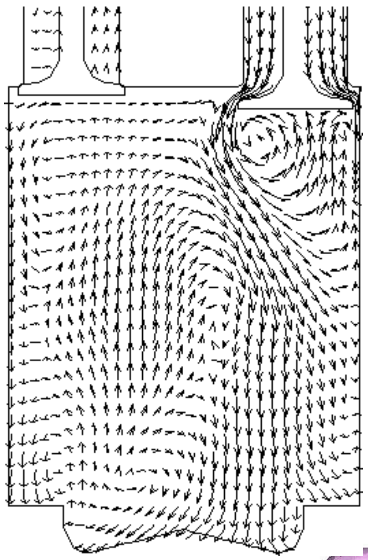
Annual IMEM-User group meeting: UW-ERC/MTU

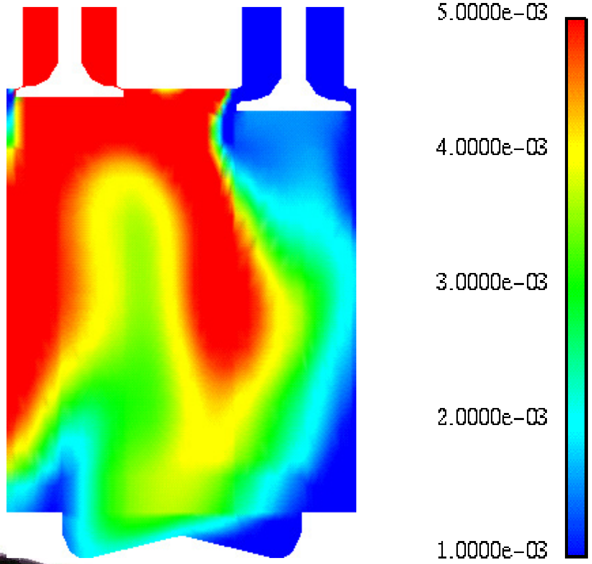
SAE Multidimensional Modeling Sessions, ASME…..

CFD Prediction of volumetric efficiency

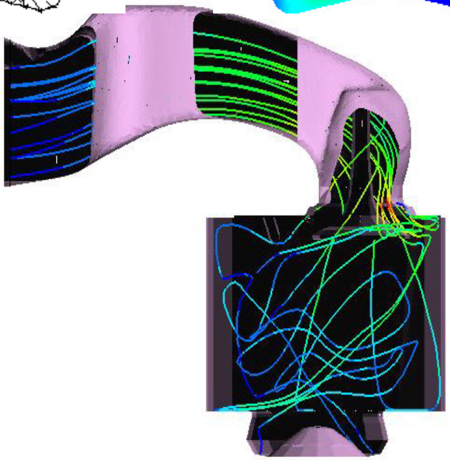
CFD flow velocity and residual gas distribution during gas exchange in plane of valves (intake valves about to close 144 degrees ATDC - 1600 rev/min)

Accurate descriptions of valve flow losses require consideration of multi-dimensional flow separation phenomena and their effect initial conditions at intake valve closure (IVC)





Highest mixing of incoming fresh charge and combustion products occurs when intake flow velocities are largest due to high flow turbulence (half-way through Intake stroke)

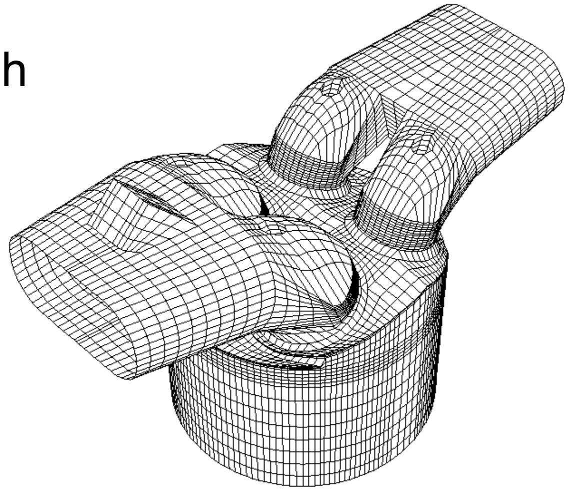


Intake-flow-generated swirl and tumble flows greatly affect flow mixing

Amsden, 1989, 1997

3-D CFD model equations

Solve conservation equations on (moving) numerical mesh



Mass

spray source terms

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = \dot{\rho}^s$$

Species

$$\frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m \mathbf{u}) = \nabla \cdot \left[\rho D \nabla \left(\frac{\rho_m}{\rho} \right) \right] + \dot{\rho}_m^c + \dot{\rho}_m^s$$

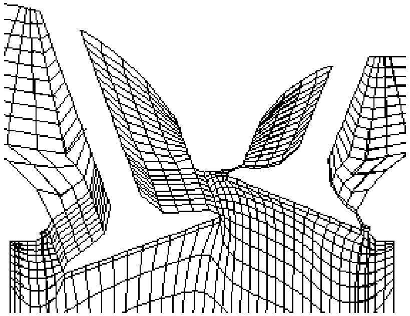
Momentum

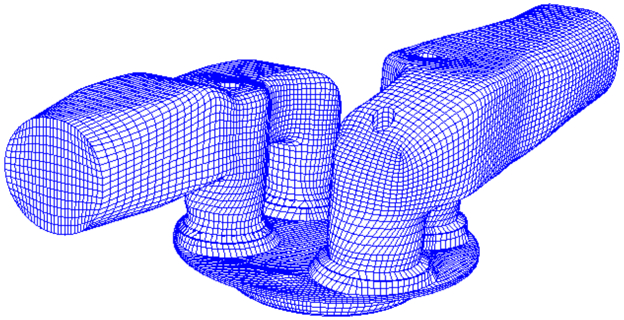
combustion source terms

$$\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u}\mathbf{u}) = \rho \mathbf{g} + \mathbf{F}^s - \nabla p + \nabla \cdot \bar{\sigma}$$

Energy

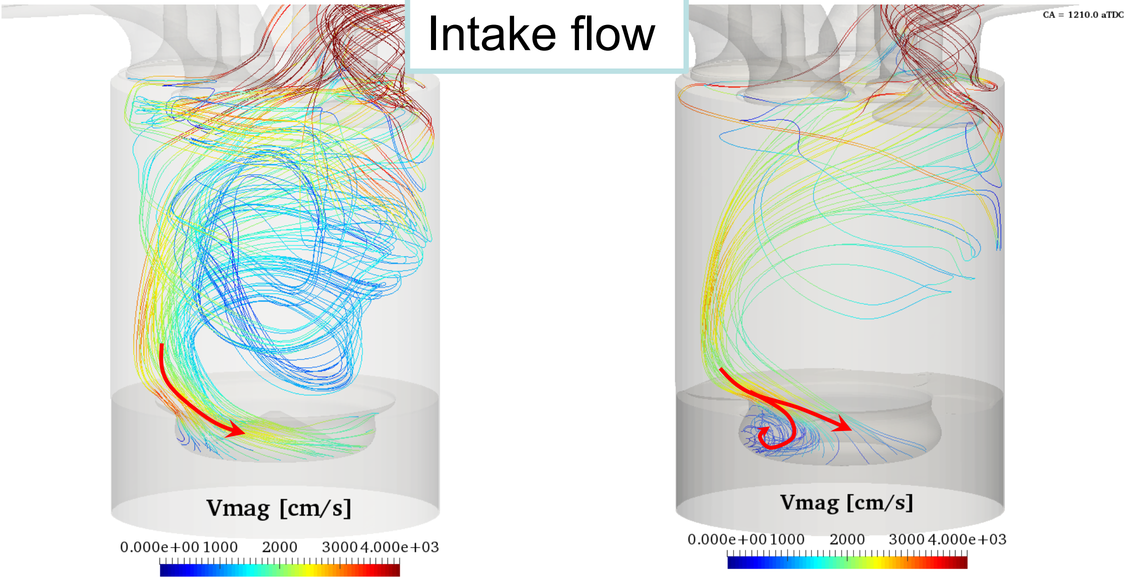
$$\frac{\partial (\rho I)}{\partial t} + \nabla \cdot (\rho \mathbf{u} I) = -\nabla \cdot \mathbf{J} + \dot{Q}^c + \dot{Q}^s - p \nabla \cdot \mathbf{u} + \bar{\sigma} : \nabla \mathbf{u}$$



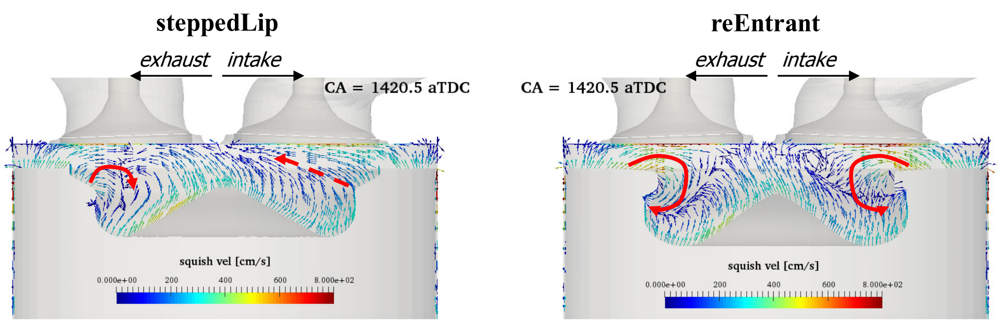


Turbulence Modeling - generation mechanisms

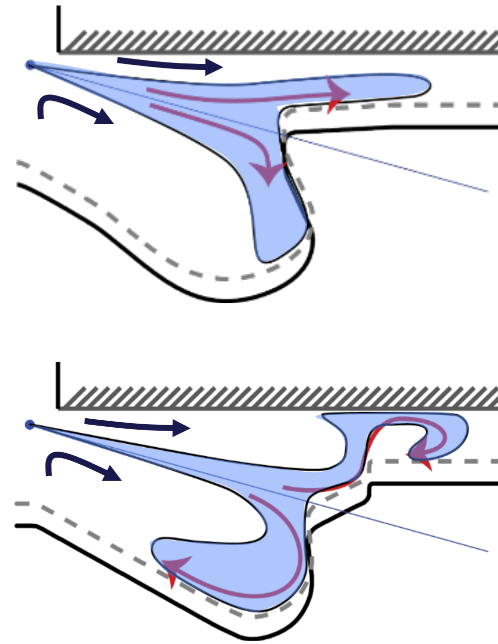
Intake flow



Squish-swirl interaction



Gas jets



Entrainment

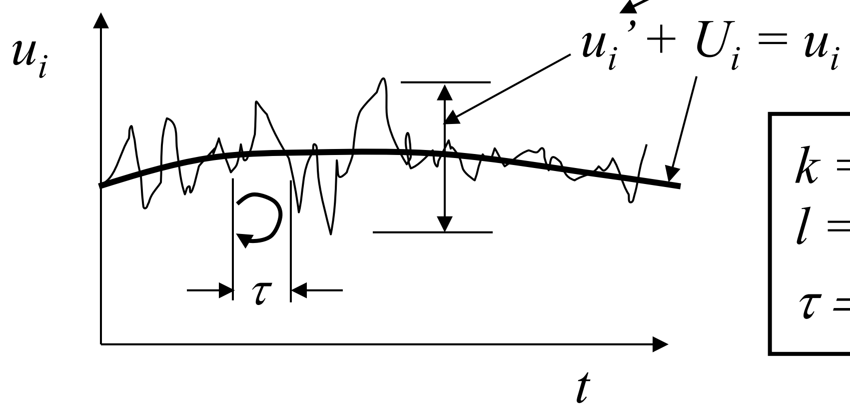
Jet-wall interaction

Turbulence modeling

Rutland, 2011, Wang, 2013, Perini, 2017

(RANS - RNG k-ε)

Sub-grid velocity



$k = 3u_i^2/2$

$l = U_i\ \tau$

$\tau = k/\varepsilon$

RANS – Reynolds Averaged Navier Stokes

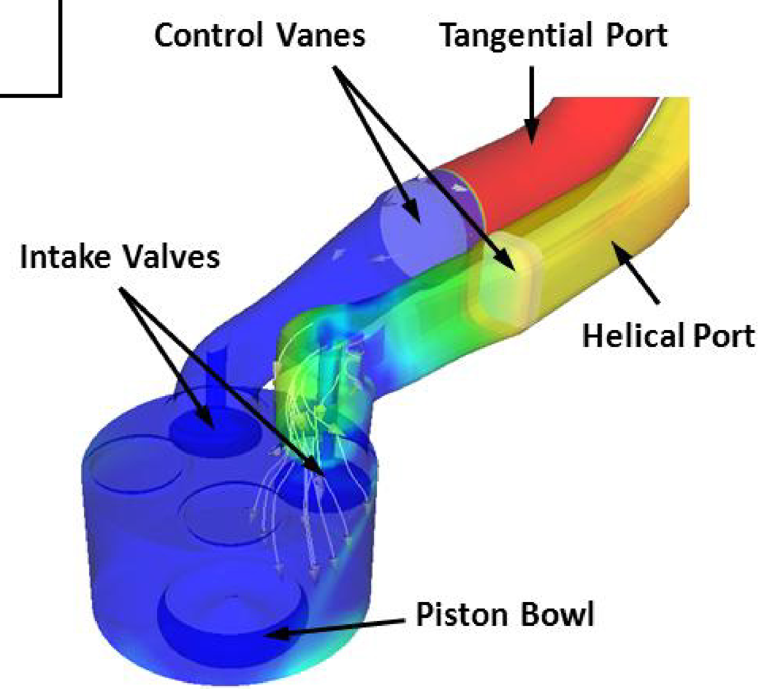
LES – Large Eddy Simulation

LES Filtered flow

- Smagorinsky
- Dynamic Smagorinsky

$$\nu_T = (C_S \Delta)^2 |\tilde{S}|$$

- Dynamic Structure
- One-Equation Eddy Viscosity



Perini, 2017

k-epsilon Turbulence Model

The most widely adopted class of 2-equation isotropic models

Transport equations for turbulence kinetic energy and dissipation rate

$$\frac{\partial(\rho k)}{\partial t} + \mathbf{u}\cdot\nabla(\rho k) = P - \rho\varepsilon + \nabla\cdot(\alpha_k \mu_t \nabla k) + P_{spray}$$

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \mathbf{u}\cdot\nabla(\rho\varepsilon) = \frac{\varepsilon}{k}(C_1 P - C_2\varepsilon) - \rho R + C_3\rho\varepsilon\nabla\cdot\mathbf{u} + \nabla\cdot(\alpha_\varepsilon \mu_t \nabla\varepsilon) + c_s P_{spray}$$

Boussinesq assumption: linear stress-strain closure

Mean flow strain rate

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

→

Reynolds stress

$$\boldsymbol{\tau} = 2\mu_t \mathbf{S} - \frac{2}{3}\rho k \delta_{ij}$$

→

tke production

$$P = -\boldsymbol{\tau} : \nabla\mathbf{u}$$

Han, 1995, Perini, 2017

ReNormalization Group closure

Result of a coarse-graining procedure: energy-invariant iterative wavelength-filtering from the Kolmogorov up to integral scale

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\mathbf{u}\cdot\nabla(\rho\varepsilon)=\frac{\varepsilon}{k}(C_1P-C_2\varepsilon)\boxed{-\rho R}\boxed{+C_3\rho\varepsilon\nabla\cdot\mathbf{u}}+\nabla\cdot(\alpha_\varepsilon\mu_t\nabla\varepsilon)+c_sP_{spray}$$

$$R=\frac{C_\mu\eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}\frac{\varepsilon^2}{k},\qquad \eta=\sqrt{2}\|\mathbf{S}\|_F\frac{k}{\varepsilon},$$

- scale-invariant *renormalized* dissipation source R
- Based on turbulent-to-mean-strain time scales

$$C_3=\frac{-4+2C_1}{3}+\frac{\partial\nu_0}{\partial t}\frac{1}{\nu_0(\nabla\cdot\mathbf{u})}-\frac{\sqrt{6}C_\mu}{3\beta\eta_0}\operatorname{sgn}(\nabla\cdot\mathbf{u})$$

$$(\nabla\cdot\mathbf{u})\,k/\epsilon\ \gg 1$$
$$C_\eta\eta\to\ -1/(\beta\eta_0)$$

Compressibility (Han and Reitz)
Rapid distortion limit assumption:

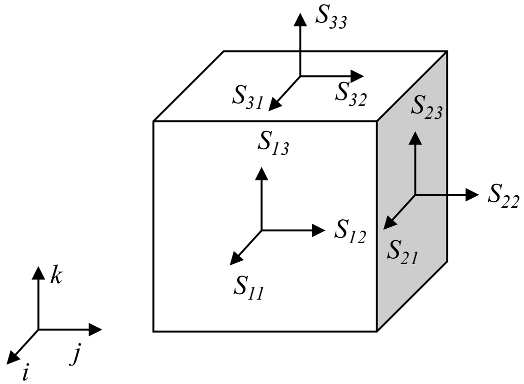
→ C3 is a constant:

$$C_3=\begin{cases}1.726, & \nabla\cdot\mathbf{u}<0;\\ -0.90, & \nabla\cdot\mathbf{u}>0.\end{cases}$$

Generalized RNG k-epsilon

Wang, 2011

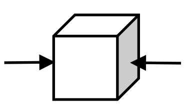
Incorporate effects of anisotropy from the strain rate tensor via *augmented* isotropic dissipation coefficients



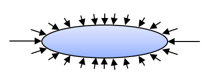
$$\begin{cases} a = 3\left(S_{11}^2 + S_{22}^2 + S_{33}^2\right) / \left(|S_{11}| + |S_{22}| + |S_{33}|\right)^2 - 1 \\ n = 3 - \sqrt{2a} \end{cases}$$

$$\begin{cases} C_{2,G} = b_0 + b_1 n + b_2 n^2, \\ C_{3,G} = -\dfrac{n+1}{n} + \dfrac{2}{3} C_1 + \sqrt{\dfrac{2+2a}{3}} C_\mu C_\eta \eta \,\mathrm{sgn}(\nabla \cdot \mathbf{u}) \end{cases}$$

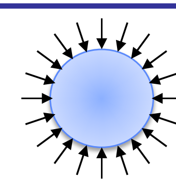
- ✓ Like a von-Mises-type measure of **effective strain**
- ✓ **a** = Weight of the normal components (compression/expansion)
- ✓ **n** = Effective flow ‘dimensionality’



Unidirectional
a = 2
n = 1



Plane 2D (radial)
a = .5, n = 2

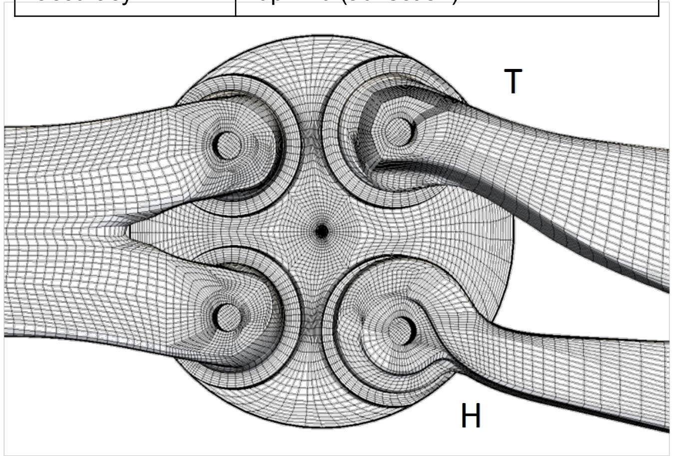


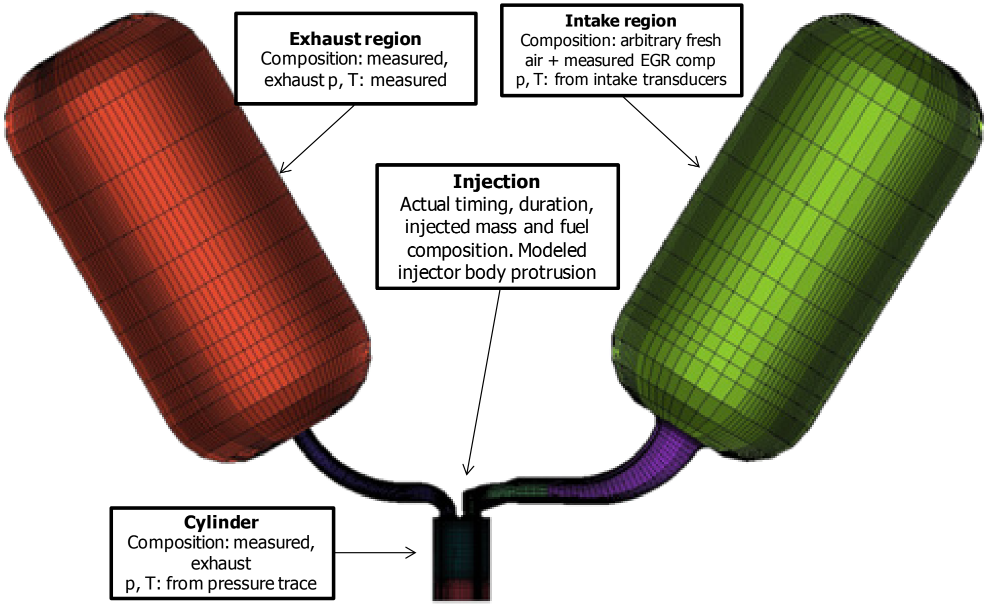
Isotropic 3D (spherical)
a = 0, n = 3

Perini, 2017

In-cylinder flow modeling

Engine configuration	
Compression ratio	16.1 : 1
Squish height at TDC [mm]	1.36
Piston bowl geometry	Stepped-lip
Operating conditions	
Engine speed [rev/min]	1500
Intake pressure [bar]	1.5
Intake temperature [K]	372
Swirl Ratio (Ricardo) [-]	2.2
Intake charge [mol fr.]	10% O_2, 81% N_2, 9% CO_2
FRESCO solver setup	
mesh accuracy	Body-fitted, unstructured hexa
time accuracy:	hybrid 1st-order implicit (diffusion, momentum) / explicit (advection)
spatial accuracy:	2nd-order (diffusion) upwind (advection)

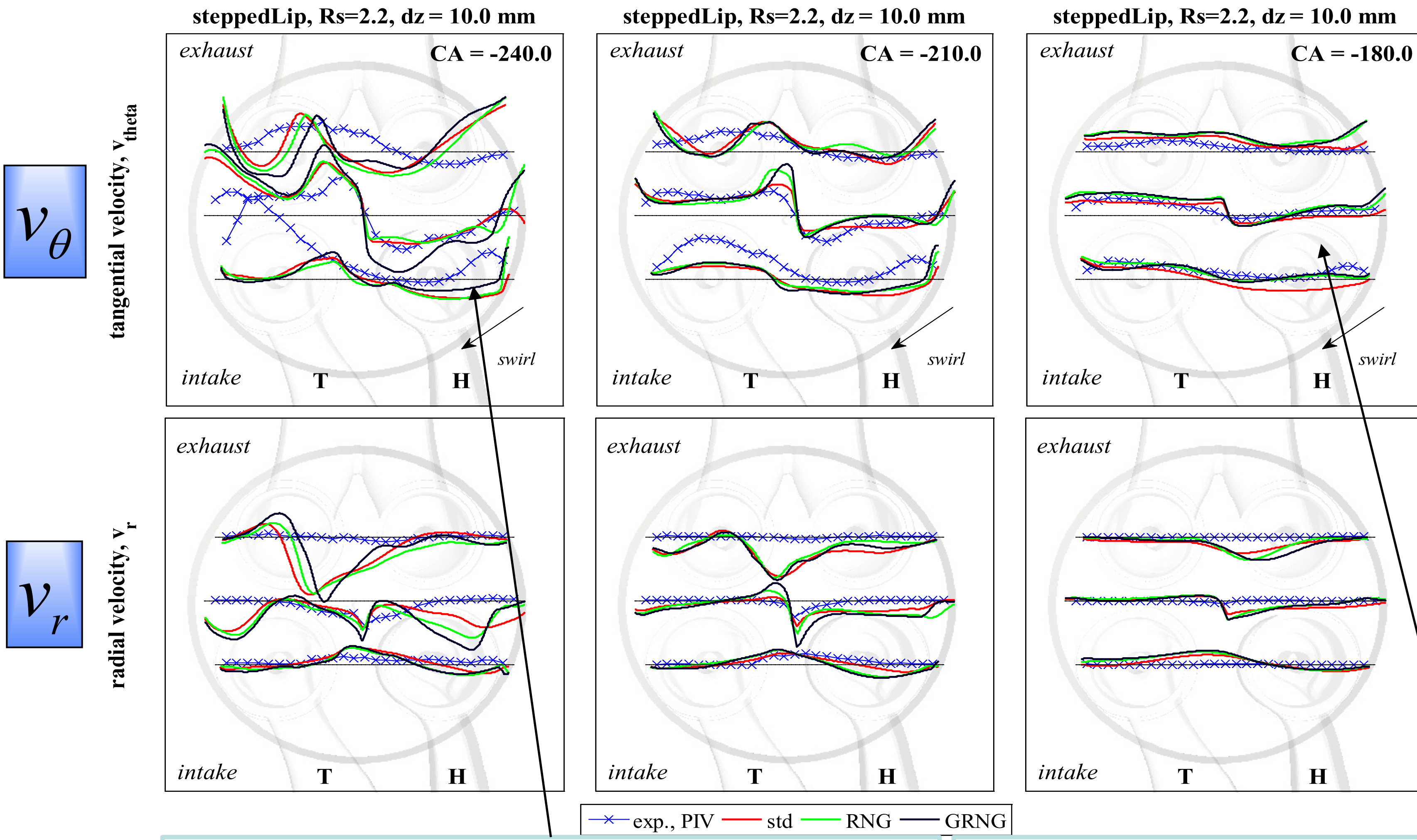




- Flow configuration from moderately-boosted, low-load operating condition ("LTC3")
- Experimental PIV measurement campaign provides ensemble-averaged flow structure at in-cylinder horizontal plane locations during the intake and compression strokes
- dz = 3.0, 10.0, 18.0 mm from fire-deck

Perini, 2017

Intake flow



Peak tangential velocities overestimated early after IVO
→ best predictions from std. k-epsilon

RNG models better during late intake (swirl + compression motion)

Summary

Engine system models can be used to provide estimates of engine efficiencies, if combustion details (e.g., timing and duration) and heat transfer losses are assumed

Multi-dimensional (3-D) models are available to predict flow and combustion details (combustion to be discussed in next lectures)

3-D models require accurate turbulence modeling for compressible engine flows – DNS data useful for model formulation.

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