Advanced Powertrain Engineering MMME4066
Heat Release and Combustion Characteristics

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content

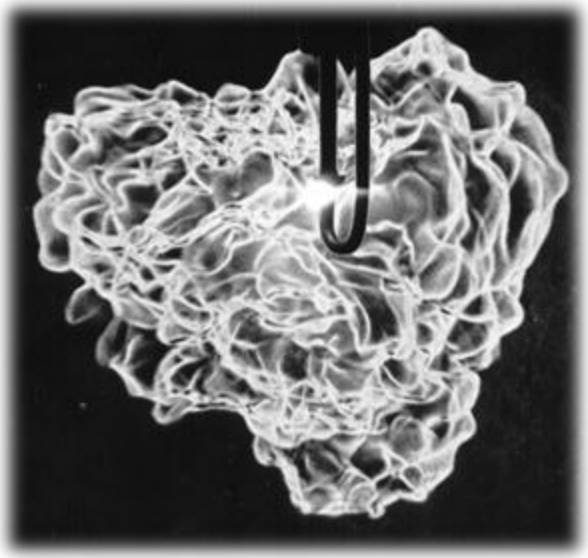
- Combustion in SI engines
- Mass fraction burned
- Maximum brake torque (MBT) timing
- Combustion in CI engines
- Worked example

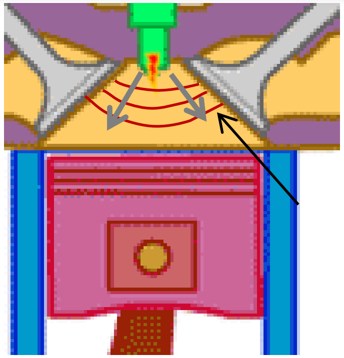
Lecture Notes
pp. 53-59
and
pp. 77-81

Combustion in IC engines

Combustion and Heat Release from SI and Diesel is usually discussed separately due to the difference in combustion processes

SI combustion is of homogeneous mixture initiated by a spark



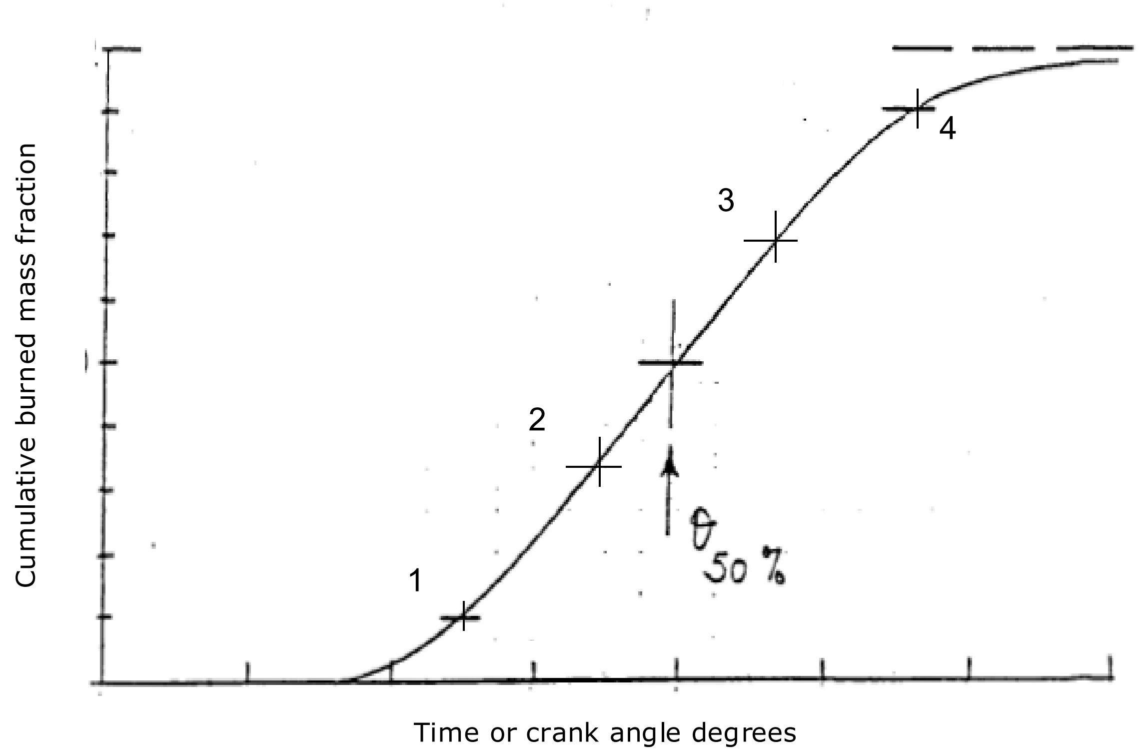


CI combustion is of fuel directly injected in Cyl. Heterogeneous mixture ingnited by auto-ignition

Characterisation of combustion processes (SI)

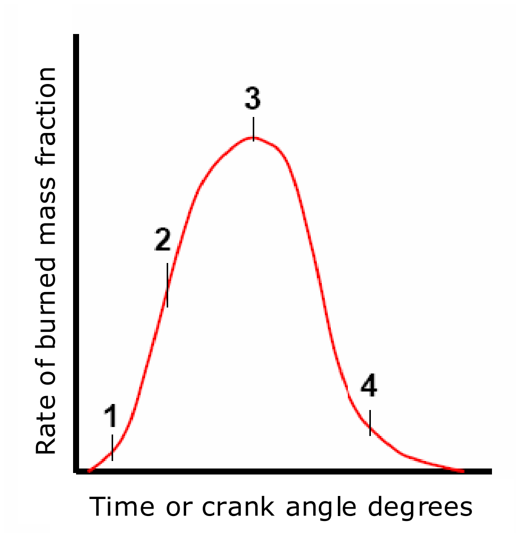
Convenient to use mass fraction burned to characterise different stages of SI combustion process by their duration in CA (defining fraction of cycle occupied)

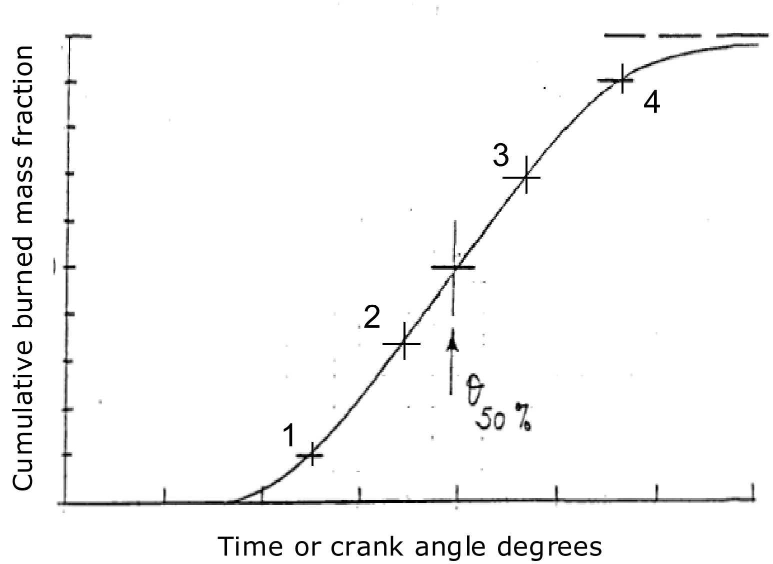
Mass fraction burned as function of crank angle has a characteristic S-shape



The burn rate is low initially because the very small flame area does not generate sufficient energy to quickly heat the surrounding gases required for sustained, rapid flame propagation.

Characterisation of combustion processes (SI)

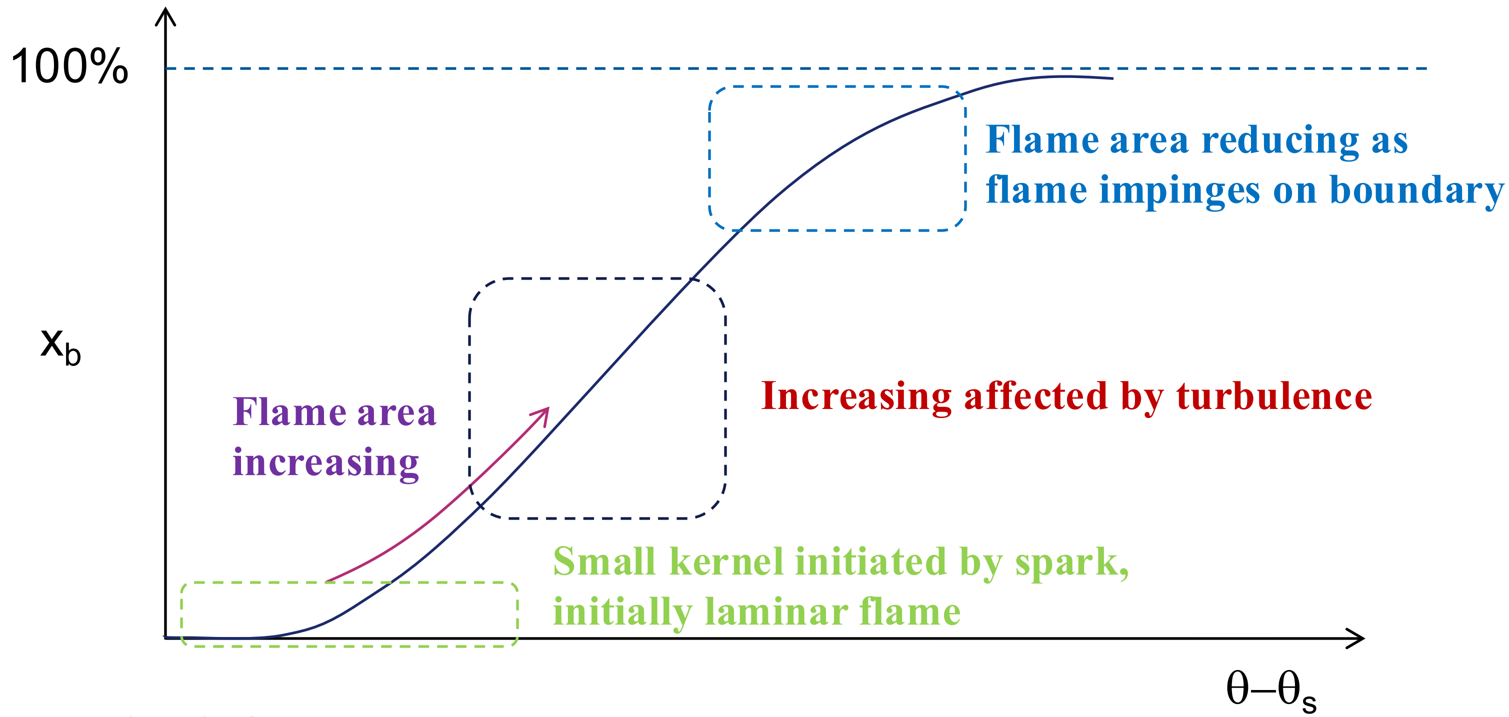




2 to 3: Flame area continues to grow due to flame propagation. At location 3, the flame area is at its maximum.

3 to 4: Flame area is being reduced because it has reached the combustion chamber walls at many locations. At location 4, the burning rate is being retarded by heat loss from the flame at the combustion chamber walls (flame quenching).

Mass Fraction curves



Factors which influence rate and shape

Mixture composition and state – (affect laminar flame)
dilution (EGR), AFR, P and T

Difference in density between burned and unburned gas in combustion chamber
Pressure is the same at any instant but product of combustion at higher T

Turbulence – small scale turbulence greatly increases flame speed (several times higher than laminar speed)

Shape of combustion chamber and position of spark plug

Combustion in SI engines

A functional form (Wiebe function) often used to represent mass fraction burned versus crank angle

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^n\right]$$

x_b Mass fraction burned

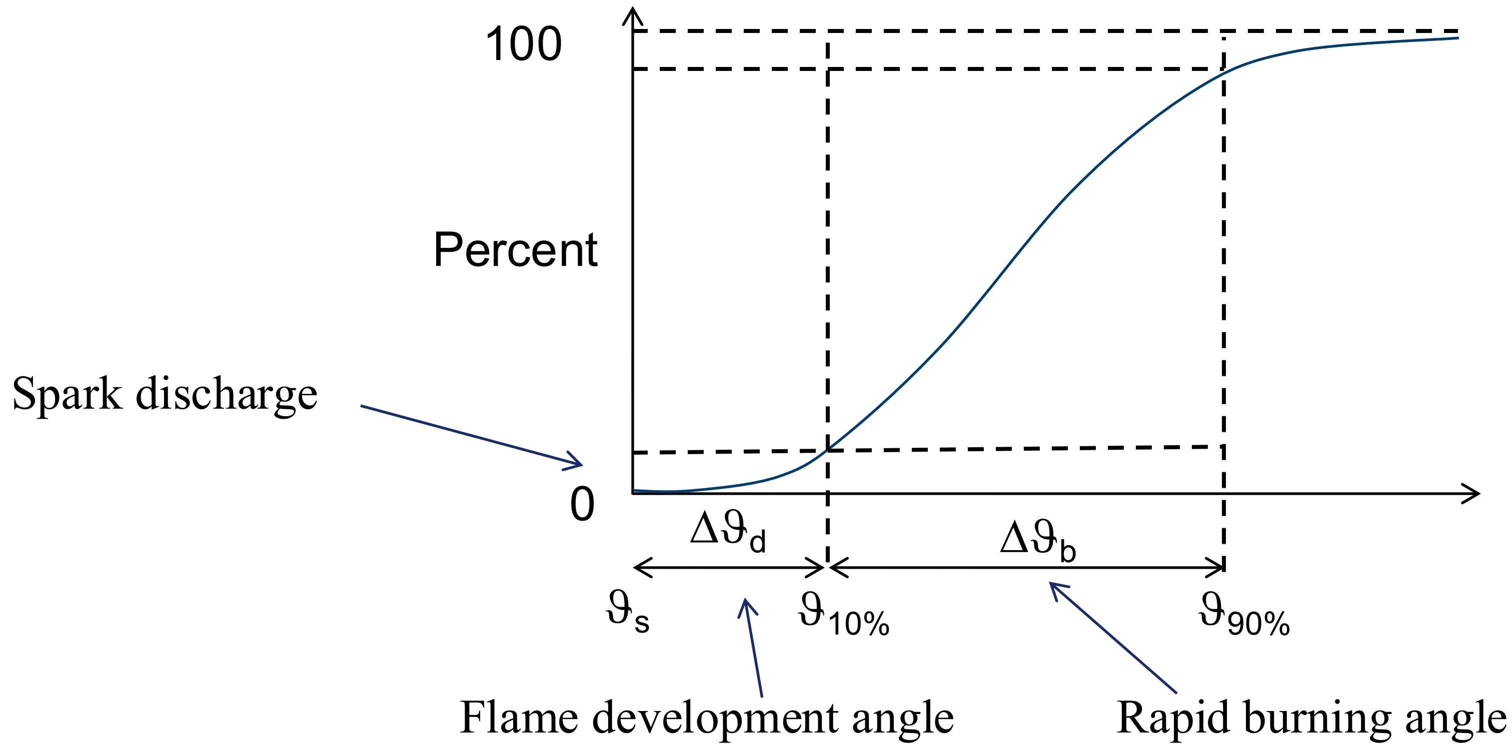
θ_s Spark timing

θ_d Crank angle value over which 90% of the mass is burned

a, n Constants

a and *n* can be chosen to vary the shape of the mass fraction burned curve. Typically *a=5* and *n=3*

Characterisation of combustion processes (SI)



Flame-development angle: CA interval between spark and the time when a small but significant fraction of cylinder mass has burned (a fraction of typically 10%)

Rapid-burning angle: CA interval required to burn the bulk of charge. Interval between the end of flame development stage and the end of flame-propagation process (typically 90%)

Relationship between x_b and volume fraction

Total mass and volume must be the sum of burned and unburned contributions

Volume fraction

$$y_b = \frac{V_b}{V_b + V_u}$$

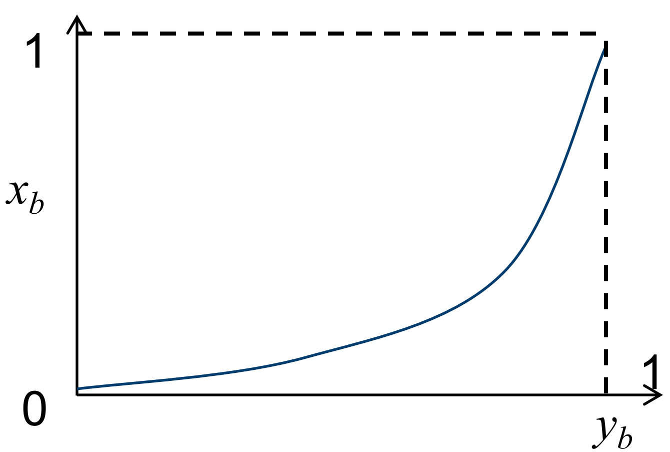
and perfect gas equation

$$x_b = 1 - \exp\left[1 + \frac{\rho_u}{\rho_b}\left(\frac{1}{y_b} - 1\right)\right]^{-1}$$

Temperature rise due to combustion is a function of fuel heating value and mixture composition. $\phi = 1, T_u = 700K, T_b = 2800K$ $\quad \frac{1}{\rho} \propto T$

$$\frac{\rho_u}{\rho_b} \cong 4$$

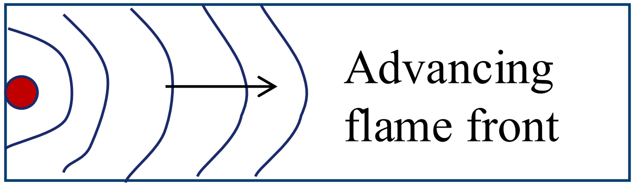
% of Volume occupied by burned gas increases more rapidly than x_b



Expansion mechanism leads to flame stretch – Increase in flame surface area

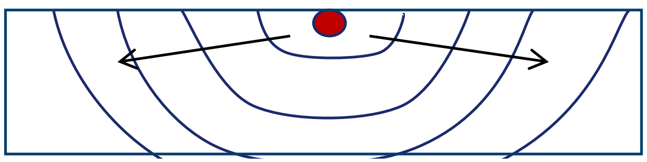
Unburned gas "compressed" ; increase its density and rate of charge burning

Effect of combustion chamber geometry on shape

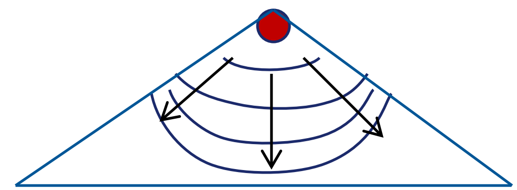


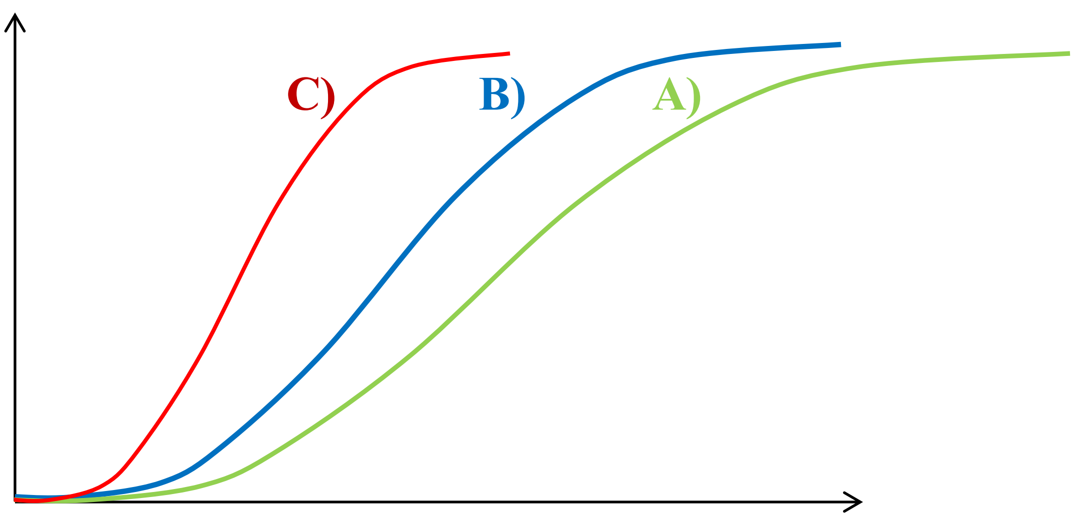
A) Disc shaped combustion chamber . Piston below, Cyl head above, spark plug at side

B) Disc shaped combustion chamber . Piston below, Cyl head and spark plug above



C) Pent-roof combustion chamber. Spherical flame front

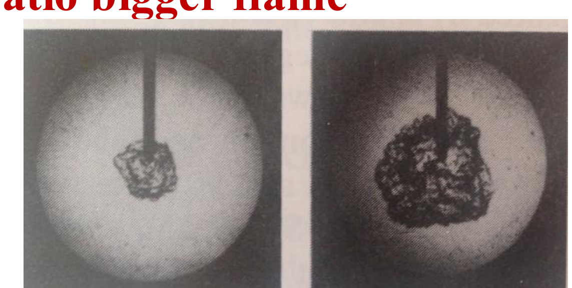




A) One flame front

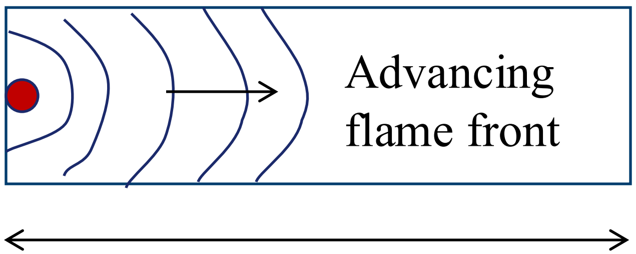
B) Flame front faster

C) Spherical flame front faster, steeper, larger volume to SA ratio bigger flame



Average flame travel speed

Assume we have a disc shaped combustion chamber. Spark plug at side



Distance travelled: Bore [m]

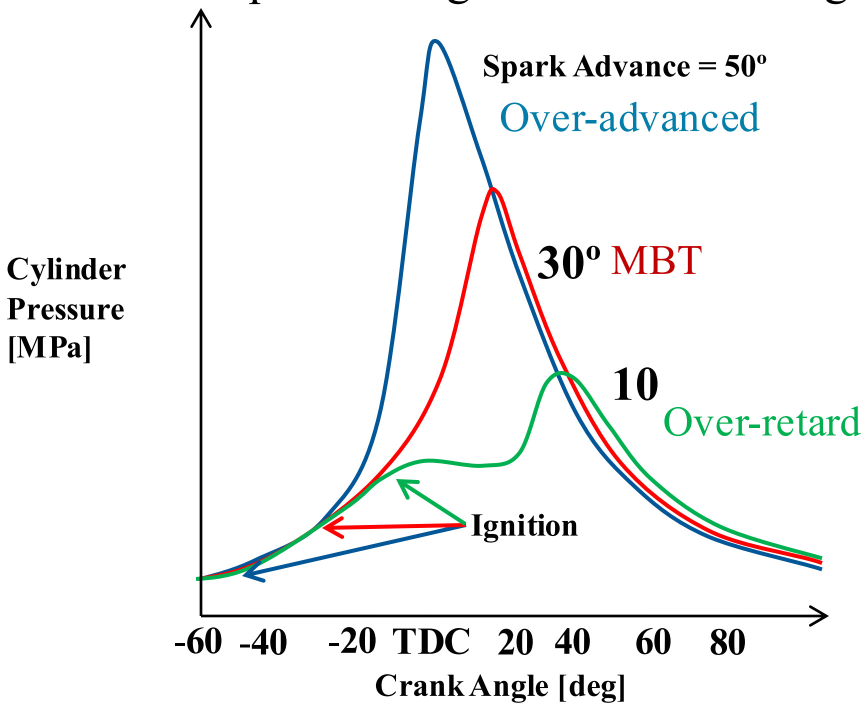
$$Speed = \frac{Distance}{Time}$$

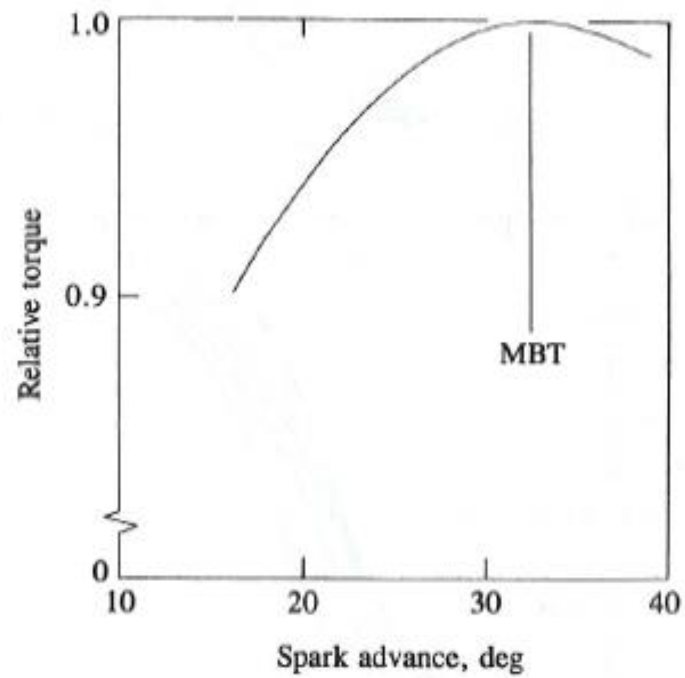
Time = ? Average flame; Use the rapid burning angle $\Delta\vartheta_b$

$$Speed = \frac{Bore}{\left(\frac{\Delta\vartheta_b}{360^o}\cdot\frac{60}{Engine\ \ Speed}\right)} = [m/s]$$

Engine torque varies as spark timing varies

P vs CA spark timing effects at fixed engine speed and intake manifold conditions





Spark too late – over retarded

fuel energy release occurs far into the expansion which leads to lower pressure rise, hence potential work cannot be fully extracted.

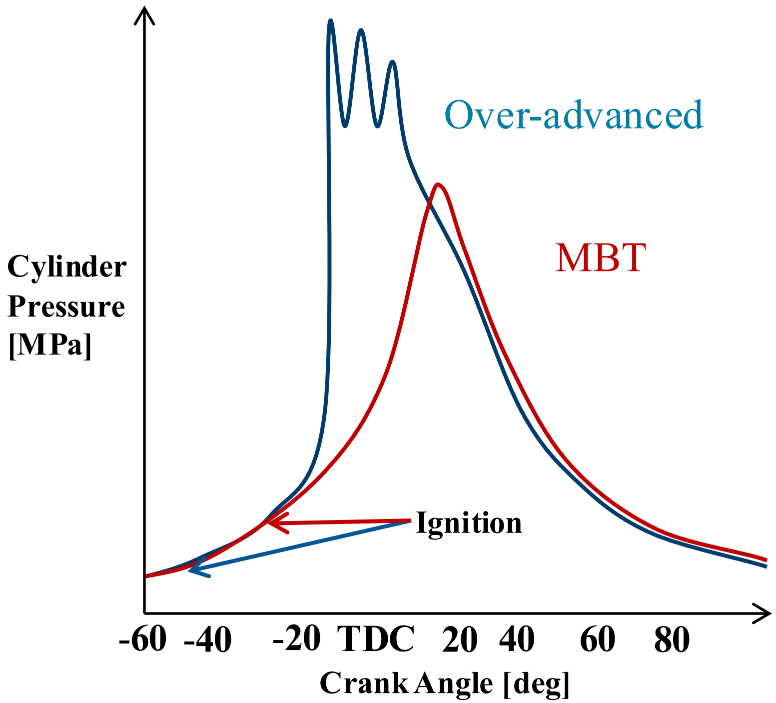
Spark too early – Over advanced

pressure rises too high before TDC, causing large negative compression work, high mechanical stress and increased heat transfer losses

Optimal timing - MBT

Maximum brake torque (MBT) timing is the spark timing that maximises torque output at constant airflow – varies with speed, load, temperature, EGR and engine burn rate

Spark timing fixes combustion phasing



MBT

No benefit gained from advancing spark beyond MBT

Spark advance is spark timing in crank angle degrees before TDC

Spark advance may be limited by Knock

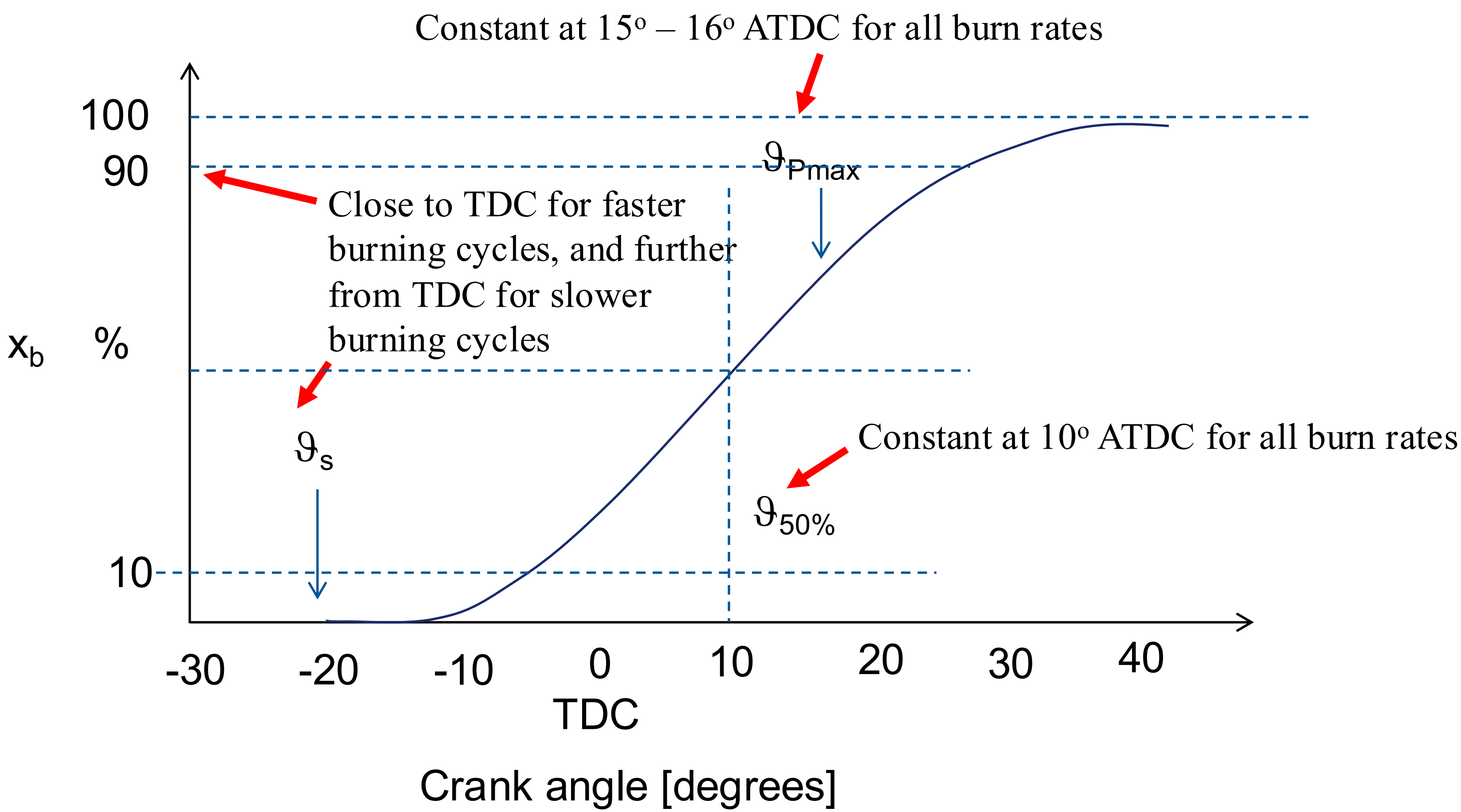
Over advanced - Likely to produce knock

Knock: uncontrolled combustion of end-gas causing pressure wave which propagates to and fro across the combustion chamber

End gas – the last part of the unburned charge in front of the flame.

Characterisation of combustion processes (SI)

At MBT timing,



Optimum phasing of combustion burns 50% of the charge by a crank angle of 10°ATDC

Example pag 59

At MBT spark timing, the crank angle by which 50% of the charge is burned is typically 10-12^{o} ATDC. If the mass fraction burned variation is described by the Wiebe function with a=5 and n=3, calculate MBT spark timing if the 0-90% burn duration θ_d=60^{o}

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^n\right]$$

x_b= mass fraction burned
θ_s= spark timing
θ_d= burn duration (0-90% of charge)

a=5 and n=3; θ_d=60^{o}

Assume optimum phasing of combustion burns 50% of the charge by a crank angle of 10°ATDC

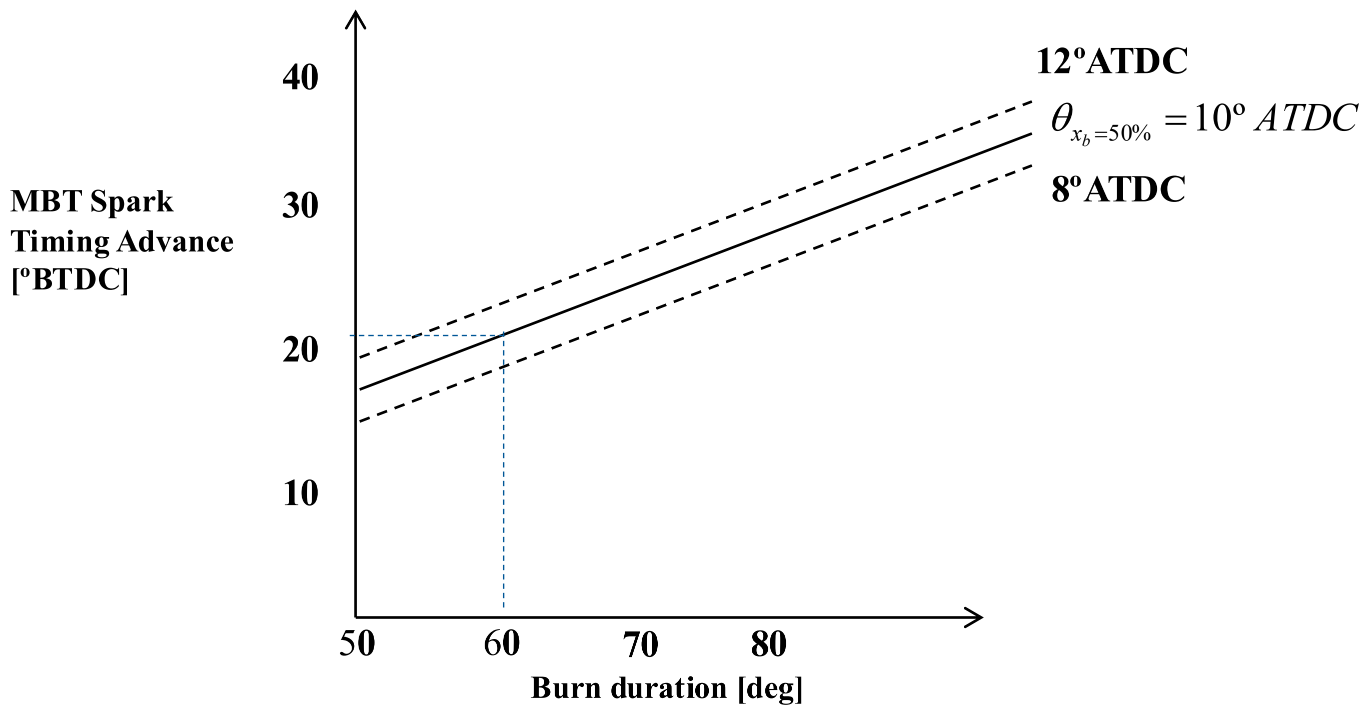
$$\theta_s = \theta - \theta_d \sqrt[n]{\frac{1}{a}\ln\left[\frac{1}{(1 - x_b)}\right]}$$

Example pag 59

$$\theta_s = 10 - 60 \times \sqrt[3]{\frac{1}{5}\ln\left[\frac{1}{(1-0.5)}\right]} =$$

$$\theta_s = 10 - 60 \times \sqrt[3]{0.1386} = 10 - 60 \times 0.5175 = -21^{\circ}\ ATDC$$

Spark advance is 21ºBTDC



Summary

We discussed:

- Combustion reaction equations
- Heat release equation
- Combustion in SI engines
- Engine torque and spark timing
- Maximum brake torque (MBT) timing
- Worked example

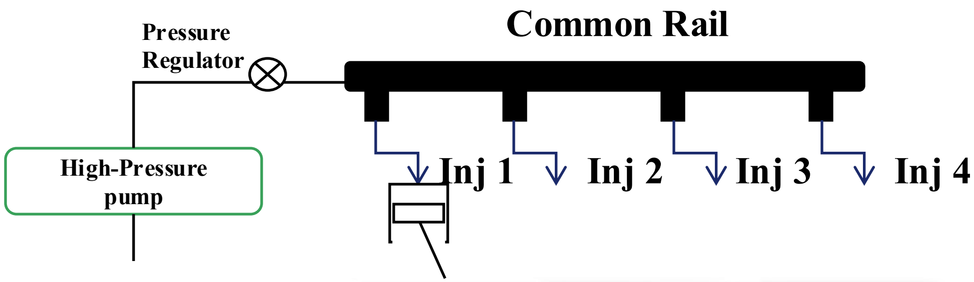
content

- Fundamentals of Combustion in CI engines
- Analyse critically the Heat Release Rate & Ignition delay
- Apply fundamental knowledge to an engine problem (Worked example)
- Analyse current industry design trends: Pilot injections/ Unconventional strategies
- Appraise engineering design solutions using appropriate level of explanation. Past exam papers

Combustion in CI engines

CI combustion is of fuel directly injected in Cyl. Heterogeneous mixture ignited by auto-ignition

Modern diesels: DI injection – common rail



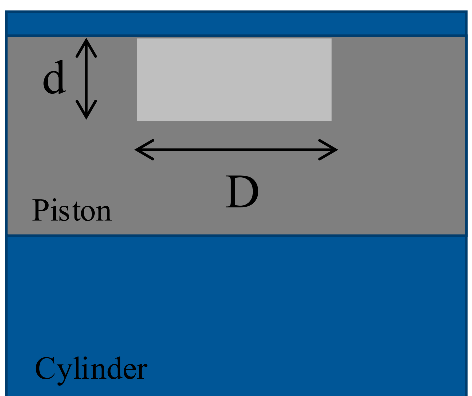
In-cylinder Air (+ EGR & residuals) compressed

High enough P, T to cause auto-ignition when fuel is injected

Bowl-in-piston design

E.g. for calculations:

Bowl in piston of cylindrical shape and has a diameter of "D" and a depth "d"



http://www.youtube.com/watch?v=7tA52k2CB4E

Characteristics of diesel combustion

Processes

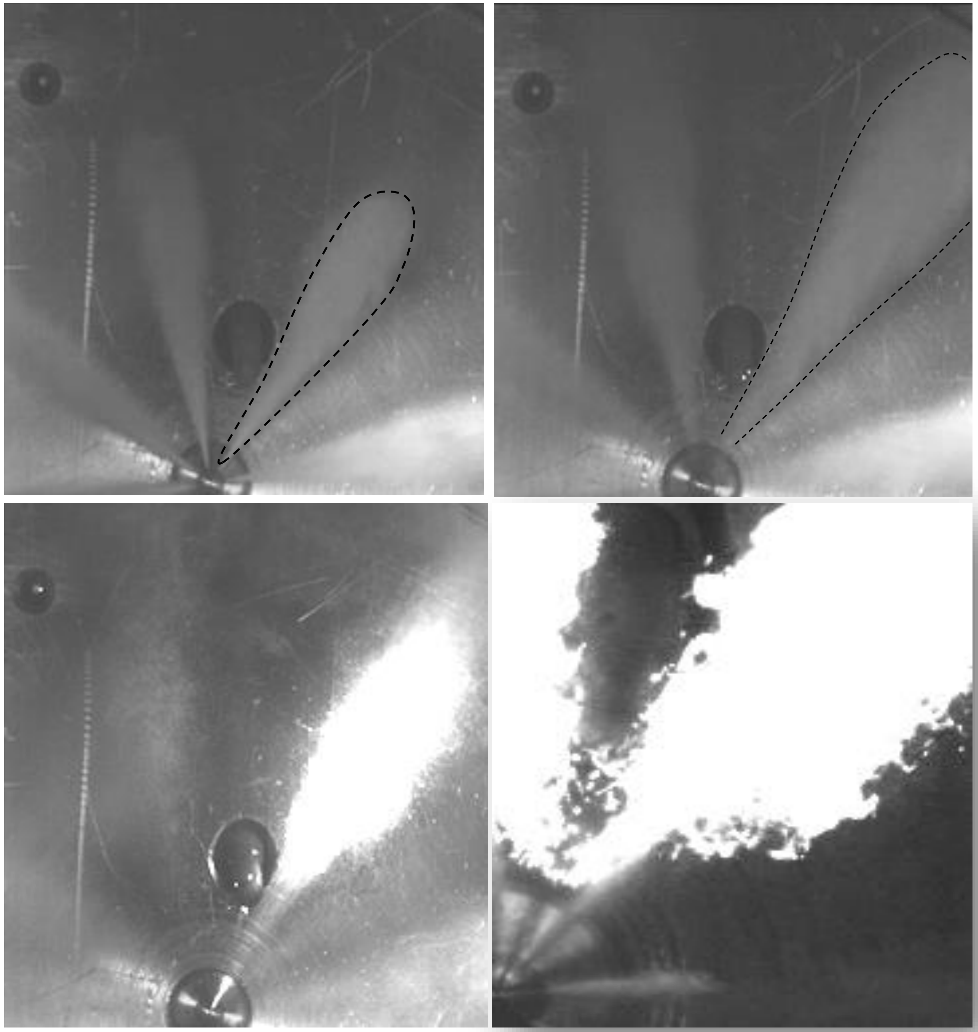
a) Liquid fuel injected into compressed charge

b) Fuel atomises/evaporates and mixes with hot air (physical delay)

c) Fuel does not ignite immediately chemical delay (pre-combustion reactions) (Ignition delay in diesel is typically 2-5ºCA)

d) Auto-ignition with rapid burning of fuel-air that is 'premixed' during ignition delay period

e) This is followed by diffusion burning as the fuel and air mix – controlled burning

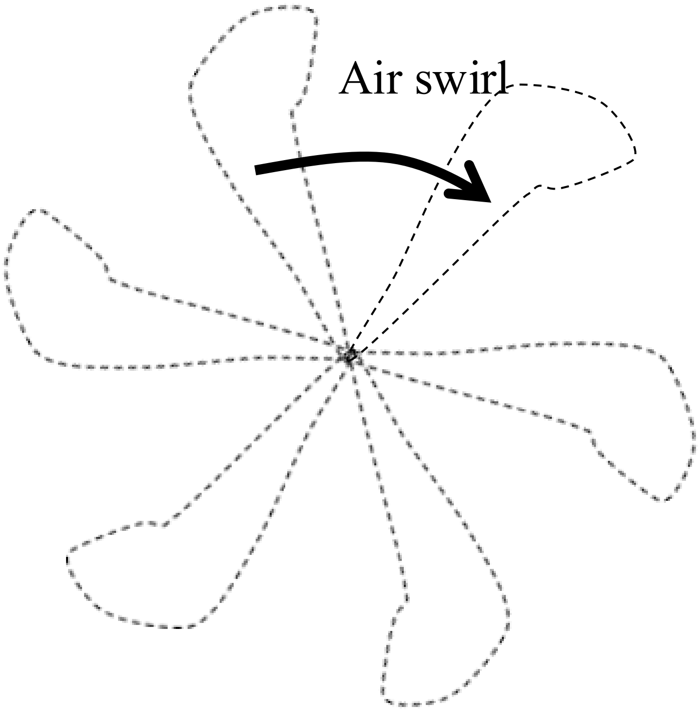


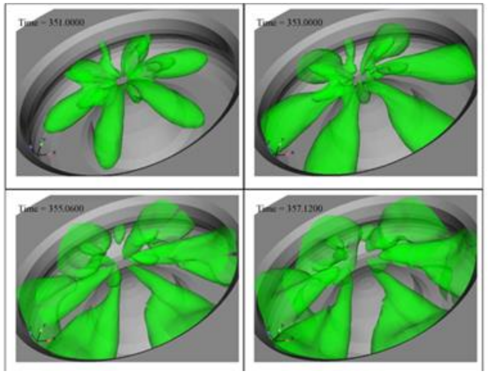
Optical Study of Diesel Combustion
(Quiescent chamber low temperature)
Engine Research Group University of Nottingham

Characteristics of diesel combustion

Fuel spray injected radially outward from the injector axis into a quiescent chamber

Fuel spray injected radially outward from the chamber axis into a swirling air flow





Fuel-air mixing

Fuel jet momentum / wall interaction has a larger influence on the early part of the combustion process.

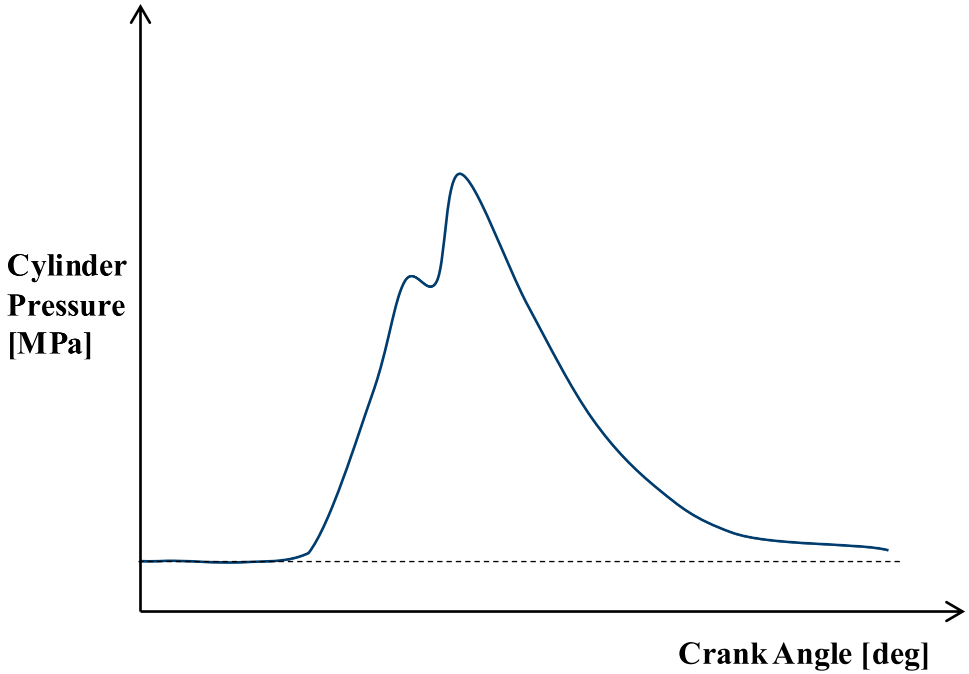
Charge motion (swirl) has a larger influence on the later part of the combustion process.

Heat Release Rate

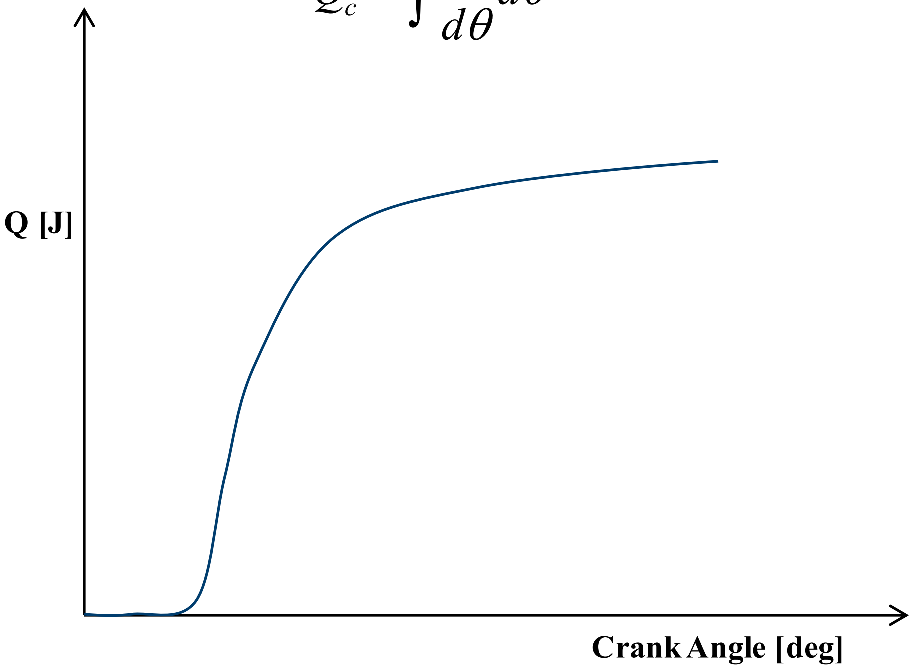
Net heat release function of change of cylinder volume and pressure

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta}$$

Common way to present HR is as cumulative HR variation or HRR

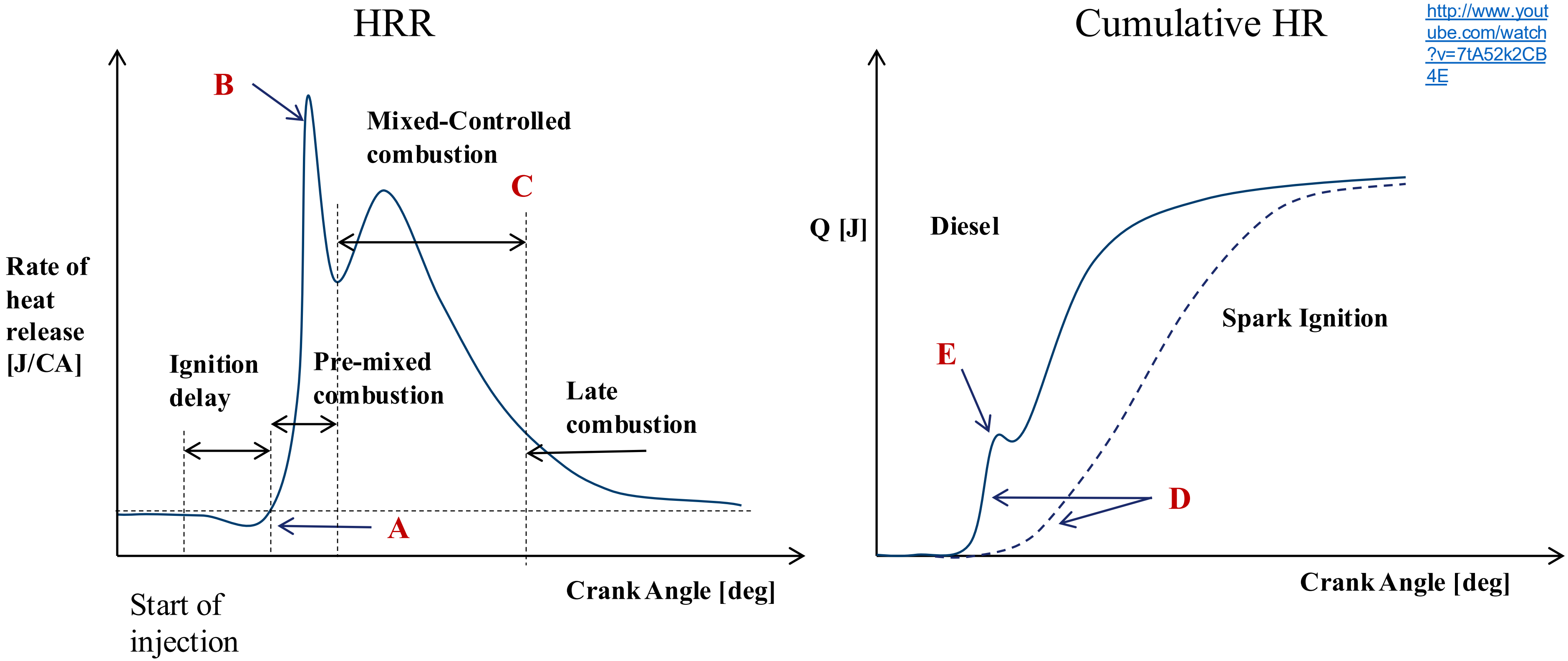


$$Q_c = \int \frac{dQ}{d\theta} d\theta$$



Cumulative HR equivalent of mass fraction burned for SI

Combustion characteristic and cumulative HR



A – Heat release appear –ve because fuel vaporisation lowers temperature

B – Peak heat release can be reached during pre-mixed phase, but less obvious than this at some operating conditions

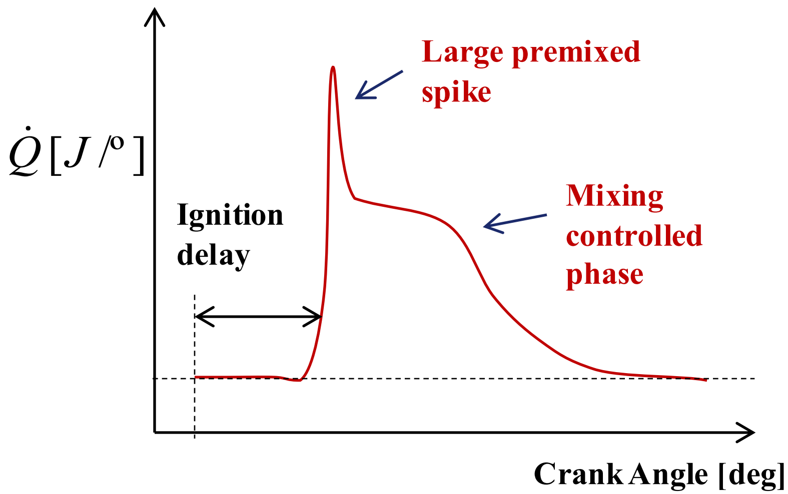
C – Arbitrary boundary

D – SI flame grows from a point source. Diesel multi-spray allows high rate of heat release after ignition delay

E – Endo of pre-mixed, start of mixing controlled phase

Worked example

For a DI diesel engine, sketch the typical heat release variation for relatively short and long ignition delay conditions, respectively showing the effect of premixed and mixing-controlled combustion phases [6]

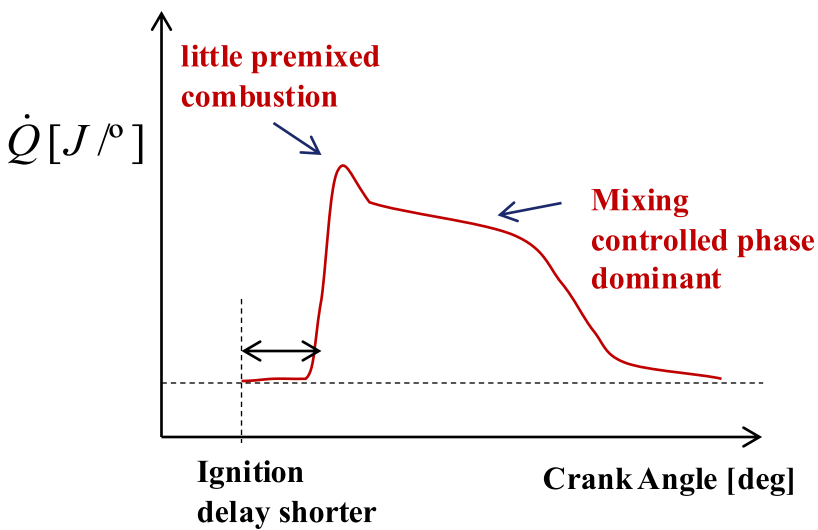


Long ignition delay

More time for mixture prep – Larger premix spike

Short ignition delay

Less time for mixture prep – little premix combustion

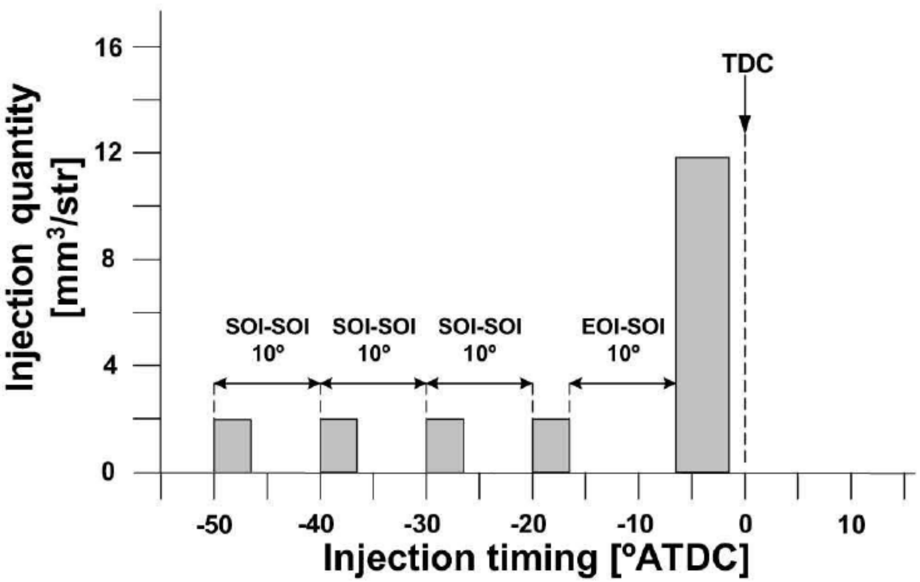


Pilot injection

Electronically controlled common rail fuel injection systems allow for the possibility of rate shaping and Pilot Injections

Pilot injection: small injection (1-2mg) before main injection (5-60mg)

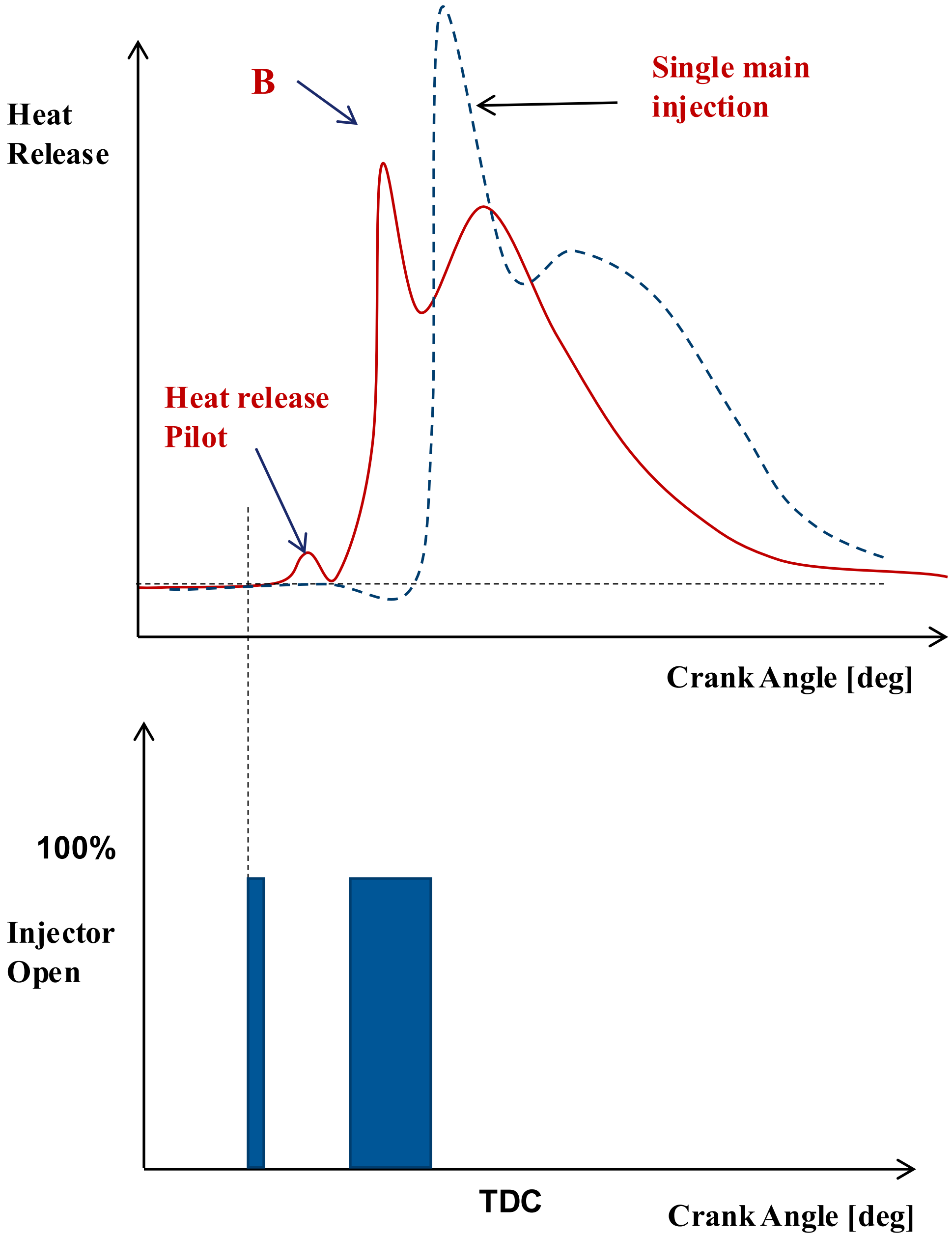
- Shortens ignition delay of main (premix combustion)
- Limits R of HR (lower noise)
- Assists/ Improves combustion stability



Pilot injection is usually injected few degrees in advance of the main

New common rail fuel injection systems allow the fuel to be injected in up to six closely spaced injections

Pilot injection



Injection of Pilot

Heat release from pilot assists combustion of main

Too early in Compression stroke

- won't auto ignite until required P and T reached
- may diffuse to "fumigate" charge (not so useful as pilot HR in promoting reactions in main)

Too late in Compression stroke

- pilot does not burn in advance of main (becomes part of the main)

Should be injected 20ºCA<pilot<5ºCA

Example pag 54

A cylinder receives 60mg/stroke of diesel fuel which burns with a combustion efficiency of 98%. The start of combustion occurs just before piston TDC crank position, i.e. before 0°ATDC . The combustion period is 40 crank angle degrees. Calculate the average gross rate of heat release if the lower heating value of the fuel is 42MJ/kg. If the net rate is 80% of the gross rate, calculate the rate of pressure rise at 0°ATDC using the average net rate of heat release if the combustion chamber volume is 35cc and γ is 1.4

Fuel supplied is 60mg/stroke

Combustion efficiency 98%

$$\frac{dQ_n}{d\vartheta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\vartheta} + \frac{1}{\gamma-1} V \frac{dp}{d\vartheta}$$

Gross heat released

$$Q_g = \eta_c m_f Q_{LHV} \quad = 0.98 \times 60 \times 10^{-6} \times 42 \times 10^{6} \quad = 2470J$$

Net heat released

$$Q_n = 80\% Q_g \quad = 0.8 \times 2470 \quad = 1976J$$

Example pag 54

Average rates:

$$\left(\frac{dQ_g}{d\vartheta}\right)_{Average} = \frac{2470}{40} = 61.7\frac{\text{J}}{\text{degree}}$$

$$\left(\frac{dQ_n}{d\vartheta}\right)_{Average} = 0.8\times 61.7 = 49.4\frac{\text{J}}{\text{degree}}$$

At TDC

$$\frac{dV}{d\vartheta} = 0 \qquad \frac{dQ_n}{d\vartheta} = \frac{\gamma}{\gamma-1}p\cancelto{0}{\frac{dV}{d\vartheta}} + \frac{1}{\gamma-1}V\frac{dp}{d\vartheta}$$

$$\frac{dp}{d\vartheta} = \frac{\gamma-1}{V}\left(\frac{dQ_n}{d\vartheta}\right)_{Average} = \frac{1.4-1}{35\times 10^{-6}}\times 49.4 = 5.6\frac{\text{bar}}{\text{degree}}$$