

Lecture 10.

Turbulent Combustion in Piston Engines

part 1

Content

- Piston engines – introduction and definitions
- Flow and turbulence in piston engines
- Laminar and turbulent burning velocity
- Combustion duration
- Engine knock and compression ratio

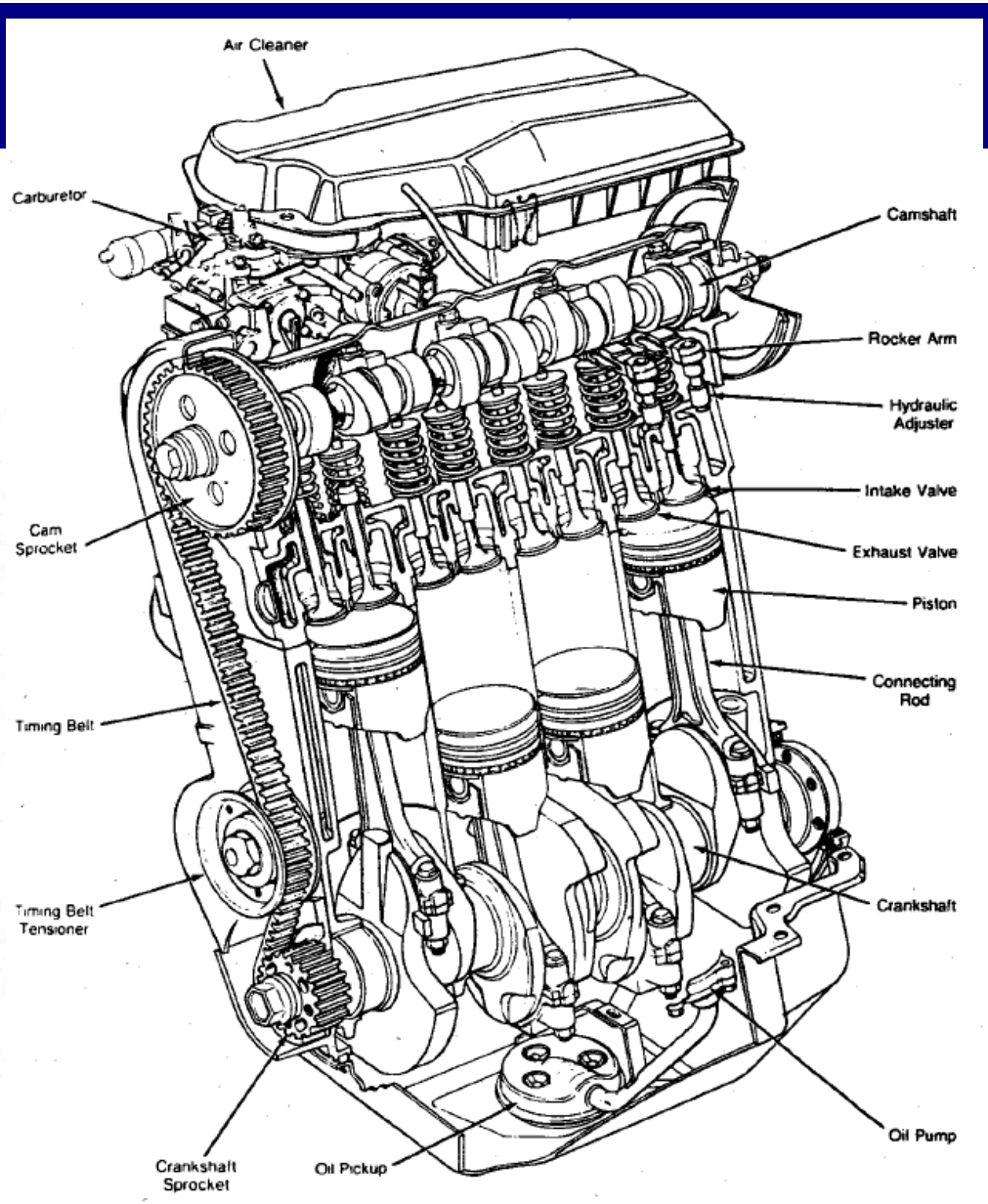
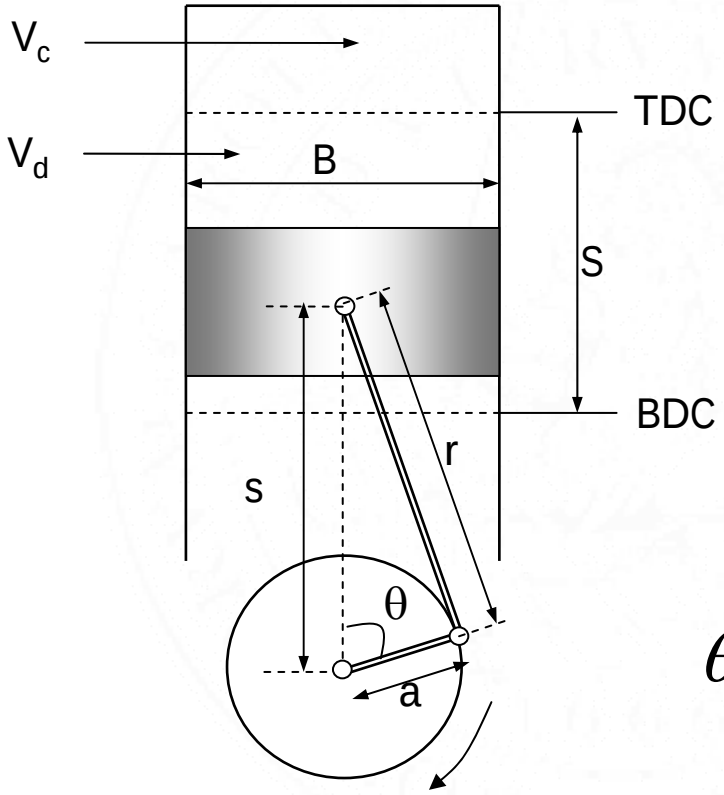


FIGURE 1-4
Cutaway drawing of Chrysler 2.2-liter displacement four-cylinder spark-ignition engine.[11] Bore 87.5 mm, stroke 92 mm, compression ratio 8.9, maximum power 65 kW at 5000 rev/min.

Schematical diagram of Reciprocating internal combustion engine



B = bore
S = stroke
r = connecting rode length
a = crank offset
s = piston position
θ = crank angle
V_c = clearance volume
V_d = displacement volume

$$\theta = \pm\pi/2,\ U_p = \mp 2\pi aN$$

$$\bar{U}_p \sim 5-20\ \text{m/s}$$

Table 10.1 Examples of engine operating parameters

	Very small engines	Small engines	Large engines
Bore (cm)	2.00	9.42	50
Stroke (cm)	2.04	9.89	161
Displacement/cylinder (L)	0.0066	0.69	316
Engine speed (N, rpm)	13000	5200	125
Power/cyl (kW)	0.72	35	311
Average piston speed (m/s)	8.84	17.1	6.71
Power/displacement (kW/L)	109	50.7	0.98
BMEP* (kPa)	503	1170	472
Examples	Model airplane two-stroke	Automobile four-stroke	large stationary two-stroke

*BMEP = brake mean effective pressure: it represents cylinder pressure that contributes effectively to mechanical work

Formula 1 engine run at higher speed – 14000 rpm, 35 m/s

Thermodynamic analysis of motored run

Compression ratio

$$r_c = \frac{V_c + V_d}{V_c}$$

motor = motor run = no combustion

Thermodynamic analysis of motored run

In-cylinder pressure and temperature due to isentropic compression

$$pv = \sum\left(\frac{Y_i}{W_i}\right)R_u T = RT \quad \text{(Equation of state)}, \quad \Rightarrow V = mv$$

$$\Rightarrow \frac{p_{TDC,motor} V_c}{T_{TDC,motor}} = \frac{p_{BDC,motor}}{T_{BDC,motor}}(V_d + V_c),$$

$$\frac{p_{TDC,motor}}{T_{TDC,motor}}\left(\frac{p_{BDC,motor}}{T_{BDC,motor}}\right)^{-1} = \frac{(V_d + V_c)}{V_c} = r_c,$$

isentropic compression: entropy=constant

$$Tds = C_v dT + pd\frac{1}{\rho} = C_v dT + pdv = 0, \quad \frac{p_{TDC,motor}}{T^{\gamma/(\gamma-1)}_{TDC,motor}} = \frac{p_{BDC,motor}}{T^{\gamma/(\gamma-1)}_{BDC,motor}}$$

motor = motor run = no combustion

Thermodynamic analysis of motored run

Compression ratio

$$r_c = \frac{V_c + V_d}{V_c}$$

In-cylinder pressure and temperature due to abiabatic compression

$$\frac{p_{TDC,motor} V_c}{T_{TDC,motor}} = \frac{p_{BDC,motor}}{T_{BDC,motor}} (V_d + V_c),$$

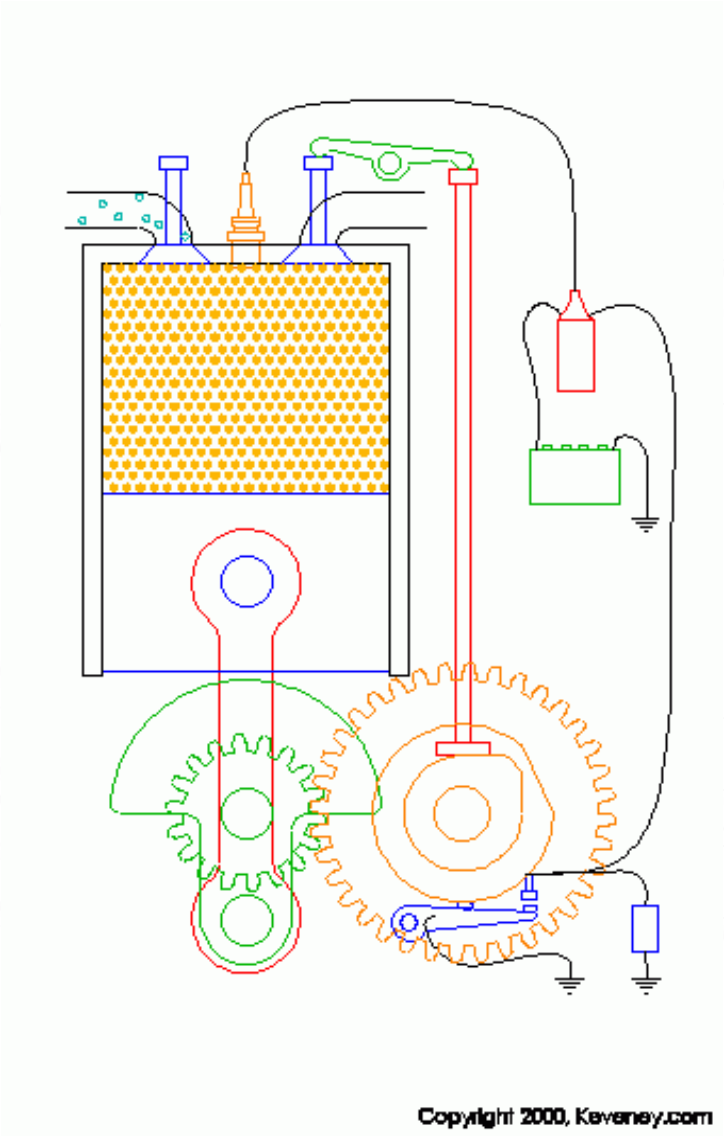
$$\frac{p_{TDC,motor}}{p_{BDC,motor}} = r_c^{\gamma},$$

$$\frac{p_{TDC,motor}}{T^{\gamma/(\gamma-1)}_{TDC,motor}} = \frac{p_{BDC,motor}}{T^{\gamma/(\gamma-1)}_{BDC,motor}}$$

$$\frac{T_{TDC,motor}}{T_{BDC,motor}} = r_c^{\gamma-1}$$

motor = motor run = no combustion

An example of a four stroke engine



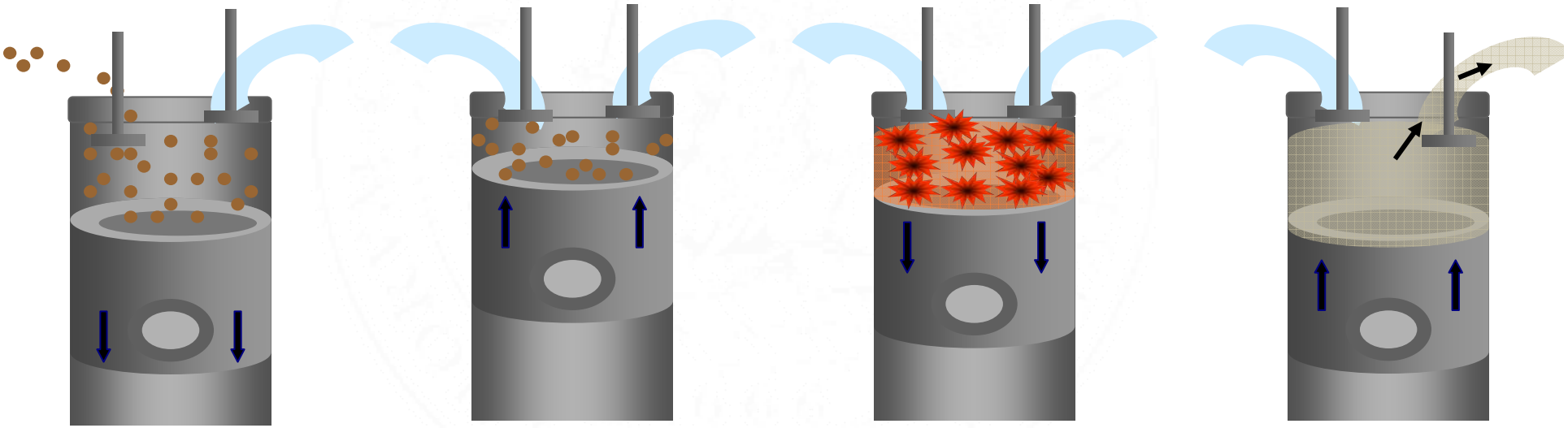
Four stroke engine cycles

Intake stroke

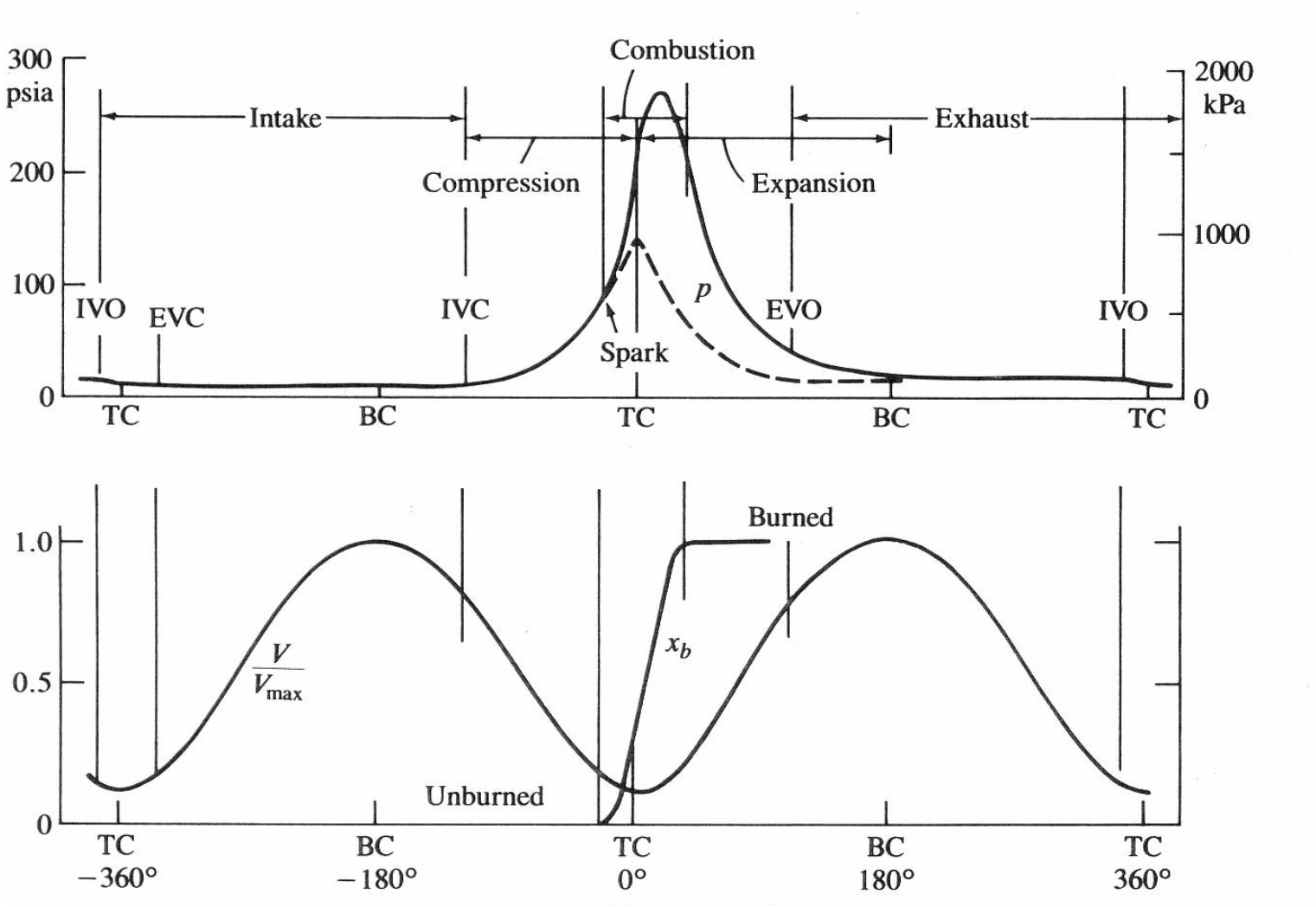
Compression stroke

Combustion/
power stroke

Exhaust stroke

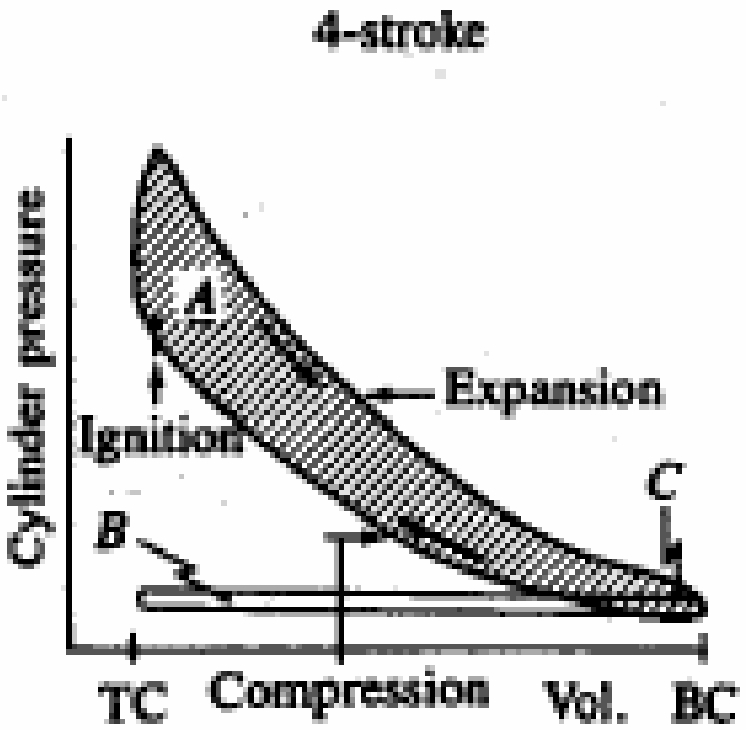


Sequences of events in four stroke SI engine operating cycle

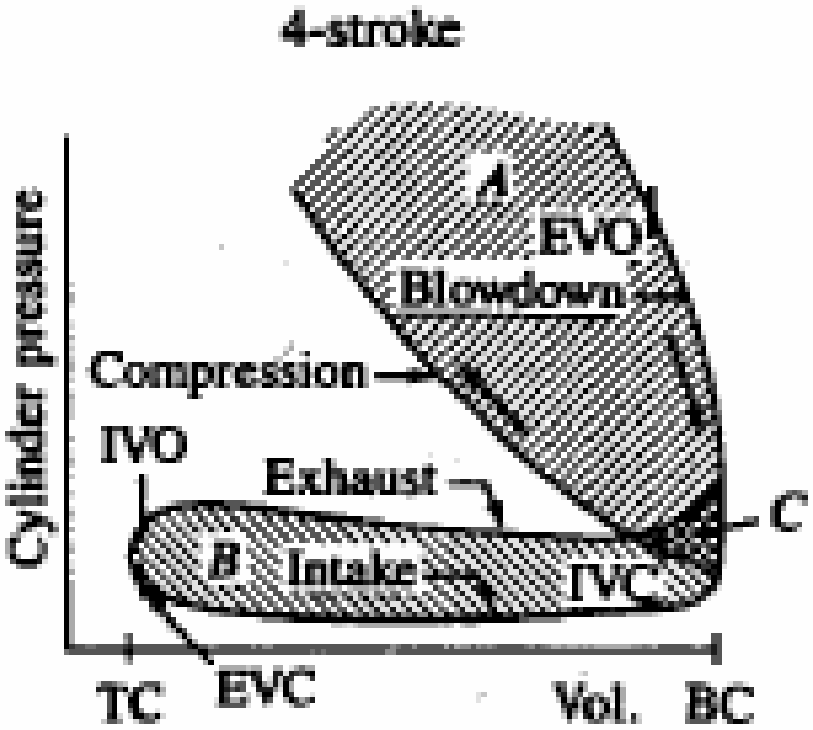


P-V graph

SI engine p-v graph



Local zoomed p-v graph



Otto cycle effciency (indicated thermal efficiency) $\eta = 1 - \left(\frac{1}{r_c}\right)^{\gamma-1}$

Works

- Indicated work: work generated in the combustion chamber (area in the P-V graph)
  - Gross indicated work: area A
  - Pump work: area B
  - Net indicated work (without supercharger): area A – area B
- Friction work
- Brake work: actual work available at the crankshaft
  - Brake work = indicated work – friction and parasitic work
- Mechanical efficiency: brake work/indicated work (40-90%)
  - lower engine speed has higher mechanical efficiency (friction)
  - lower engine speed has lower thermal efficiency (heat loss)

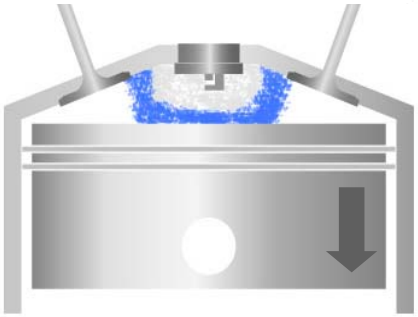
Mean effective pressure (mep) and power

- Mean effective pressure = work / displacement volume
  - imep
  - fmep
  - bmep
  - pmep

- Power: rate of work of the engine
  - Power = work x engine speed / number of revolution per cycle
  - Power = (mep) x face area of all pistons x averaged piston velocity

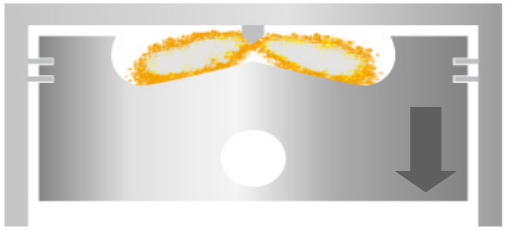
- Torque: force acting at a moment distance = brake work /2π
  - Torgue = (bmep) x dispacement distance / 2π / number of revolution per cycle

Otto



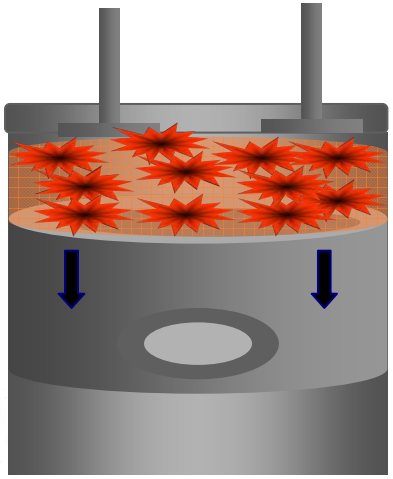
- Premixed flame (injection prior to combustion)
- Blue flame (no soot)
- lambda ~1 (three-way cat OK)

Diesel



- Combustion during injection
- Diffusion flame
- Yellow flame (=soot)
- Local lambda ~1
- Overall lambda ~1,5 (no three-way cat)

HCCI



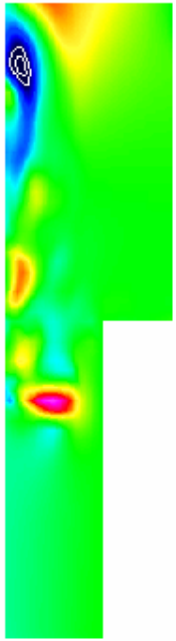
- Premixed “Homogeneous combustion”
- “zero” NOx and soot
- High efficiency ~45%

  Also at partload
- Control
- Supercharge/EGR
- Noise
- Cold start

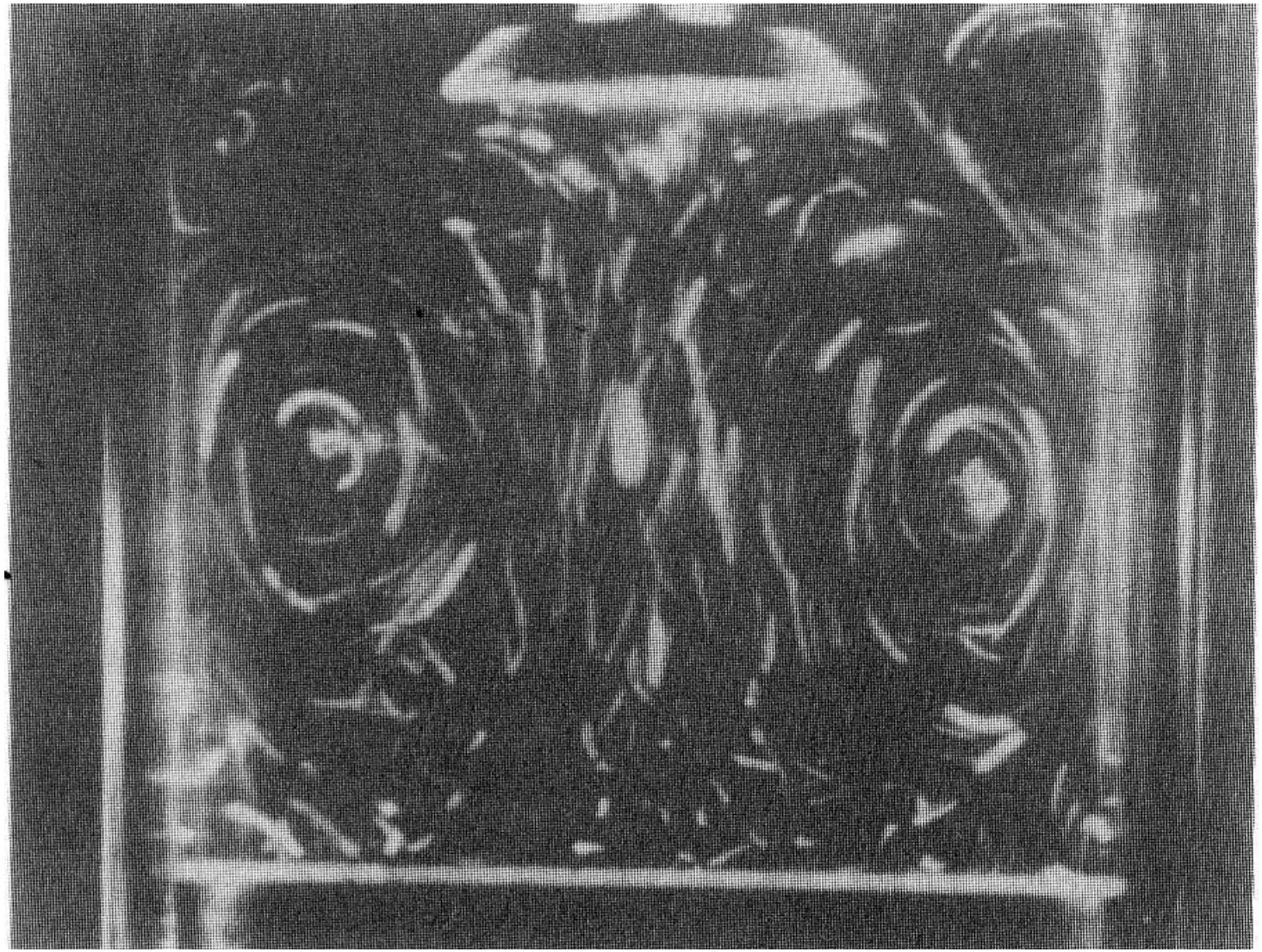
SI engines

Most cars run with SI engine

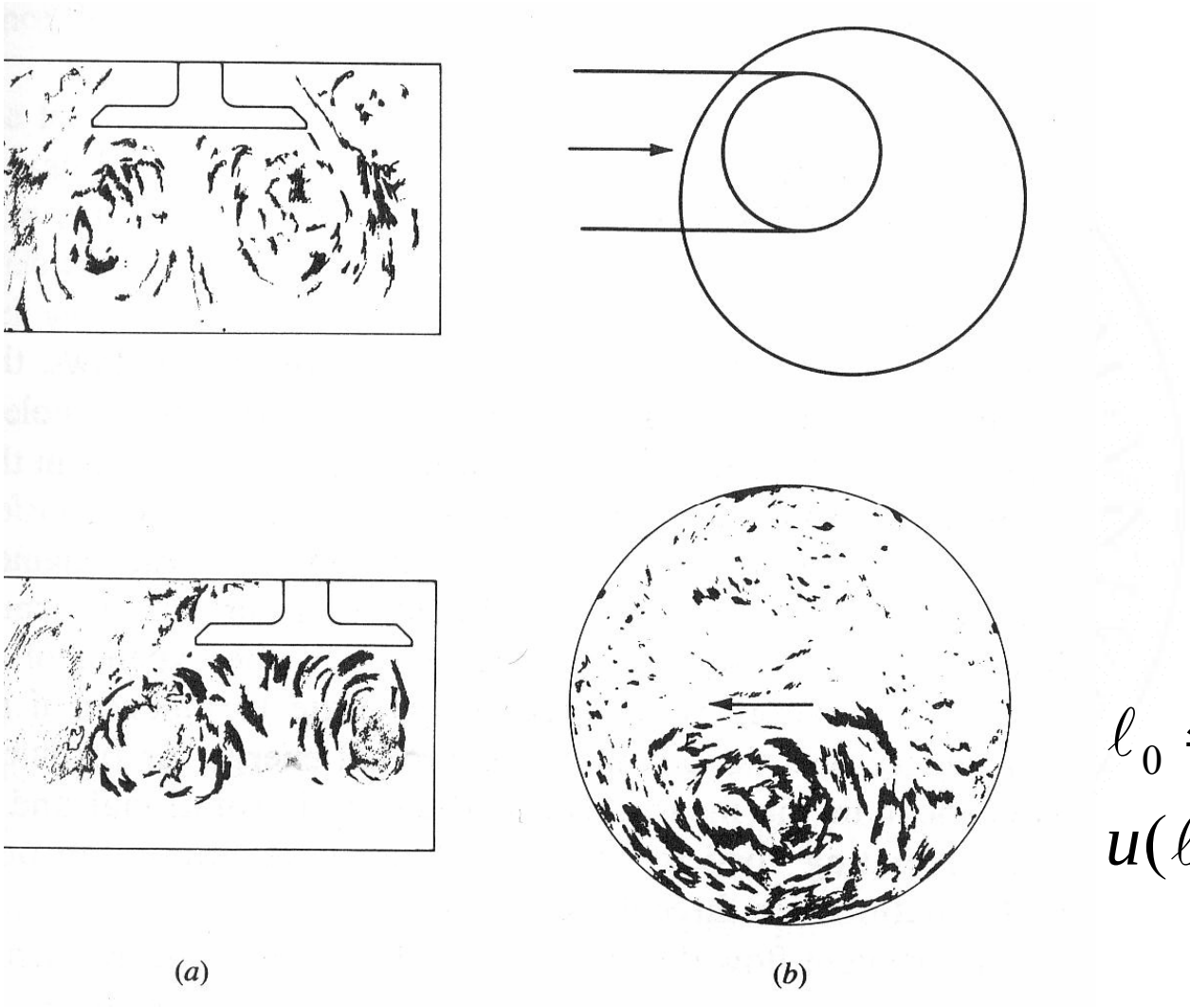




Flow field



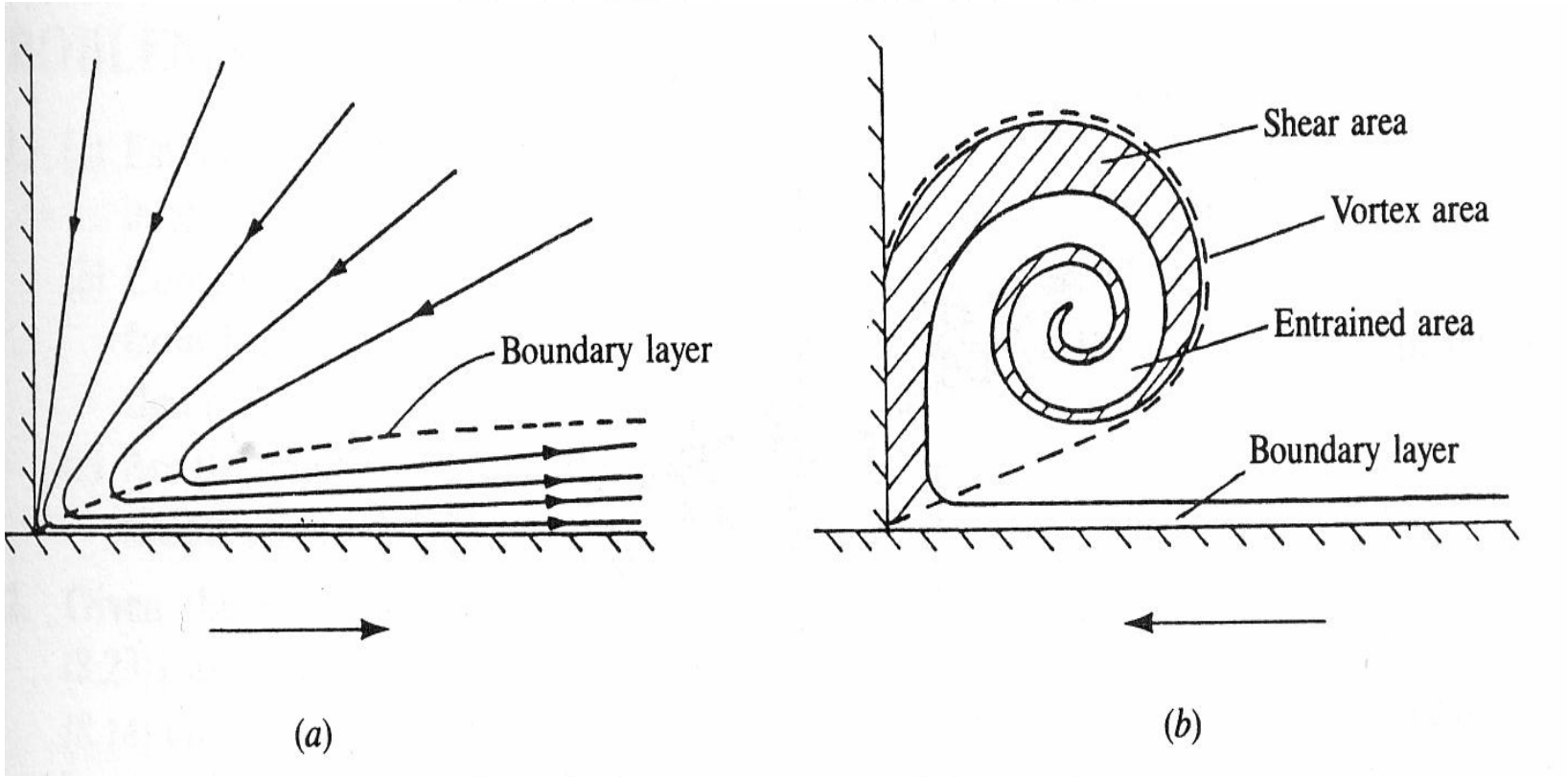
Flow field



$$\ell_0 \simeq 0.5B,$$

$$u(\ell_0) \equiv u' \simeq u'_{rms}$$

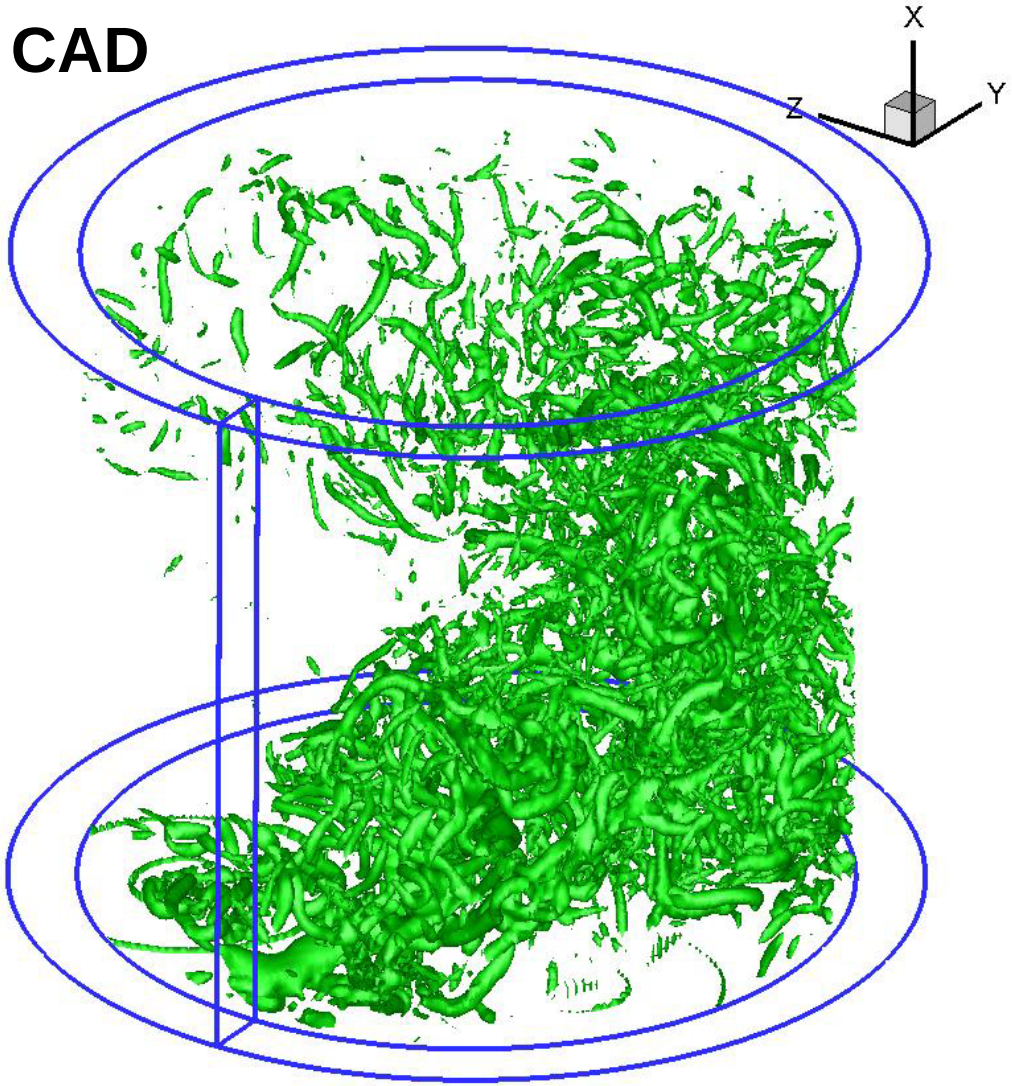
Flow field near the piston



3-D turbulence eddies in an engine

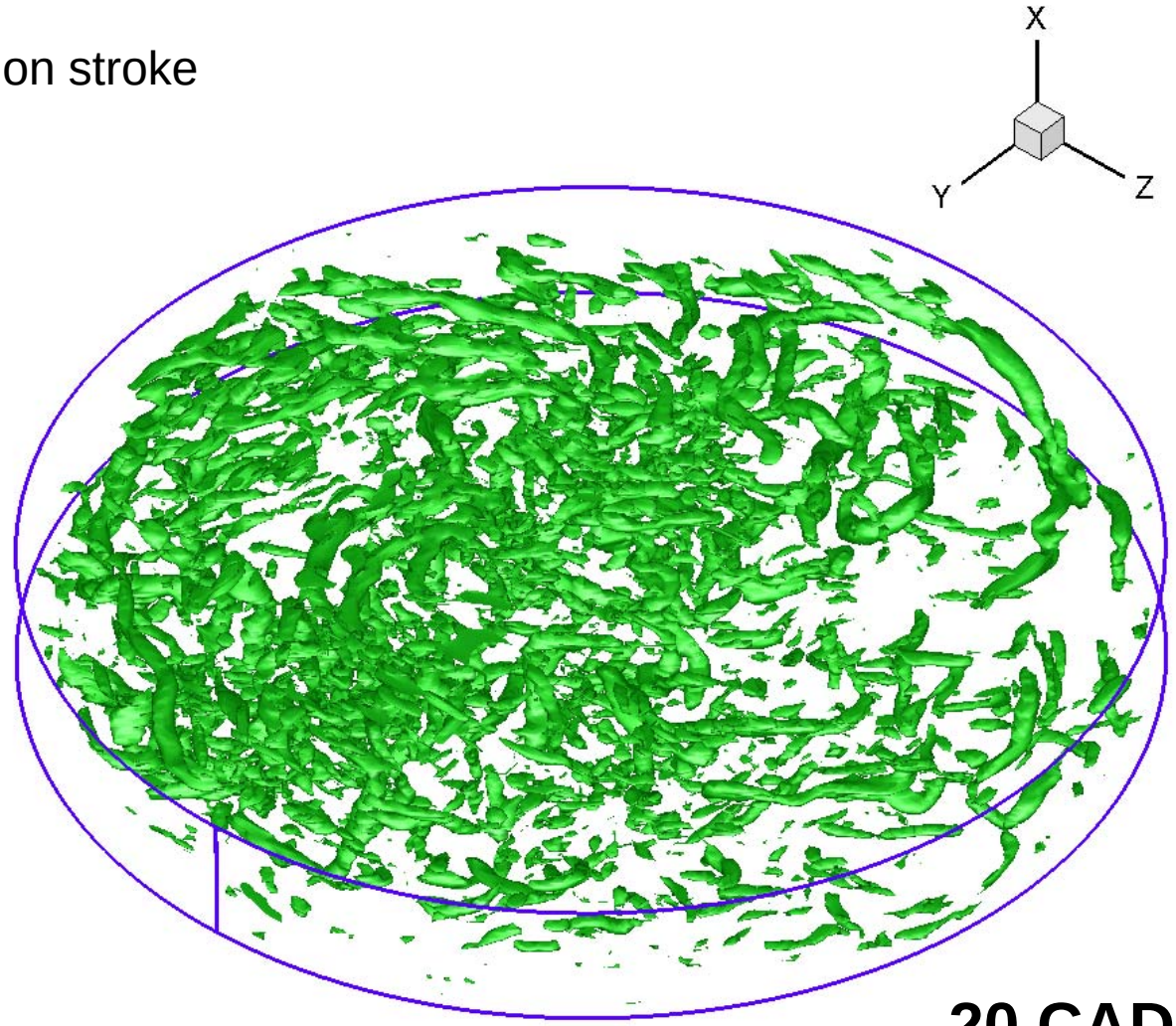
-270 CAD

Intake stroke



3-D turbulence eddies in an engine

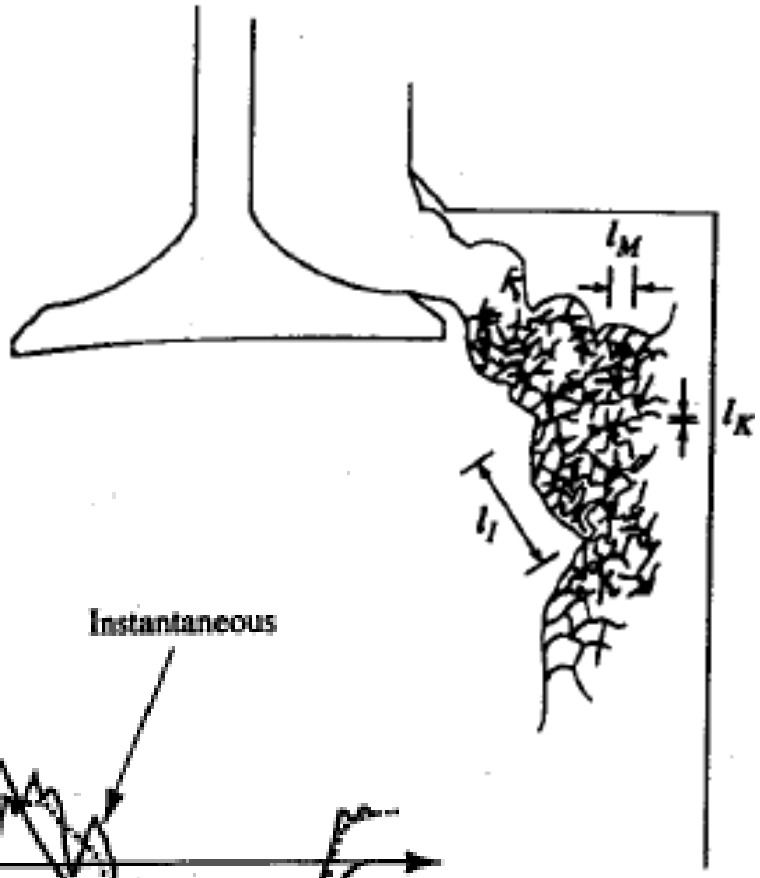
Compression stroke

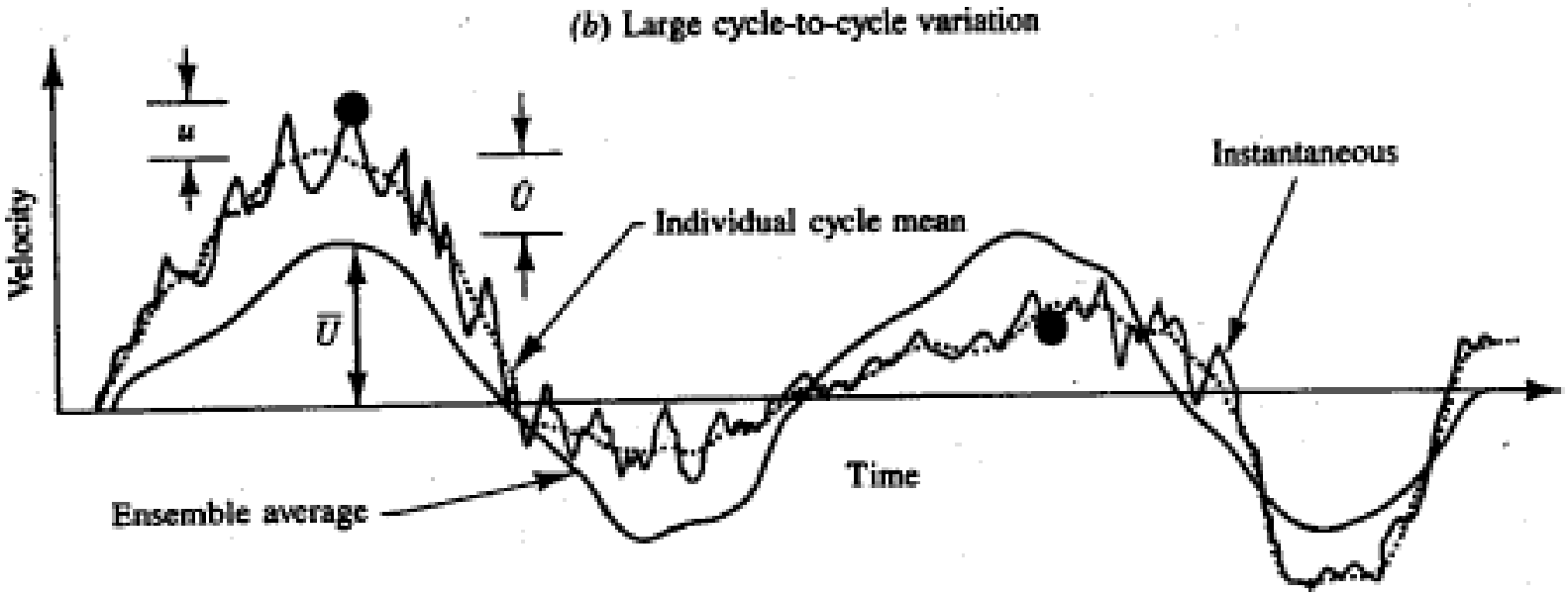


-20 CAD

Turbulent flows inside the cylinder

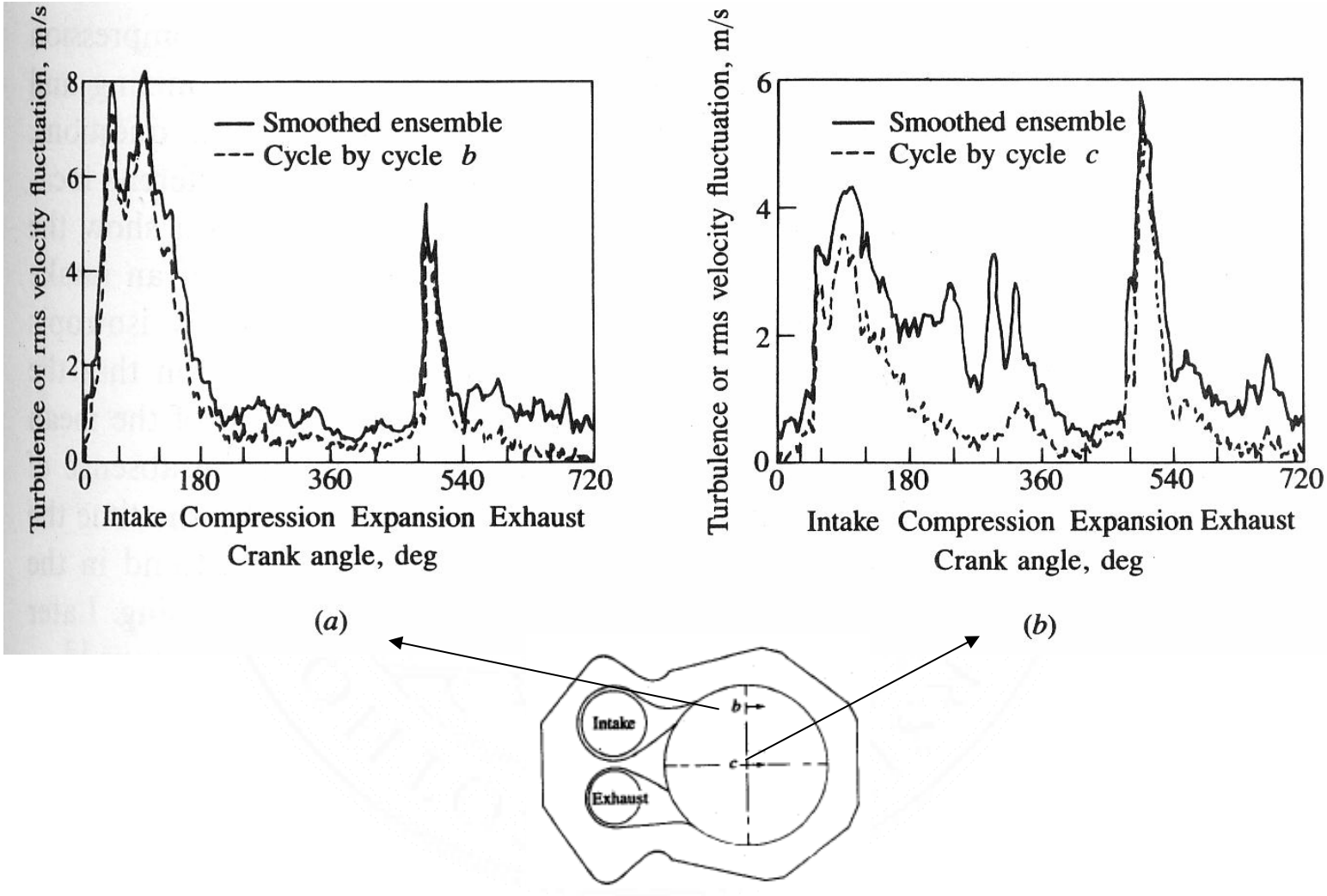
Ensemble average based on multi-cycles
Time average based on individual cycles
Turbulence velocity





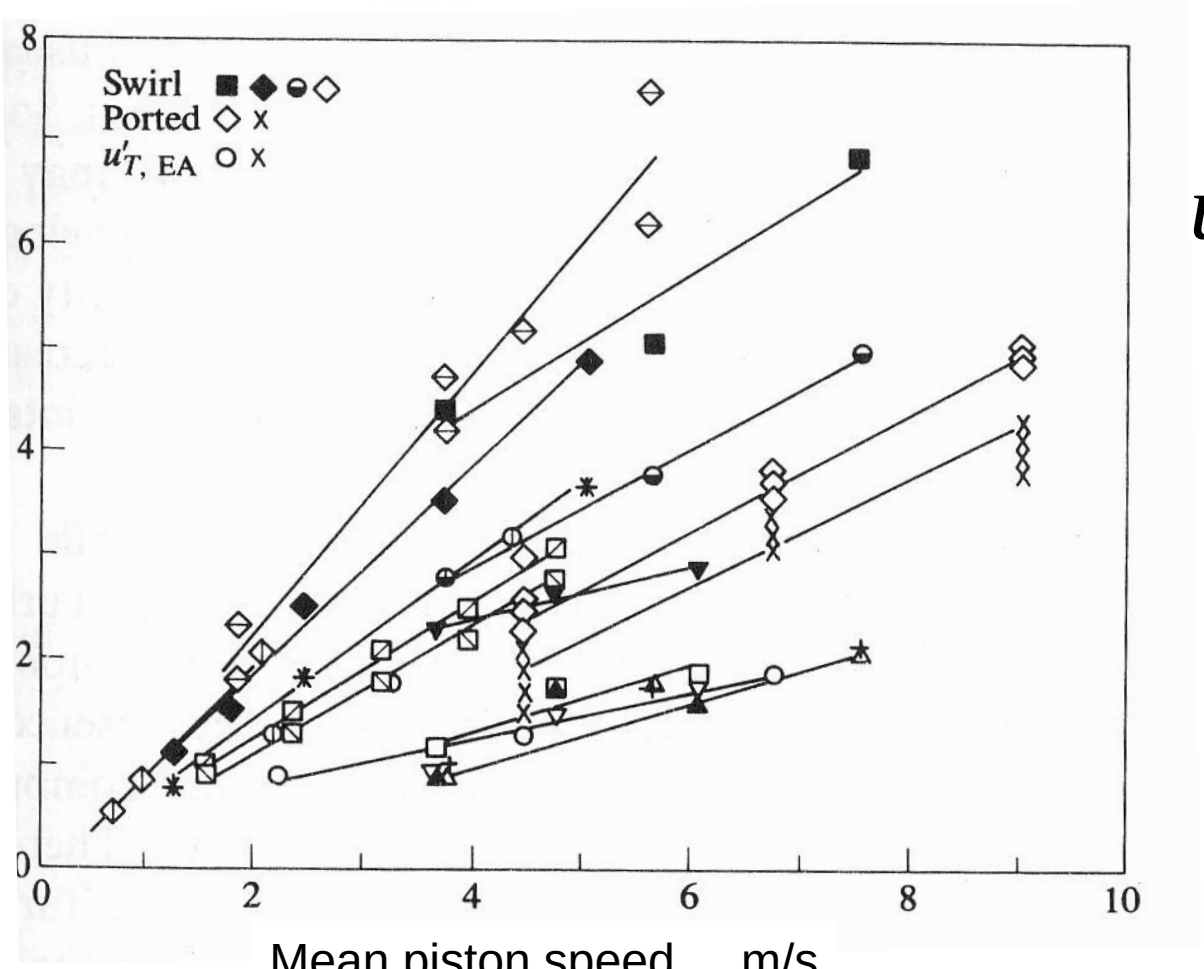
Turbulence large eddy velocity u'_{rms}

with cycle-cycle variation (solid), without cycle-cycle variation (dashed)



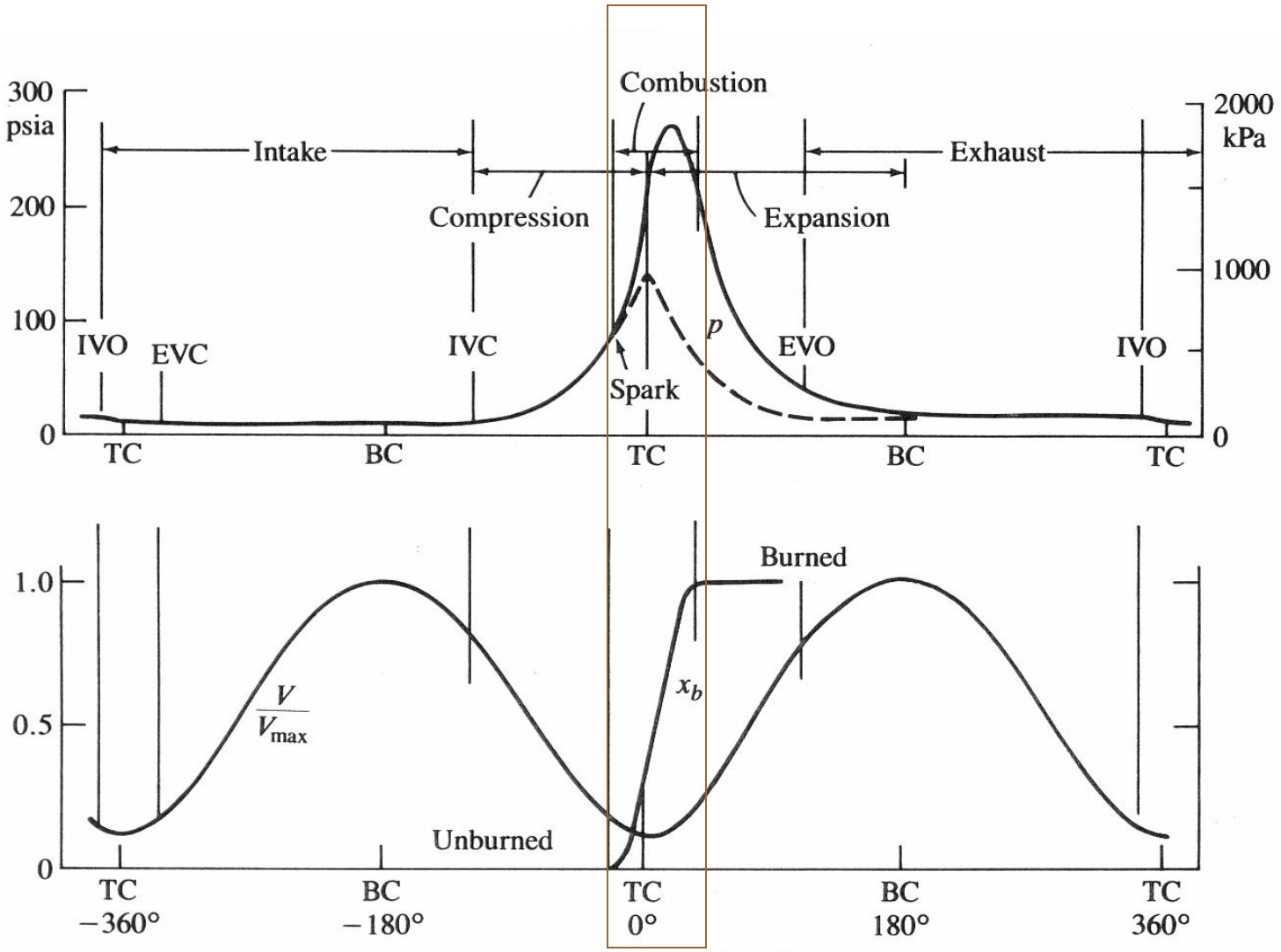
Turbulence large eddy velocity u'_{rms} at TDC

Turbulence m/s

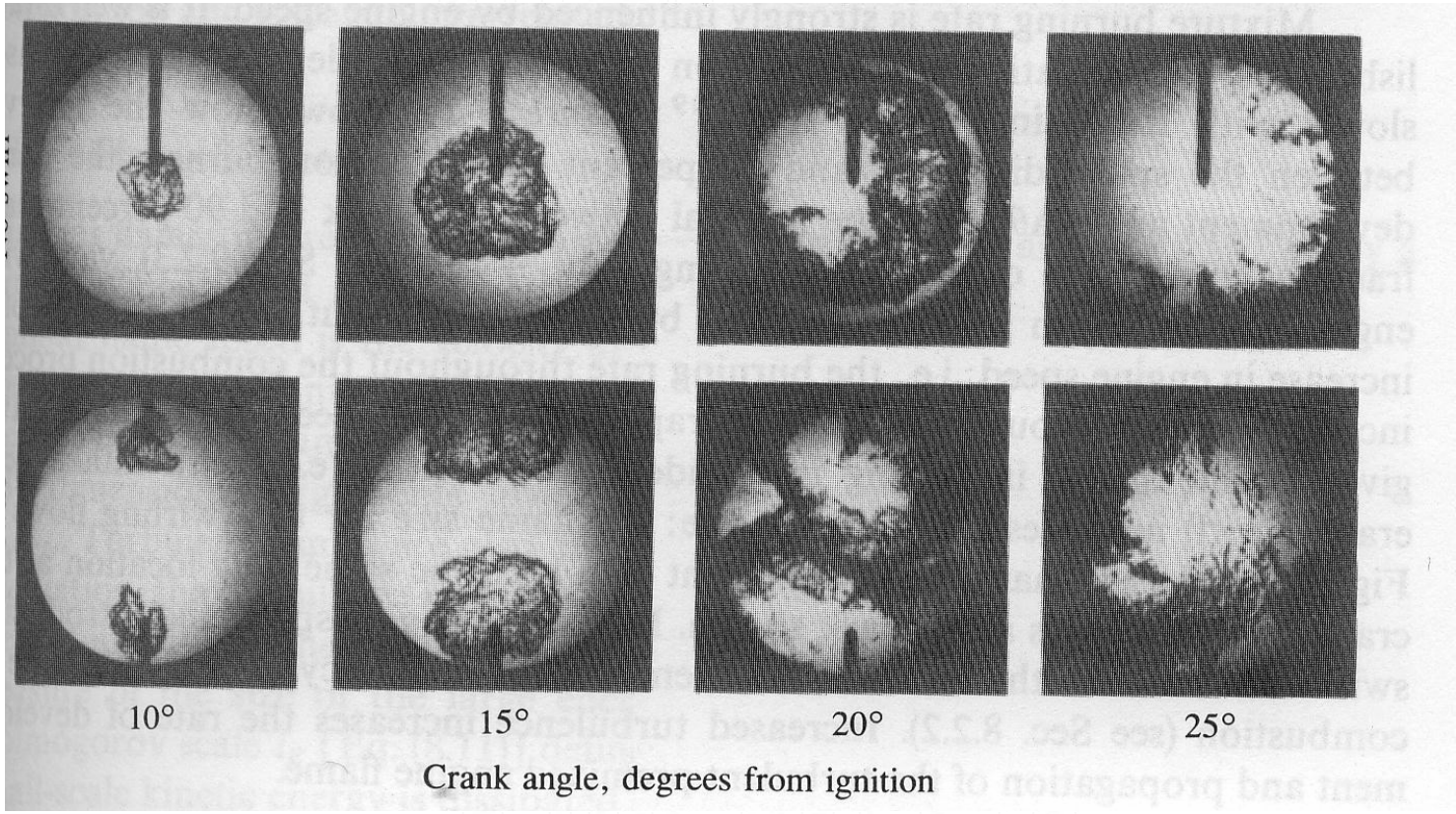


$$u' \simeq KN$$

Combustion process in SI engines



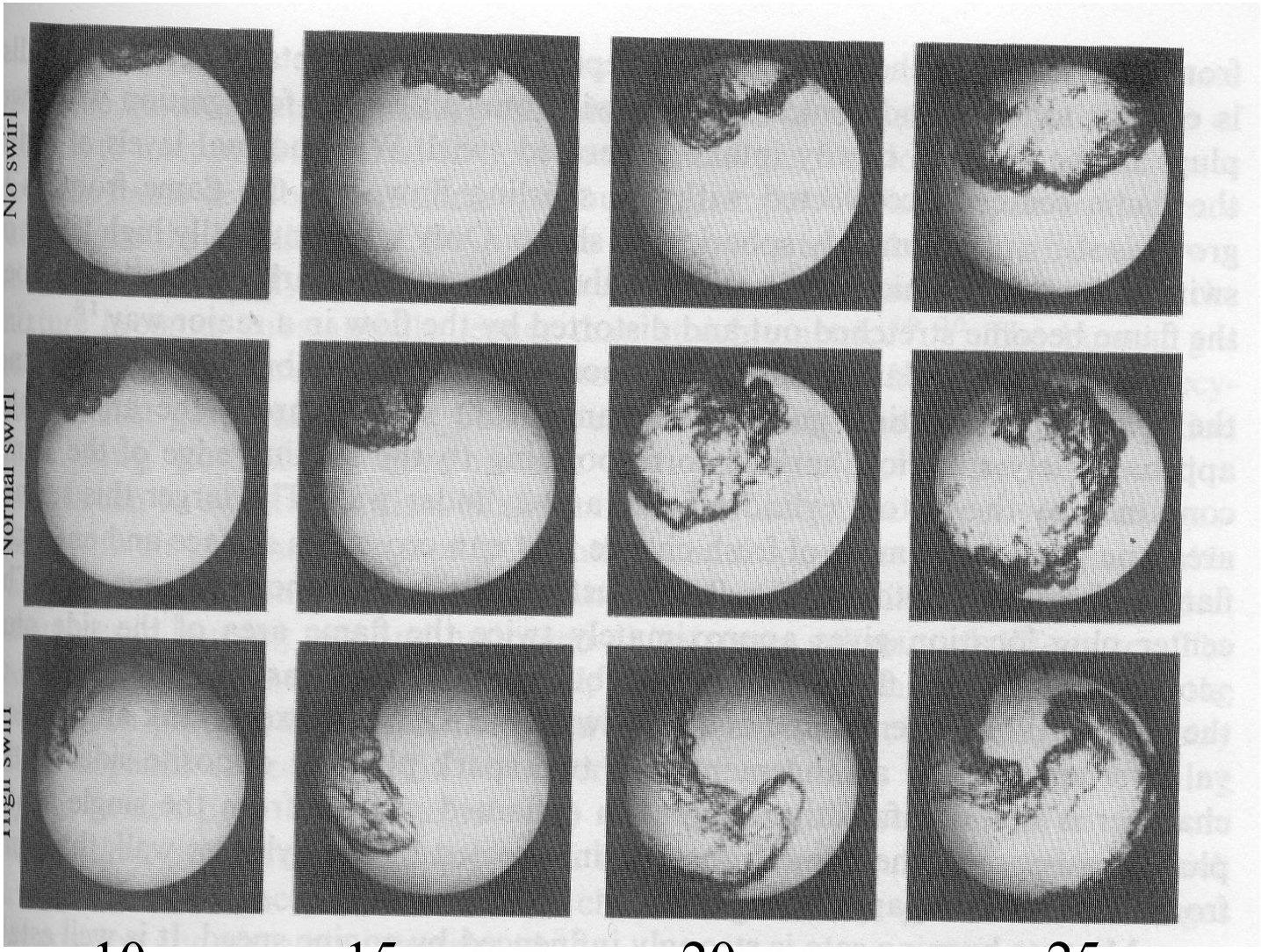
Turbulent flame propagation in SI engine



No swirl,
central plug

No swirl,
two plugs

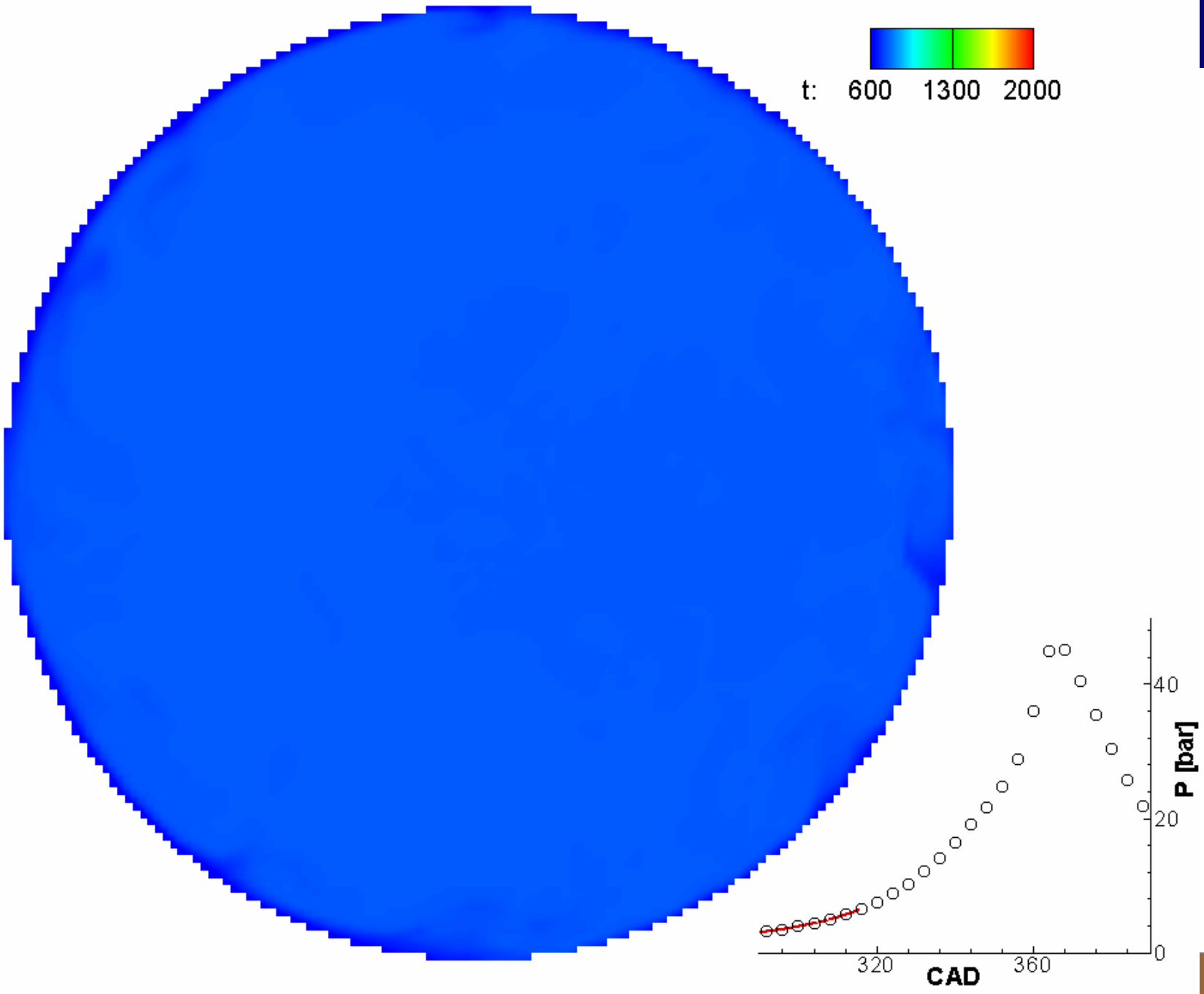
Turbulent flame propagation in SI engine



No swirl, side plug

normal swirl, side plug

high swirl, side plug


t: 600 1300 2000
40
20
0
P [bar]
320
360
CAD

laminar burning velocity

Empirical expression

$$S_L = S_{L,0}\left(\frac{T_u}{T_{u,0}}\right)^{\alpha}\left(\frac{p}{p_0}\right)^{\beta}$$

$$\alpha = 2.18 - 0.8(\Phi - 1)$$

$$\beta = -0.16 + 0.22(\Phi - 1)$$

$$S_L = S_{L,0} r_c^{\alpha(\gamma-1)+\beta\gamma} = S_{L,0} r_c^{0.648-0.012(\Phi-1)} \simeq S_{L,0} r_c^{0.648}$$

Laminar flame thickness

$$D \simeq D_0 \left(\frac{p_0}{p} \right) \left(\frac{T_u}{T_{u,0}} \right)^{1.7} \simeq D_0 r_c^{-0.72},$$

$$D \simeq \delta_L S_L,\ D_0 \simeq \delta_{L,0} S_{L,0},$$

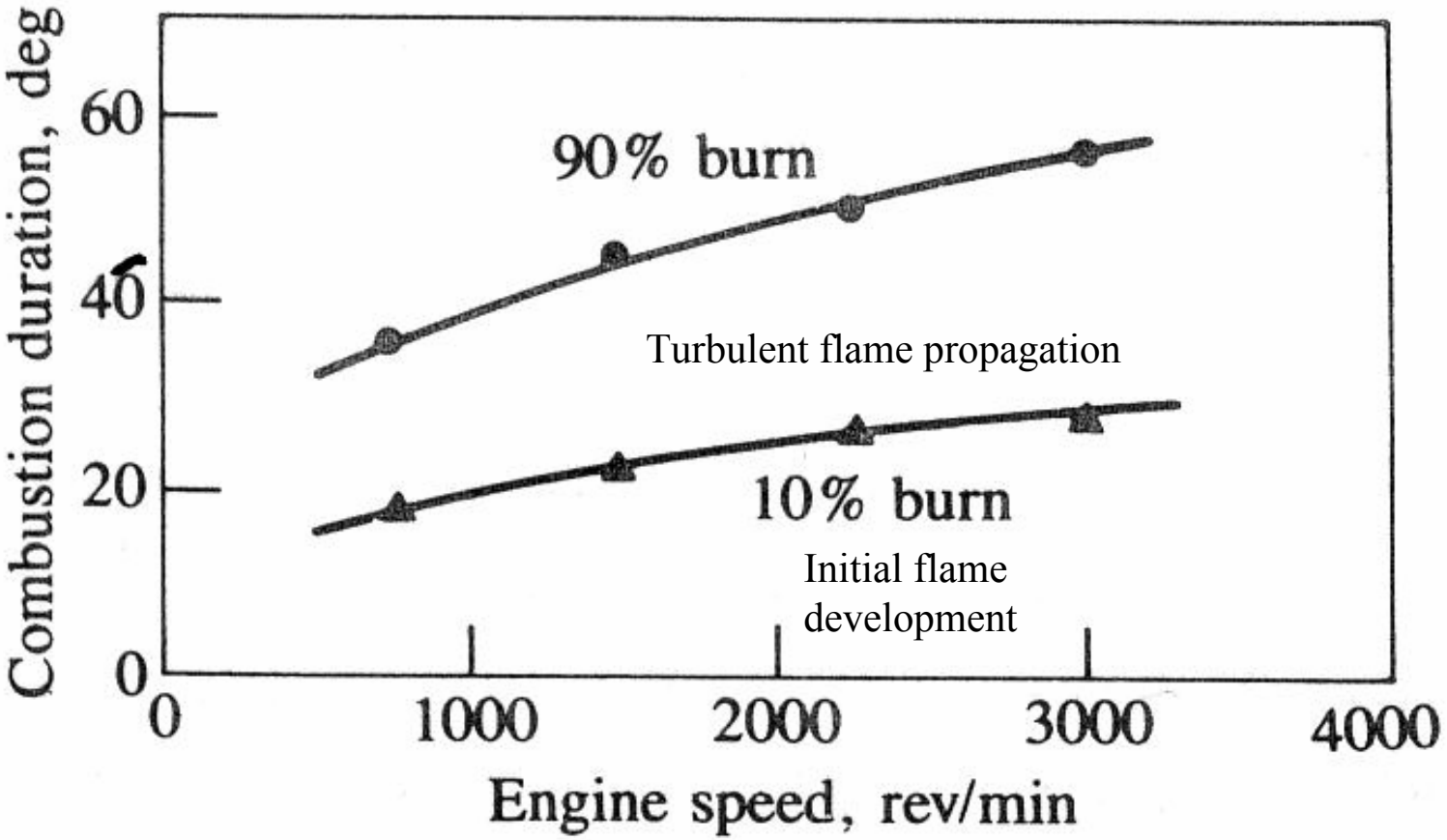
$$\delta_L \simeq \delta_{L,0} \left(\frac{S_{L,0}}{S_L} \right) \left(\frac{D}{D_0} \right) \simeq \delta_{L,0} r_c^{-1.368}$$

Turbulent burning velocity and combustion duration

$$S_T \simeq S_L + u' \simeq u' \simeq KN$$

$$\Delta\theta_{flame} \sim \frac{B}{S_T} N \sim \frac{B}{u'} N \sim \frac{B}{KN} N \sim \frac{B}{K}$$

Combustion duration in SI engine



Combustion stages

- Ignition stage: spark ignition needs to last a short time to initiate the flame propagation. The flame kernel initially is very small. The small kernel is not highly wrinkled by turbulence. The propagation speed of the small flame kernel is low.

- Turbulent flame propagation stage: As the flame kernel grows the flame surface area becomes also more wrinkled. The flame propagation speed is much higher than the laminar flame speed. The higher the turbulence level the faster the turbulent flame speed. This makes the combustion duration in terms of crank angle degrees roughly the same at different engine speed.

- Burning in the post flame zones: If the engine runs at very high speed, the combustion intermediates such as CO may not fully burned at the flame front. Since the engine is at very high pressure and temperature, these intermediate can be further oxidized later before the exhaust gas is released.

Emissions

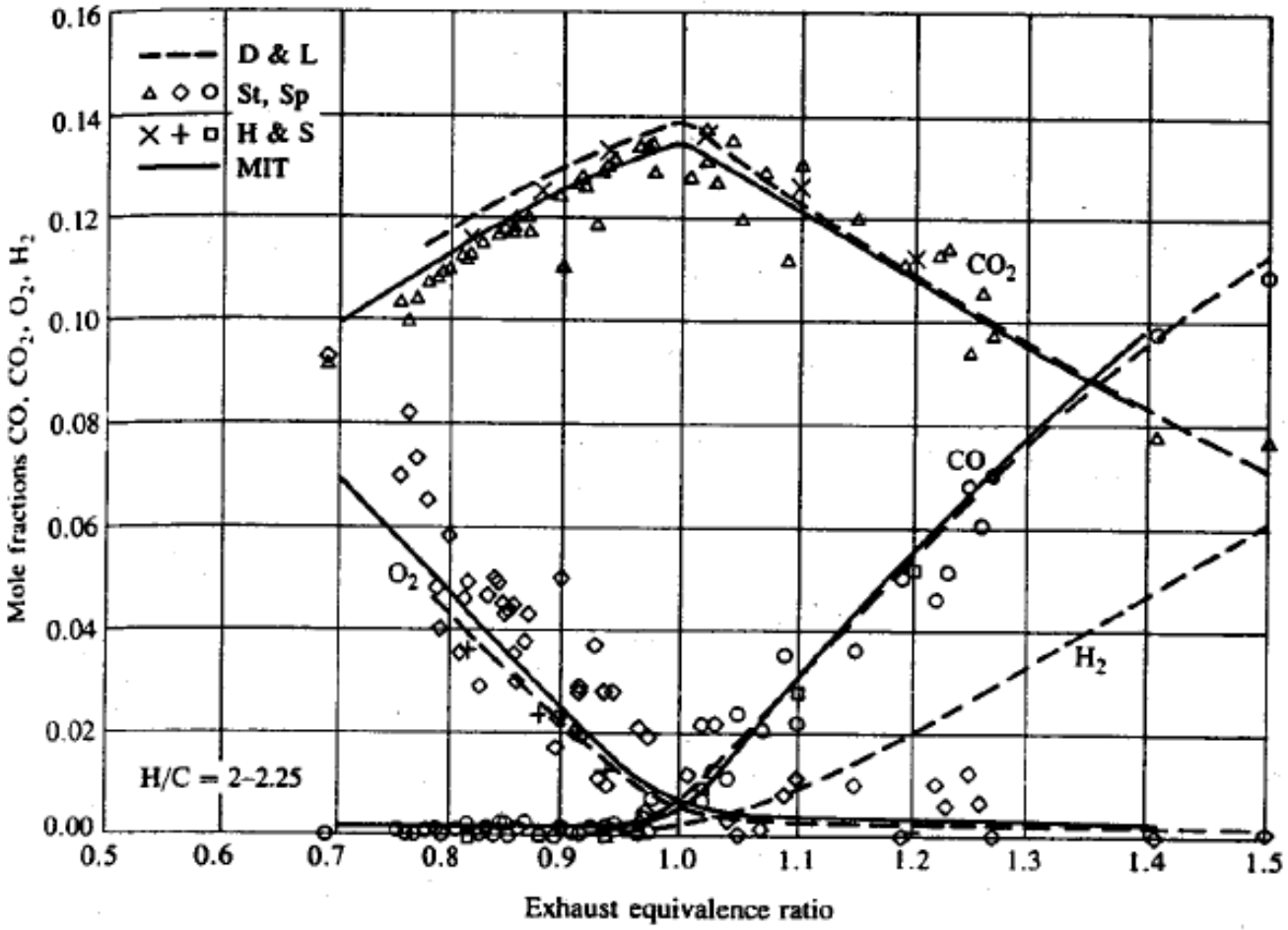
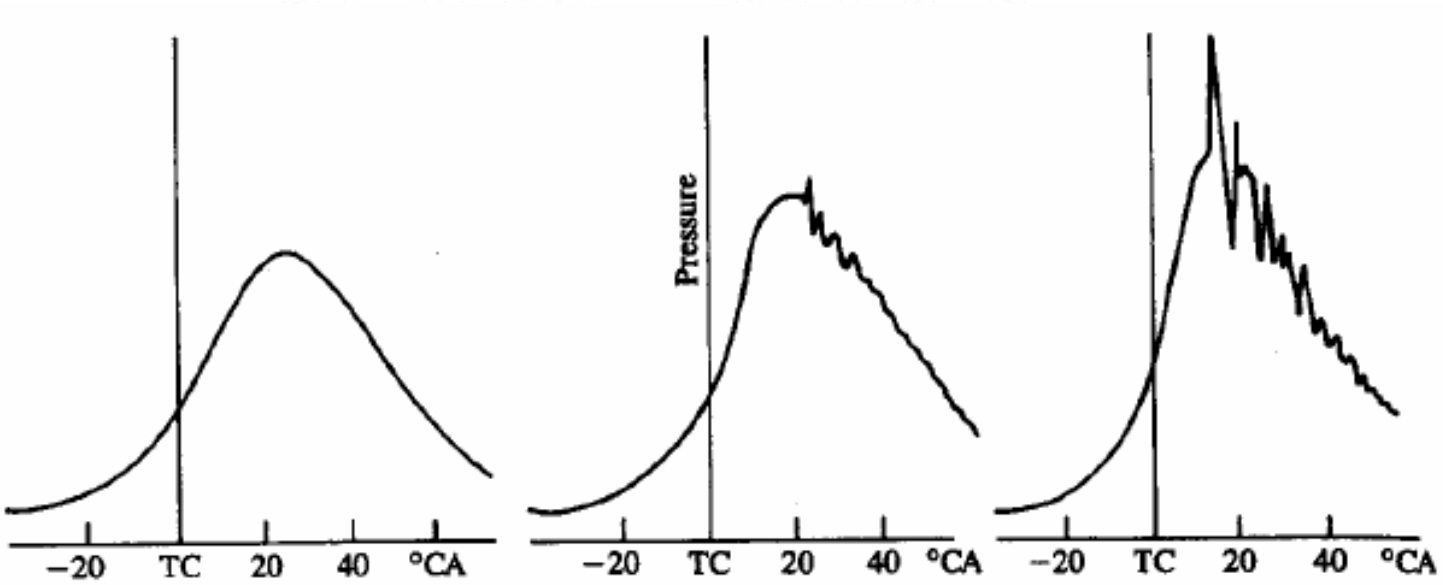


FIGURE 4-20

Spark-ignition engine exhaust gas composition data in mole fractions as a function of fuel/air equivalence ratio. Fuels: gasoline and isooctane, H/C 2 to 2.25. (*From D'Alleva and Lovell,[24] Stivender,[25] Harrington and Shishu,[26] Spindt,[27] and data from the author's laboratory at MIT.*)

Compression ratio in SI engines

- Engine knock at high compression ratio



(*a*) Normal combustion, spark 28°BTC

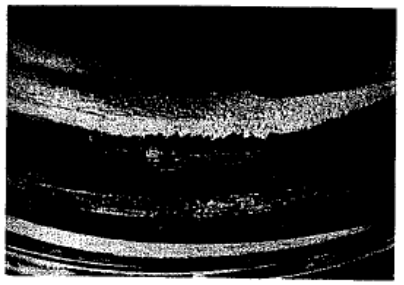
(*b*) Slight knock, spark 28°BTC

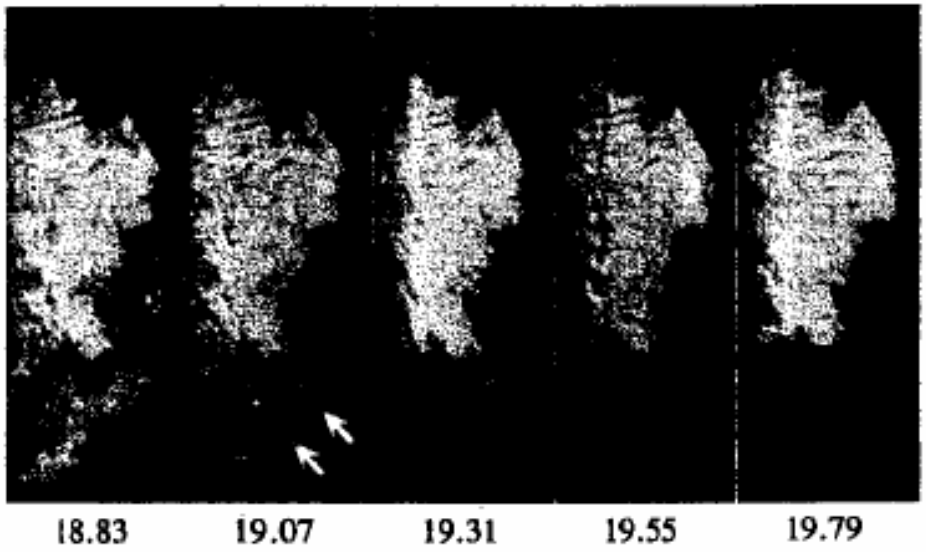
(*c*) Intense knock, spark 32°BTC

FIGURE 9-59
Cylinder pressure versus crank angle traces of cycles with (*a*) normal combustion, (*b*) light knock, and (*c*) heavy knock. 4000 rev/min, wide-open throttle, 381-cm^3 displacement single-cylinder engine.[77]

Compression ratio in SI engines

- Engine knock at high compression ratio





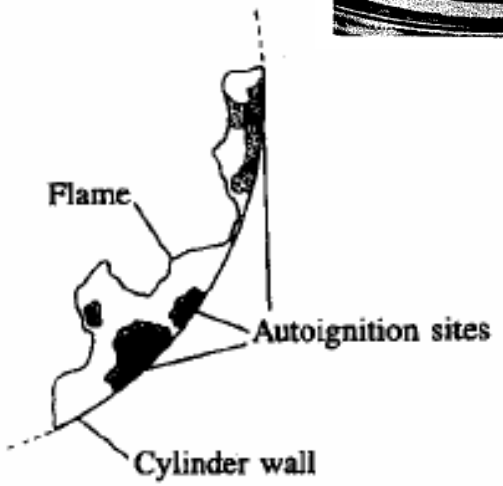
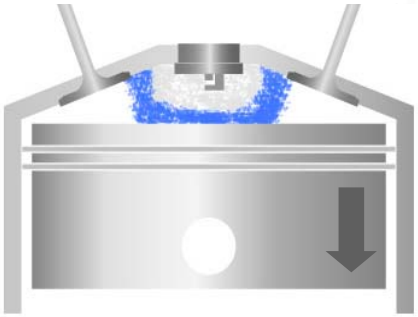


FIGURE 9-63
Five shadowgraph photographs of knocking combustion cycle identifying location of autoignition sites (arrows). Crank angle of each photo indicated: 33 μs between frames. Operating conditions and engine details as in Fig. 9-62. Photodigitized picture of second photograph showing additional details of the autoignition sites in the end-gas region on right.[89]

Spark ignited engine
(”Otto-engine”, ”gasoline engine”)



- Premixed flame (injection prior to combustion)
- Blue flame (no soot)
- lambda ~1 (three-way cat OK)

- Low efficiency, ~30% at full load, ~15% at part load, due to:
  - Load control through air amount => throttle required => low part load efficiency
  - Knock sensitivity requires low compression ratio => low overall efficiency