INTAKE AND EXHAUST PROCESS, COMBUSTION AND KNOCKING IN SI ENGINE

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Gas exchange Processes

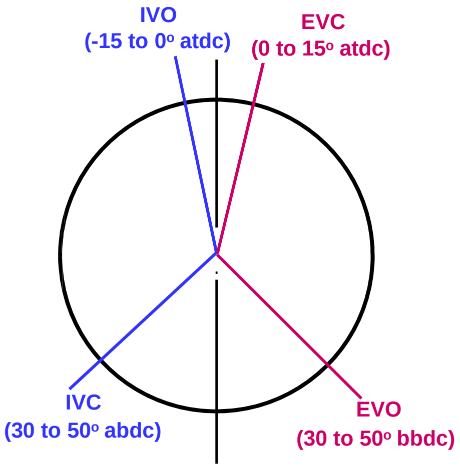
To move working fluid in and out of engine

- Engine performance is air limited
- Engines are usually optimized for maximum power at high speed

Considerations

- 4-stroke engine: volumetric efficiency
- 2-stroke engine: scavenging/ trapping efficiency
- Charge motion control; tuning; noise

Typical valve timing diagram



- Early EVO
  - Facilitates exhaust gas outflow via blow down
  - Incomplete expansion
- Late IVC
  - High speed: ram effect augments induction
  - Low speed: air loss by displacement flow
  - Lower effective compression ratio

Note that for typical passenger car engine, max piston speed is at ~70° from TDC

VVT technology –cam shifter

Toyota VVT-i
(SAE Paper 960579)

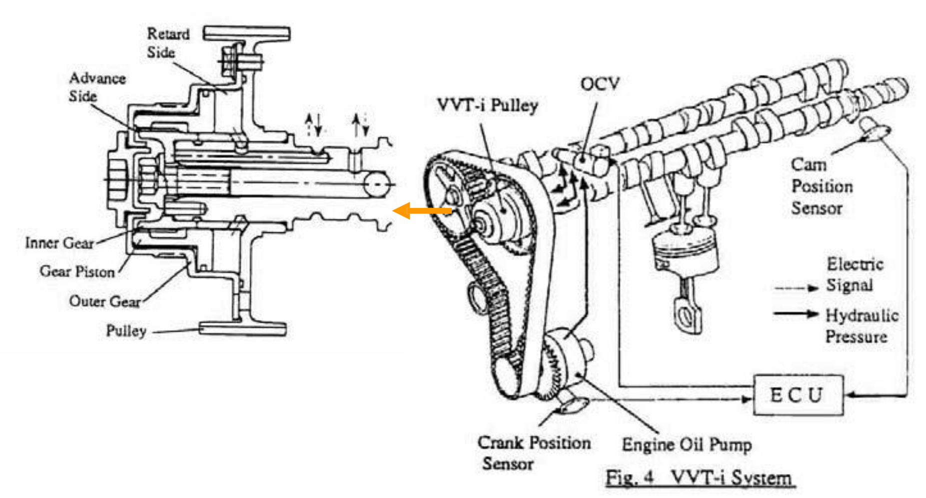


Fig. 4 VVT-i System

Volumetric efficiency: quasi-static effects

- Residual gas
  - Affected by:
    - ⤢Compression ratio
    - ⤢Exhaust gas temperature
    - ⤢Exhaust to intake pressure ratio
  - Impact:
    - ⤢Volumetric efficiency
    - ⤢Charge composition
    - ⤢Charge temperature

Volumetric efficiency: quasi-static effects
(cont.)

- Evaporative cooling effect
  - Higher charge density increases volumetric efficiency
  - Adiabatic evaporation in air to form λ=1 mixture:
    - Iso-octane: ΔT = -19°C
    - Ethanol: ΔT = -80°C
    - Methanol: ΔT = -128°C

  From both higher latent heat, lower LHV, and lower stoichiometric air/fuel ratio (Ethanol, Methanol)

  - In practice, most heat from the wall unless direct injection is used

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Volumetric efficiency: quasi-static effects
(cont.)

- Air displacement by fuel and water vapor

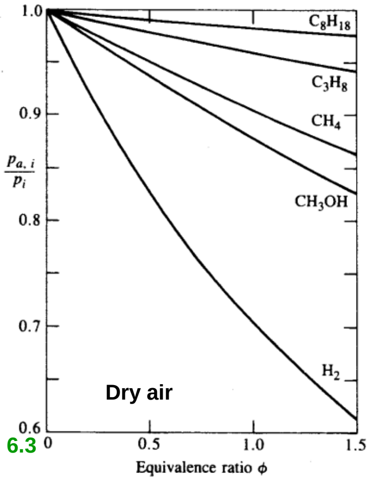


Fig. 6.3

V_i is volume inducted

P_i is intake pressure

$$m_a = \left(\frac{P_i V_i}{\tilde{R}T}\right)\tilde{x}_a W_a$$

$$\tilde{x}_a + \tilde{x}_f + \tilde{x}_w = 1$$

$$\tilde{x}_a = \frac{1}{1+\frac{\tilde{x}_f}{\tilde{x}_a}+\frac{\tilde{x}_w}{\tilde{x}_a}}$$

$$= \frac{1}{1+\frac{m_f}{m_a}\frac{W_a}{W_f}+\frac{\tilde{x}_w}{\tilde{x}_a}}$$

Volumetric Efficiency: dynamic effects

Friction

- Component i pressure drop due to friction:
  - Vi = Fluid velocity
  - ξ_i = Loss coefficient

$$\Delta P_i = \xi_i \rho v_i^2$$

Scaling:

$$v_i \propto S_P \frac{A_P}{A_i}; \qquad \xi_i \propto \frac{\ell}{D_i}$$

$$\Delta P_i \sim \rho S_P^2 \frac{1}{A_i^{2.5}} \text{ or } \propto \rho S_P^2 \frac{1}{D_i^5}$$

Flow loss in gas exchange process

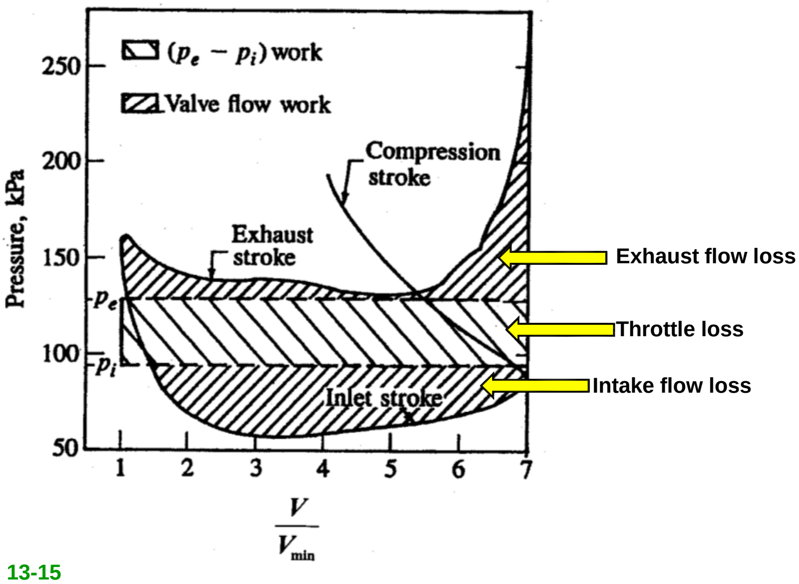


Fig. 13-15

Volumetric Efficiency: dynamic effects cont.

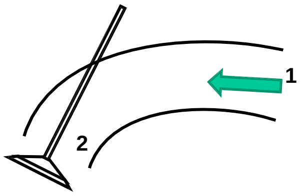
Ram effect

- Due to fluid inertia
- Intake and exhaust flow both exhibit effect

$$\Delta P = p_2 - p_1 = -\rho \int \frac{du}{dt} \cdot d\ell$$

$$\approx \rho \frac{S_P}{\frac{1}{2N}} \left(\frac{A_P}{A_{intake}} \right) \ell$$

$$= \rho S_P^2 \left(\frac{A_P}{A_{intake}} \right) \frac{\ell}{L}$$

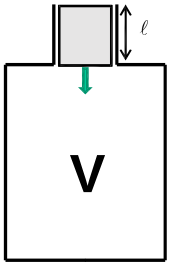


S_P	**Mean piston speed**
ℓ	**Runner length**
L	**Stroke**

Volumetric Efficiency: dynamic effects cont.

Tuning

- Helmholtz frequency $N = \frac{a}{2\pi}\sqrt{\frac{A}{\ell V}}$



- a sound velocity
- ℓ runner length
- V volume

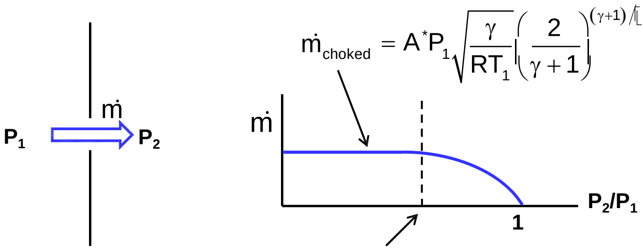
- Application:
  - V taken as $V_t/2$
  - Correction factor k=2

$$N = \frac{a}{2\pi}\sqrt{\frac{A}{\ell V}\frac{1}{K}}$$

Volumetric Efficiency: dynamic effects cont.

Choking effect

- Velocity becomes sonic at “throat”



$$\dot{m}_{choked} = A^{*}P_1\sqrt{\frac{\gamma}{RT_1}}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/[2(\gamma-1)]}$$

$$\left(\frac{P_2}{P_1}\right)_{critical} = \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)}$$

$$= 0.528 \text{ for } \gamma = 1.4; \text{ increases with } \gamma$$

Volumetric Efficiency: dynamic effects cont.

Overlap back flow

- Back flow of burned gas from exhaust/cylinder to intake port
- Increases residual gas fraction
- Prominent at low speed and load

Heat transfer

- Loss in η_v because intake charge is heated up by the hot walls
- Prominent at low speed because of longer time (overrides lower rate)

Volumetric efficiency: summary

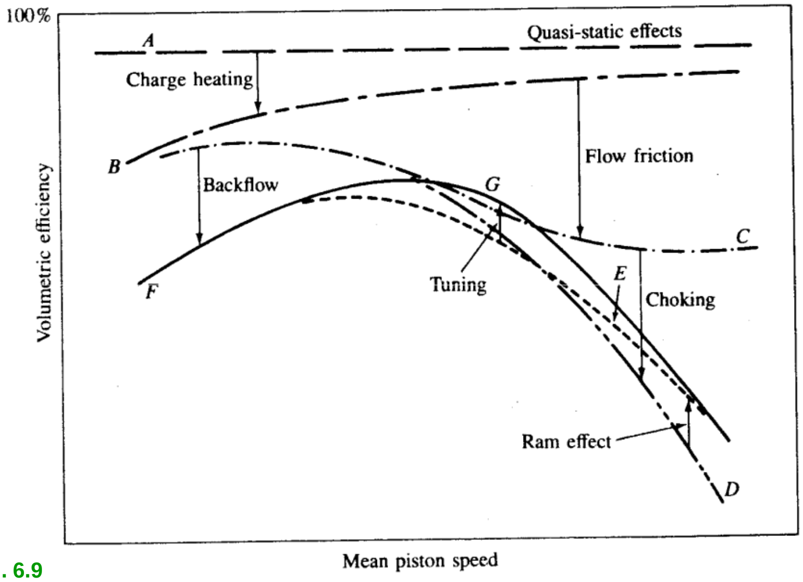


Fig. 6.9

2-Stroke engine gas exchange

Cross scavenging **Loop scavenging** **Uniflow scavenging**

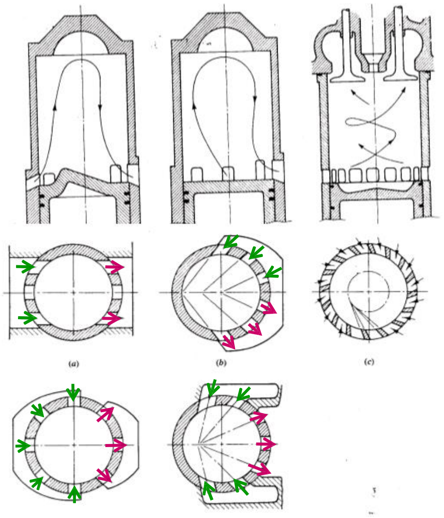
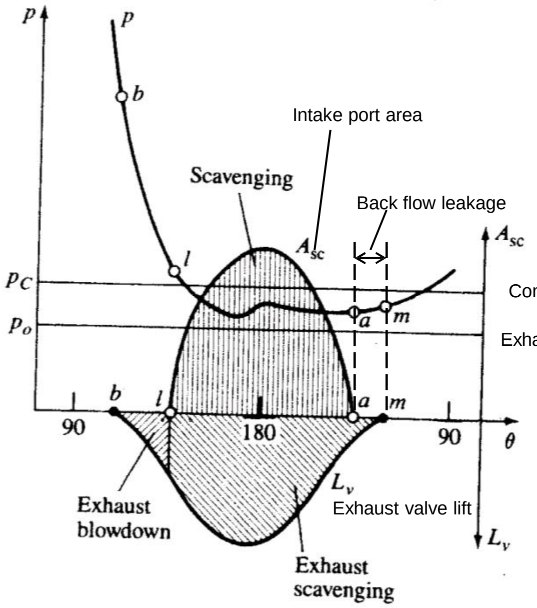


Fig. 6-23 & 24

Uniflow scavenging process



2-Stroke engine gas exchange

Delivery ratio $\Lambda = \dfrac{\text{Air mass delivered per cycle}}{\rho_{a,0} V_D}$

Trapping efficiency $\eta_t = \dfrac{\text{Air mass retained}}{\text{Air mass delivered}}$

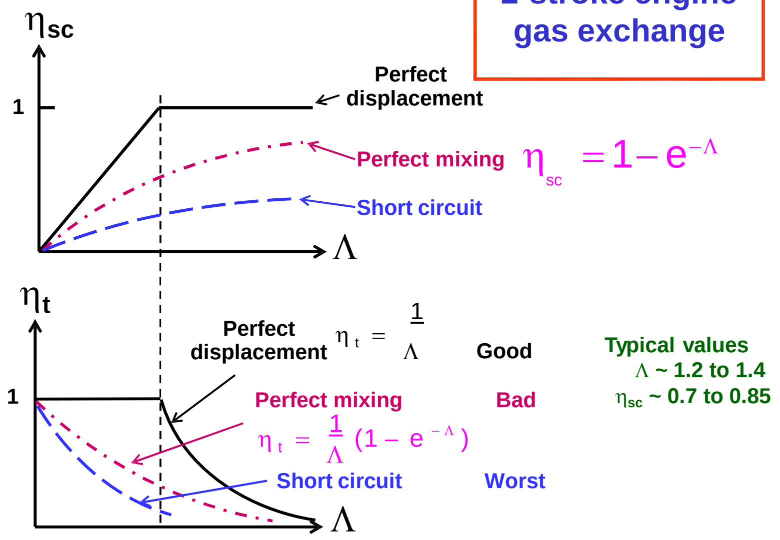
Air mass retained

$$m_a = \rho_{a,0} V_D \Lambda \eta_t$$

Scavenging ratio $\eta_{sc} = \dfrac{\text{Air mass retained}}{\text{Trapped charge mass}}$

$1 - \eta_{sc}$ is the fraction of previous cycle charge that remains

2-stroke engine gas exchange



SI engine combustion I

SI – engine combustion:
How to “burn” things?
Reactants → Products

Premixed

- Homogeneous reaction
  - Not limited by transport process
  - Fast/slow reactions compared with other time scale of interest
- Premixed flame
  - Examples: gas grill, SI engine combustion
- Detonation
  - Pressure wave driven reaction

Non-premixed

- Diffusion flame
  - Examples: candle, diesel engine combustion

SI ENGINE COMBUSTION

- Premixed flame
  - Laminar flame speed

- Turbulent enhancement of combustion
  - Wrinkled laminar flame

LAMINAR FLAME SPEEDS

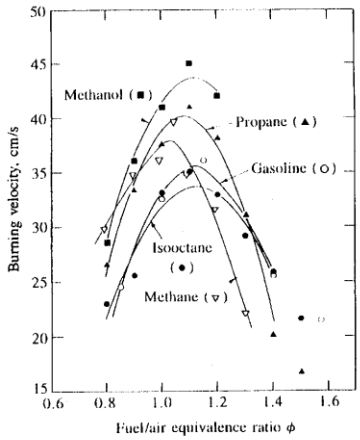


Fig. 9-25 Laminar burning velocity of several fuels as function of equivalence ratio, at 1 atm and 300 K.

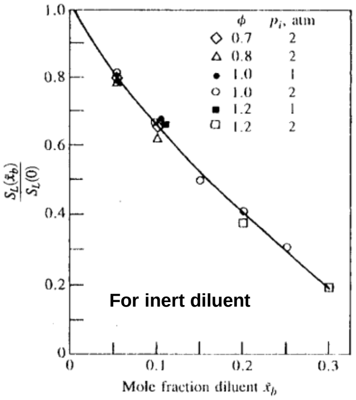
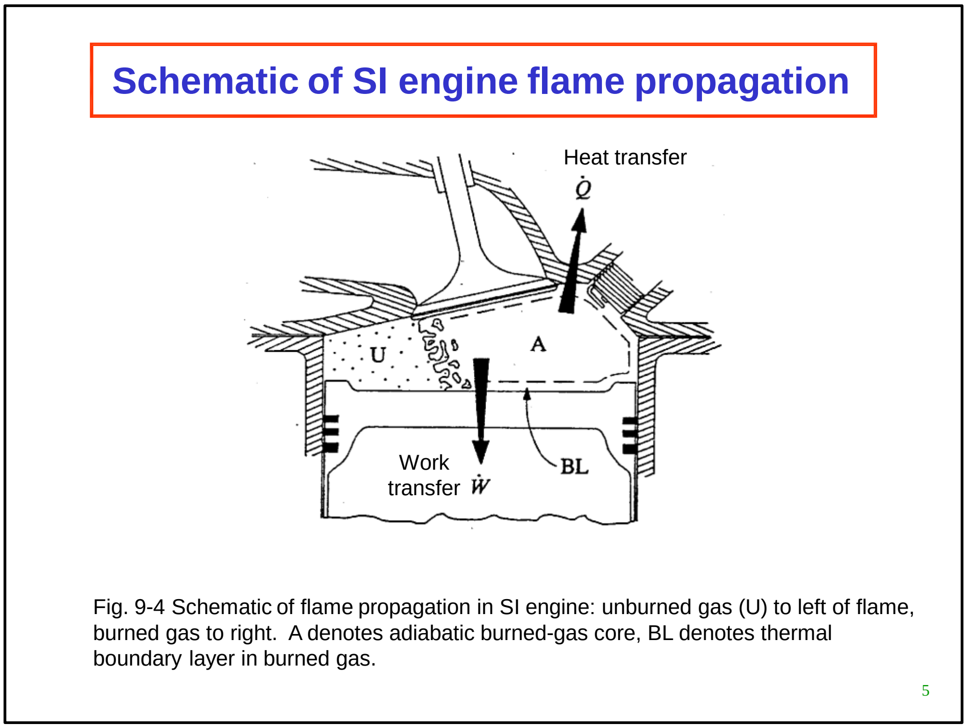
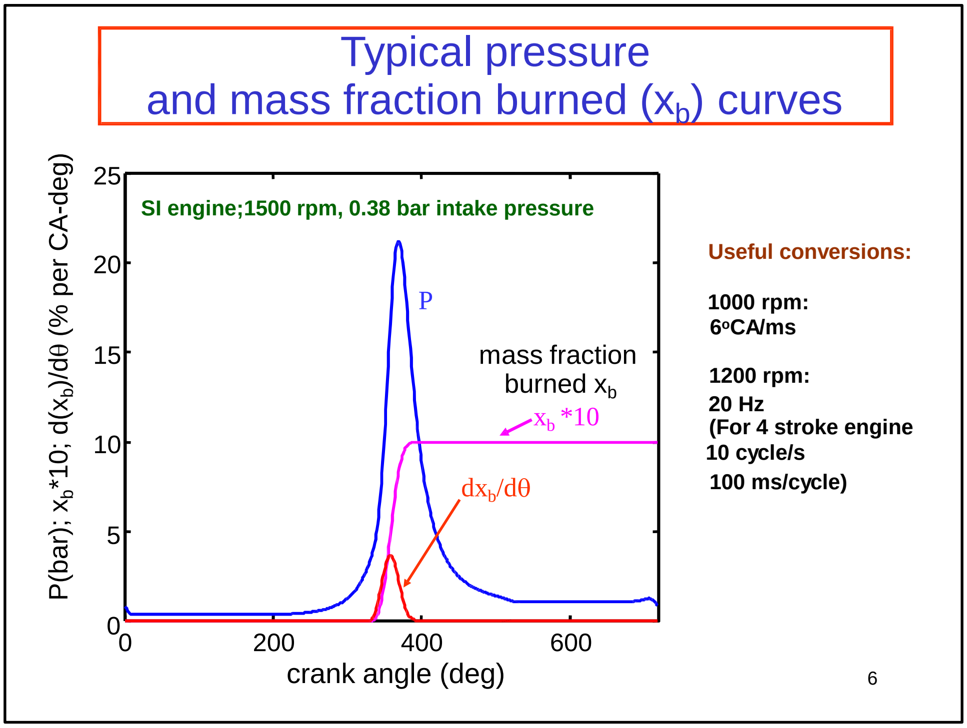


Fig. 9-26 Effect of burned gas mole fraction in unburned mixture on laminar burning velocity. Fuel: gasoline.
(Note that actual burned gas from non-stoichiometric combustion would render the charge Φ different from the metered Φ).



Schematic of SI engine flame propagation

Fig. 9-4 Schematic of flame propagation in SI engine: unburned gas (U) to left of flame, burned gas to right. A denotes adiabatic burned-gas core, BL denotes thermal boundary layer in burned gas.

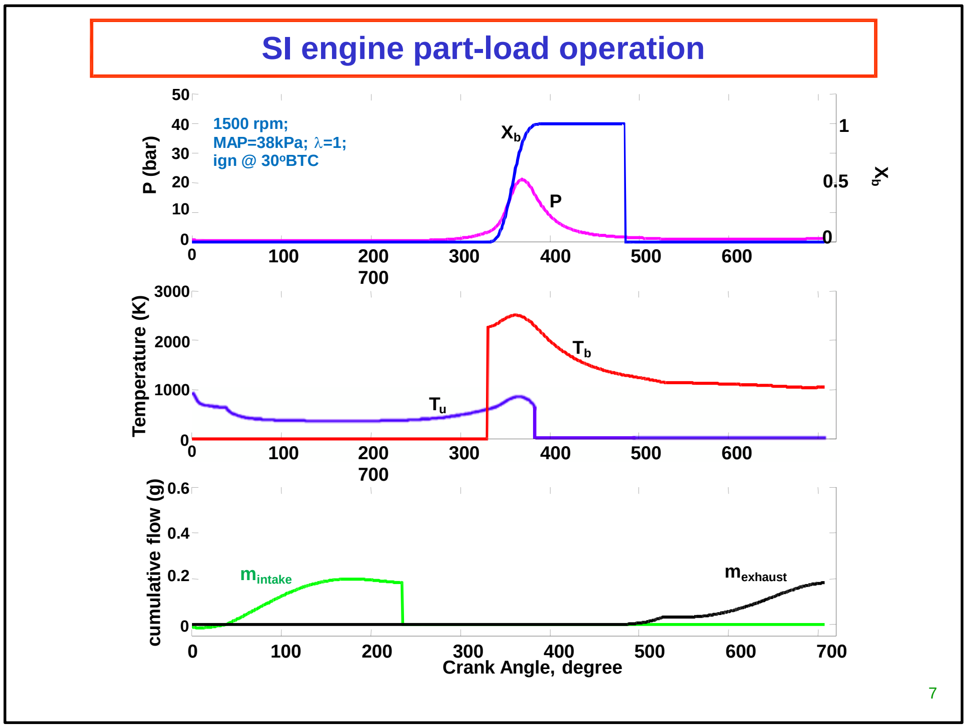


Typical pressure and mass fraction burned (x_b) curves

Useful conversions:

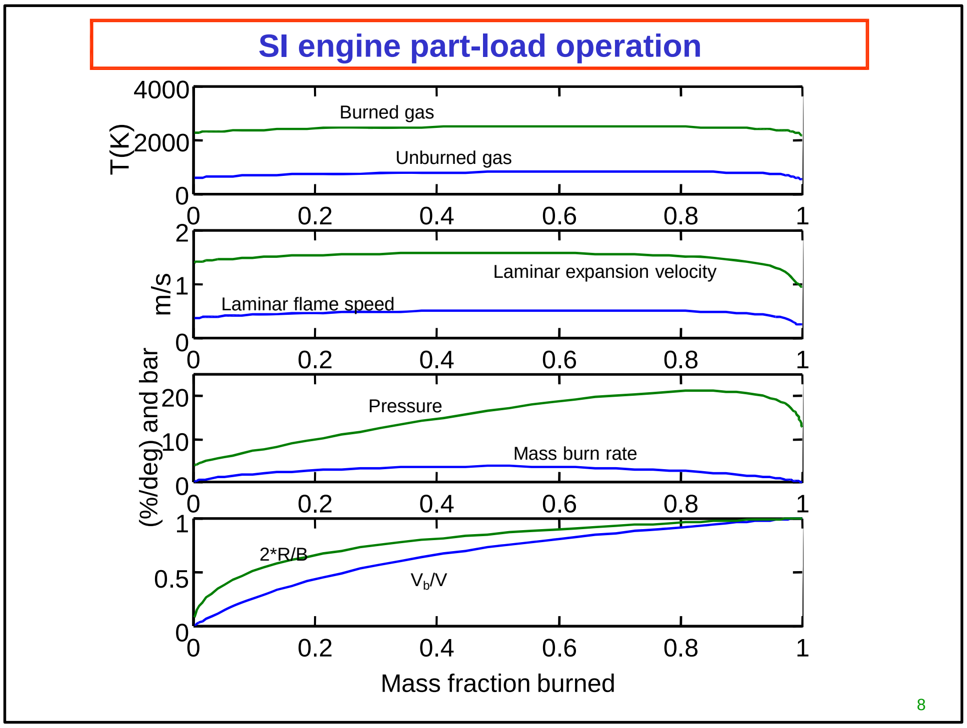
1000 rpm:
6°CA/ms

1200 rpm:
20 Hz
(For 4 stroke engine
10 cycle/s
100 ms/cycle)



SI engine part-load operation

7



SI engine part-load operation

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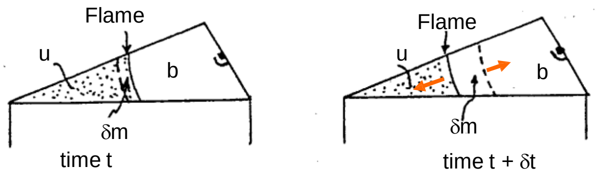
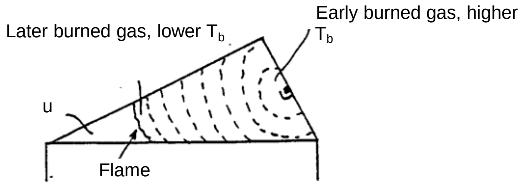
Combustion produced pressure rise



1. Pressure is uniform, changing with time
2. For mass δm: $h_b = h_u$ (because dm is allowed to expand against prevailing pressure)
3. T rise is a function of fuel heating value and mixture composition
    - e.g. at $\Phi = 1$, $T_u \sim 700$ K, $T_b \sim 2800$ K
4. Hence burned gas expands: $\rho_b \sim \frac{1}{4}\rho_u$; $\delta V_b \sim 4\ \delta V_u$

Combustion produced pressure rise

5. Since total volume is constrained. The pressure must rise by δp, and all the gas in the cylinder is compressed.
6. Both the unburned gas ahead of flame and burned gas behind the flame move away from the flame front
7. Both the unburned gas and burned gas temperatures rise due to the compression by the newly burned gas
8. Unburned gas state: since heat transfer is relatively small, the temperature is related to pressure by isentropic relationship
    - $T_u/T_{u,0} = (p/p_0)^{(\gamma_u - 1)/\gamma_u}$
9. Burned gas state:



Thermodynamic state of charge

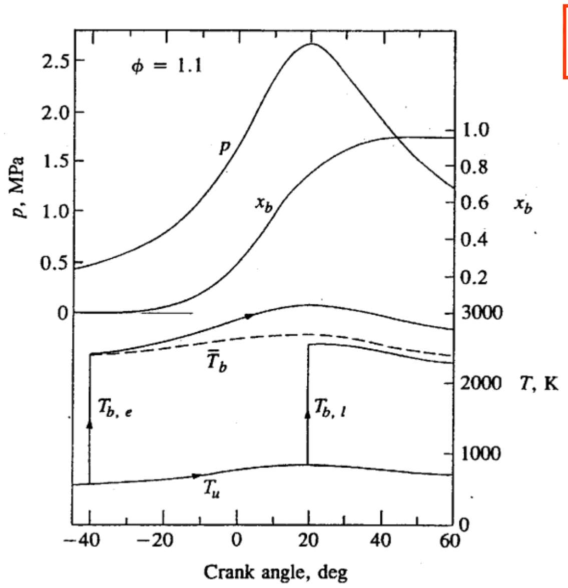


Fig. 9-5 Cylinder pressure, mass fraction burned, and gas temperatures as function of crank angle during combustion.

Burn duration

- Burn duration as CA-deg. : measure of burn progress in cycle
- For modern fast-burn engines under medium speed, part load condition:
  - $\Delta\theta_{0\text{-}10\%} \sim 15^{o}$
  - $\Delta\theta_{0\text{-}50\%} \sim 25^{o}$
  - $\Delta\theta_{0\text{-}90\%} \sim 35^{o}$
- As engine speed increases, burn duration as CA-deg. :
  - Increases because there is less time per CA-deg.
  - Decreases because combustion is faster due to higher turbulence
    - ⤢ Net effect: increases approximately as $\propto$ rpm$^{0.2}$

Optimum Combustion Phasing

- **Heat release schedule has to phase correctly with piston motion for optimal work extraction**
- **In SI engines, combustion phasing controlled by spark**
- **Spark too late**
  - **heat release occurs far into expansion and work cannot be fully extracted**
- **Spark too early**
  - **Effectively "lowers" compression ratio**
  - **increased heat transfer losses**
  - **Also likely to cause knock**
- **Optimal: Maximum Brake Torque (MBT) timing**
  - **MBT spark timing depends on speed, load, EGR, Φ, temperature, charge motion, …**
  - **Torque curve relatively flat: roughly 5 to 7°CA retard from MBT results in 1% loss in torque**

Spark timing effects

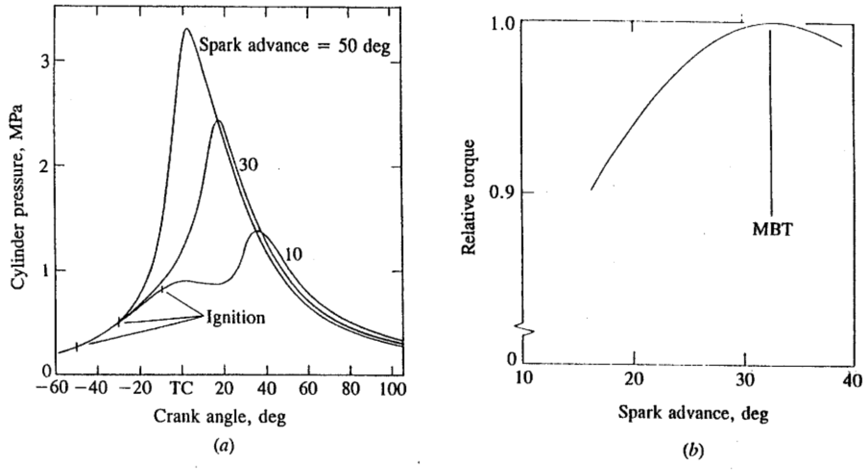


Fig. 9-3 (a) Cylinder pressure versus crank angle for overadvanced spark timing (50° BTDC), MBT timing (30° BTDC), and retarded timing (10° BTDC). (b) Effect of spark advance on brake torque at constant speed and A/F, at WOT

Control of spark timing

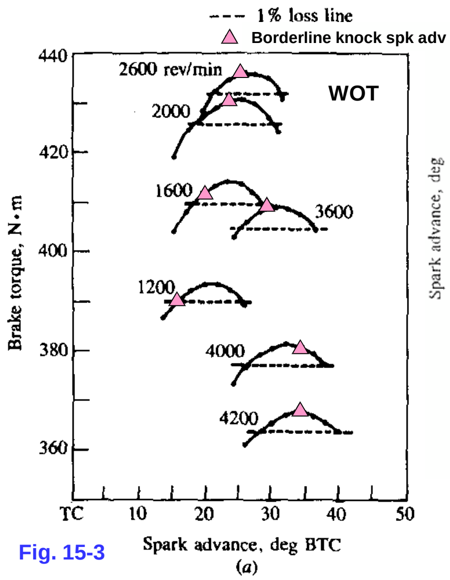


Fig. 15-3

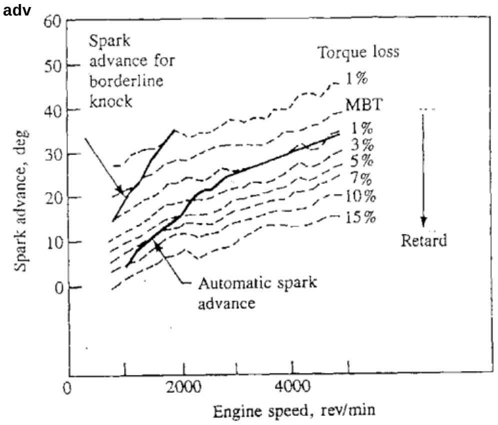
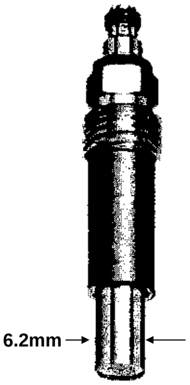


Fig. 15-17

Obtaining combustion information from engine cylinder pressure data

1. Cylinder pressure affected by:
   - a) Cylinder volume change
   - b) Fuel chemical energy release by combustion
   - c) Heat transfer to chamber walls
   - d) Crevice effects
   - e) Gas leakage
2. Obtaining accurate combustion rate information requires a) Accurate pressure data (and crank angle indexing) b) Models for phenomena a,c,d,e, above
   - c) Model for thermodynamic properties of cylinder contents
3. Available methods
   - a) Empirical methods (e.g. Rassweiler and Withrow SAE 800131)
   - b) Single-zone heat release or burn-rate model
   - c) Two-zone (burned/unburned) combustion model

Typical piezoelectric pressure transducer spec.



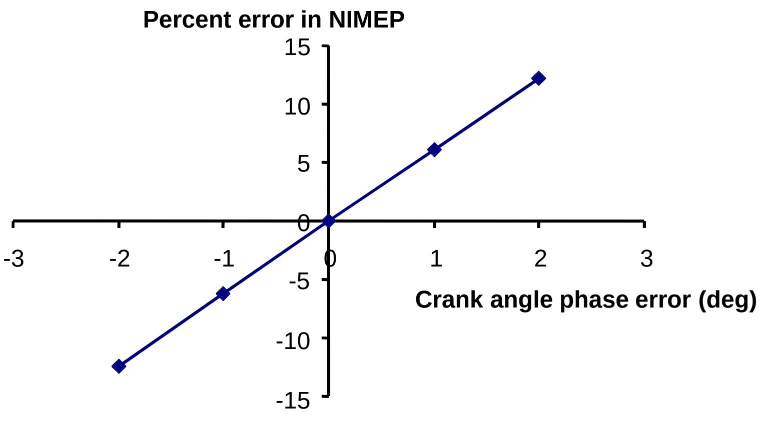
Kistler 6125

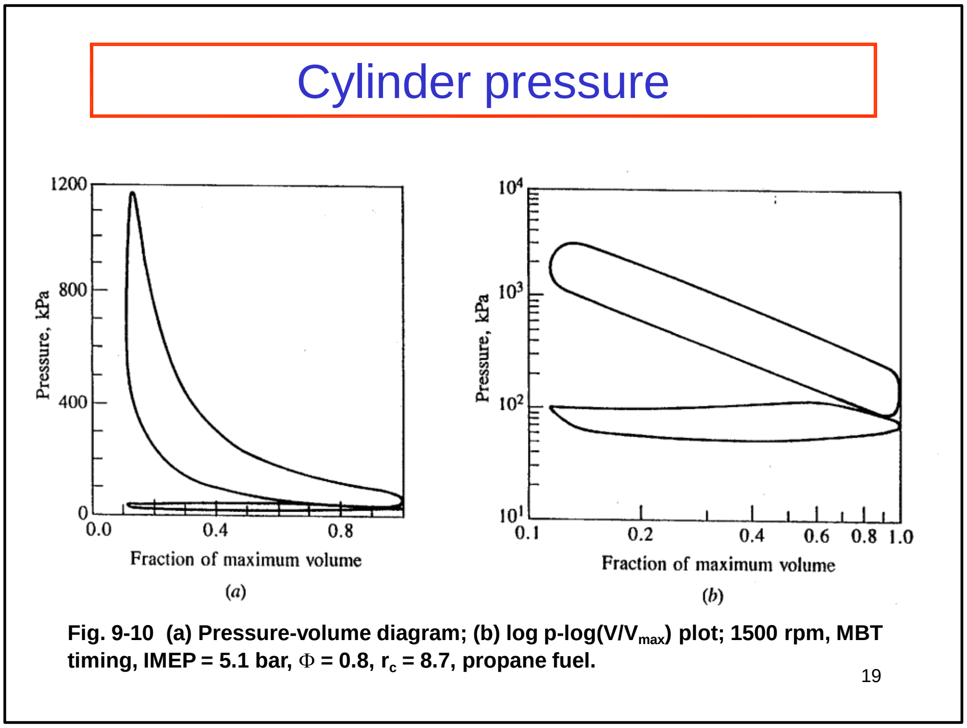
Technical Data

Range	bar	0 ... 250
Calibrated partial range	bar	0 ... 50
Overload	bar	300
Sensitivity	pC/bar	≈−16
Natural frequency	kHz	≈100
Linearity, all ranges	%FSO	≤±0,5
Acceleration sensitivity		
axial	bar/*g*	<0,0015
radial	bar/*g*	<0,0003
Operating temperature range	°C	-50 ... 350
Sensitivity shift		
20 ... 100°C	%	≈±1
20 ... 350°C	%	≤±3,5
200± 50°C	%	≈1
Insulation resistance		
bei 20°C	Ω	≥10^{13}
Shock insulation	*g*	2000
Tightening sensitivity	Nm	10
Capacitance	pF	8
Weight	*g*	10
Plug, ceramic insulator	Type	10-32 UNF

Sensitivity of NIMEP to crank angle phase error

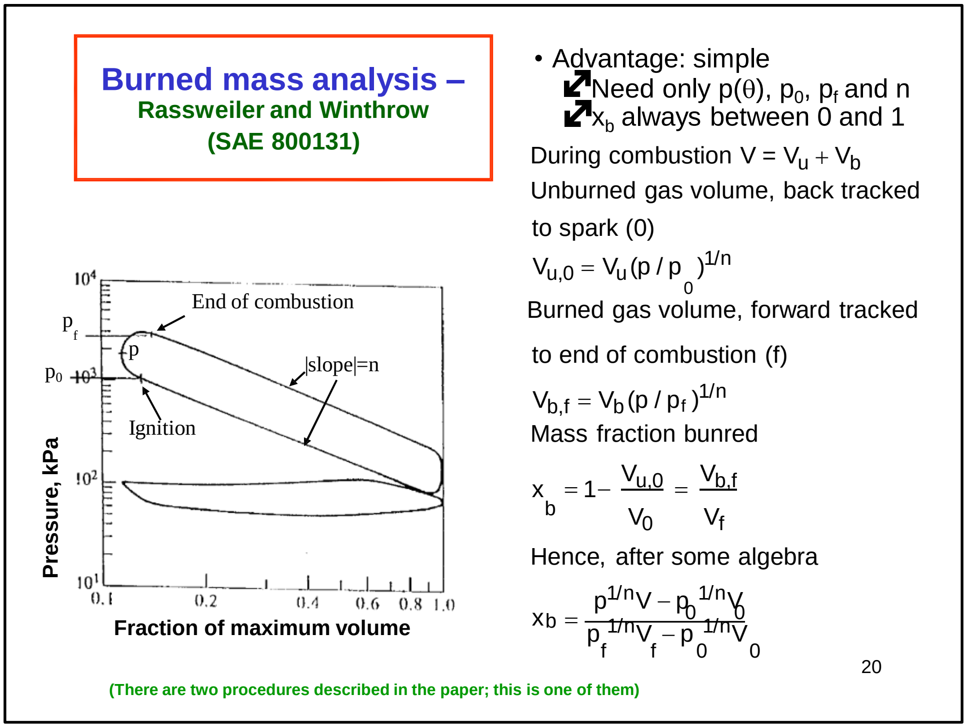
SI engine;1500 rpm, 0.38 bar intake pressure





Cylinder pressure

Fig. 9-10 (a) Pressure-volume diagram; (b) log p-log(V/V_{max}) plot; 1500 rpm, MBT timing, IMEP = 5.1 bar, $\Phi = 0.8$, $r_c = 8.7$, propane fuel.



Burned mass analysis –

Rassweiler and Winthrow

(SAE 800131)

- Advantage: simple
  - Need only $p(\theta)$, p_0, p_f and n
  - x_b always between 0 and 1

During combustion $V = V_u + V_b$

Unburned gas volume, back tracked to spark (0)

$$V_{u,0} = V_u (p / p_0)^{1/n}$$

Burned gas volume, forward tracked to end of combustion (f)

$$V_{b,f} = V_b (p / p_f)^{1/n}$$

Mass fraction bunred

$$x_b = 1 - \frac{V_{u,0}}{V_0} = \frac{V_{b,f}}{V_f}$$

Hence, after some algebra

$$x_b = \frac{p^{1/n}V - p_0^{1/n}V_0}{p_f^{1/n}V_f - p_0^{1/n}V_0}$$

(There are two procedures described in the paper; this is one of them)

Heat release analysis

1 zone model

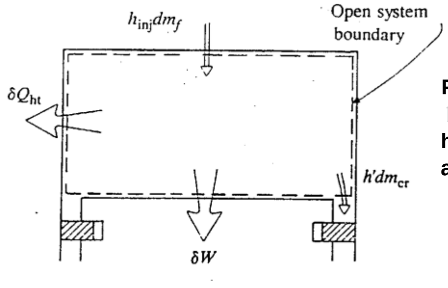


Fig. 9-11 Open system boundary for heat- release analysis

Energy balance:

Fuel chemical Energy release dQ_{ch}/dt = dU_s/dt Sensible energy change

+ pdV/dt Work transfer

+ dQ_{ht}/dt Heat loss to walls

+ $h' \, dm_{cr}/dt$ Flow into crevice

- $h_{inj} \, dm_f/dt$ Injected enthalpy

} Net heat released

Results of heat-release analysis

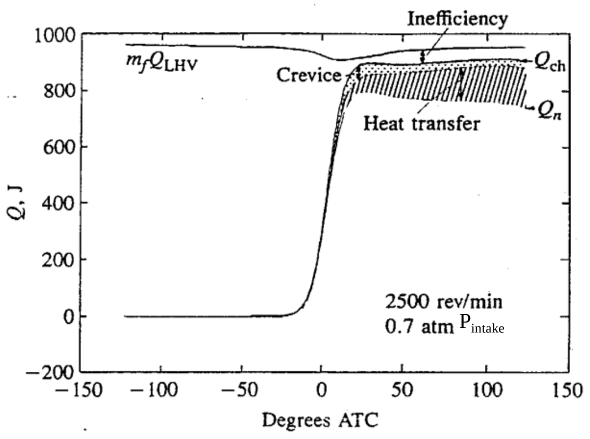


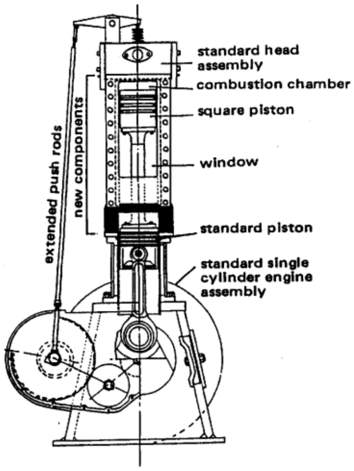
Fig. 9-12 Results of heat-release analysis showing the combustion inefficiency and the corrections due to heat transfer and crevice effect.

Flow and Combustion Process
in
Spark-Ignition Engine

A Color Schieren Movie
taken in a Special Visualization Engine

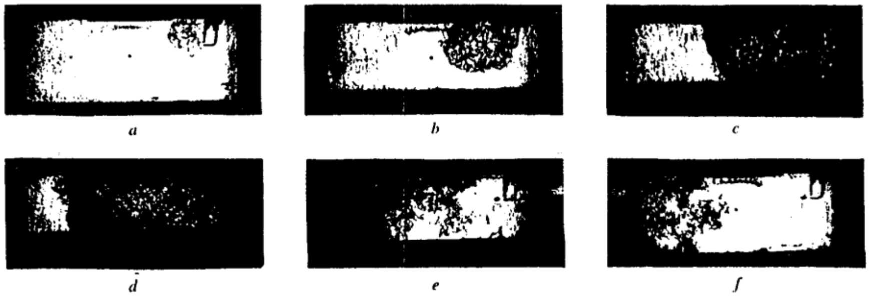
- Square piston engine
- Visualization by color-schlieren method
  - Captures density gradients
- Note:
  - Flame propagation process
  - Outgasing from crevices

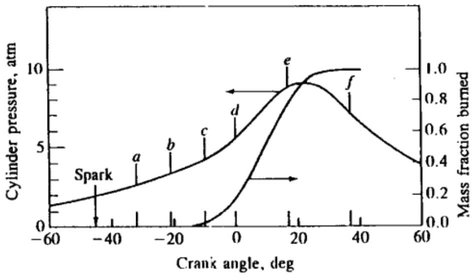
Square piston flow visualization engine



Bore	**82.6 mm**
Stroke	**114.3**
mm Compression ratio	**5.8**
Operating condition	
Speed	**1400 rpm**
Φ	**0.9**
Fuel	
propane Intake pressure	
0.5 bar Spark timing	
MBT	

Flame Propagation (Fig 9-14)





1400 rpm
0.5 bar inlet pressure

Knocking (Detonation)

Spark discharge characteristics

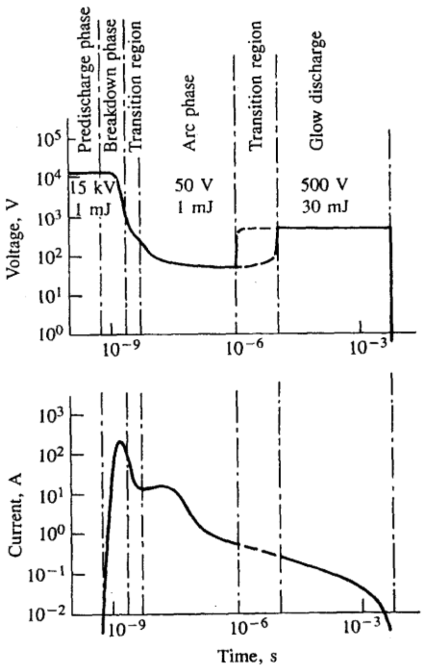
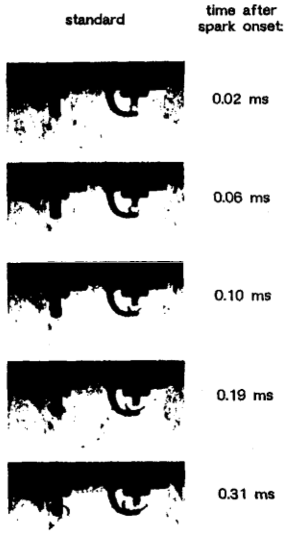


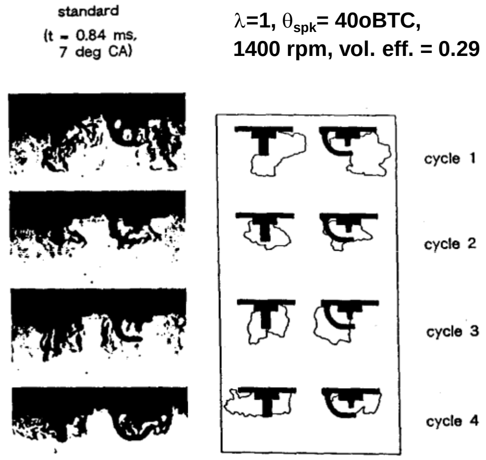
Fig.9-39
Schematic of voltage and current variation with time for conventional coil spark-ignition system.

Flame Kernel Development (SAE Paper 880518)

λ=1, θ_{spk}= 40oBTC,
1400 rpm, vol. eff. = 0.29

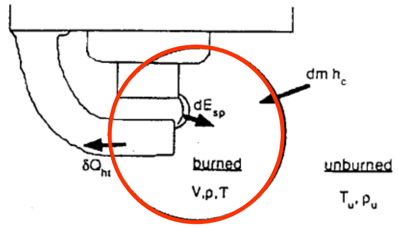


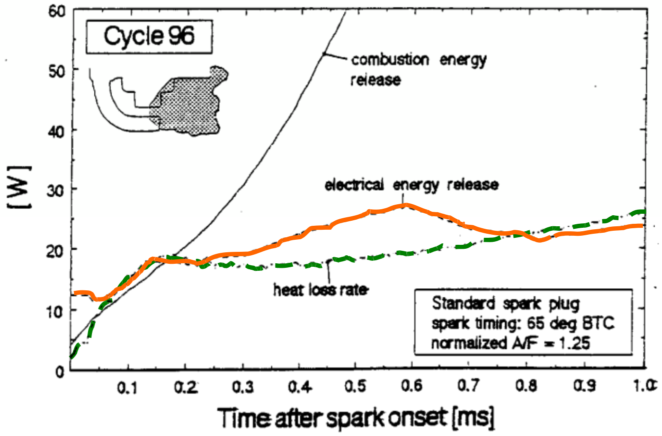
Single cycle flame sequence



Flame from 4 consecutive cycles at fixed time after spark

Energy associated with Spark Discharge, Combustion and Heat Loss





SAE Paper 880518

Ignition and Flame Development Process

1. Spark discharge creates a high temperature plasma kernel which expands rapidly (1mm, 100 μs).
2. The hot reactive gas at the outer edge of this kernel causes the adjacent fuel-air mixture to ignite, creating an outward propagating flame which is almost spherical.
3. As the flame grows larger, the flame surface is distorted by the turbulence of the fluid motion. A wrinkled laminar flame results.
4. Because of the significant surface area enhancement by the wrinkling, the locally laminar “turbulent” flame burns rapidly.

Schematic of entrainment-and-burn model

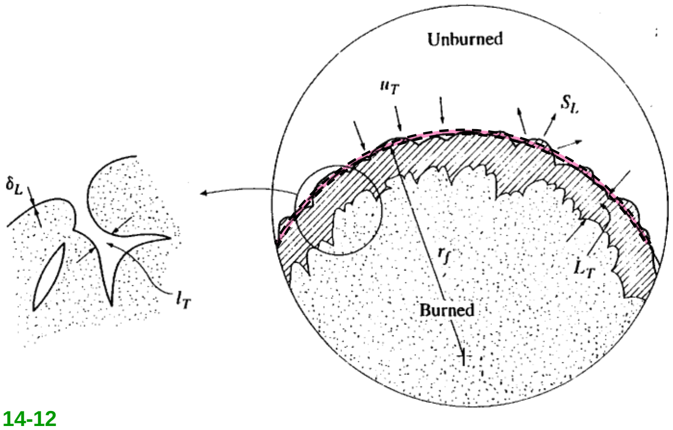


Fig. 14-12

SI engine flame propagation
Entrainment-and-burn model

Rate of entrainment:

$$\frac{GP_H}{GW} = \rho_X \$_I 6_/ + \rho_X \$_I X_7 \left(- H^{-W} \tau_E \right)$$

Laminar diffusion through flame front

Turbulent entrainment

Rate at which mixture burns:

$$\frac{GP_E}{GW} = \rho_X \$_I 6_/ + \frac{P_H - P_E}{\tau_E} \qquad \tau_E = \frac{\text{Ⓛ}_7}{6_/}$$

Laminar frontal burning

Conversion of entrained mass into burned mass

Critical parameters: u_T and Ⓛ_T

SI Engine design and operating factors affecting burn rate

)ODPH JHRPHWU\

The frontal surface area of the flame directly affects the burn rate. This flame area depends on flame size, combustion chamber shape, spark plug location and piston position.

,Q F\OLQGHU WXUEXOHQFH GXULQJ FRPEXVWLRQ

The turbulence intensity and length scale control the wrinkling and stretching of the flame front, and affect the effective burning area. These parameters are determined largely by the intake generated flow field and the way that flow changes during compression.

0L[WXUH FRPSRVLWLRQ DQG VWDWH

The local consumption of the fuel-air mixture at the flame front depends on the laminar flame speed S_L. The value of S_L depends on the fuel equivalence ratio, fraction of burned gases in the mixture (residual plus EGR), and the mixture temperature and pressure.

Cycle-to-cycle variations

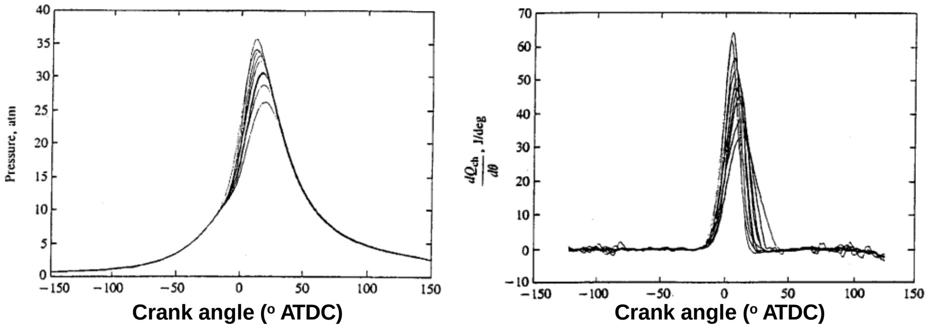
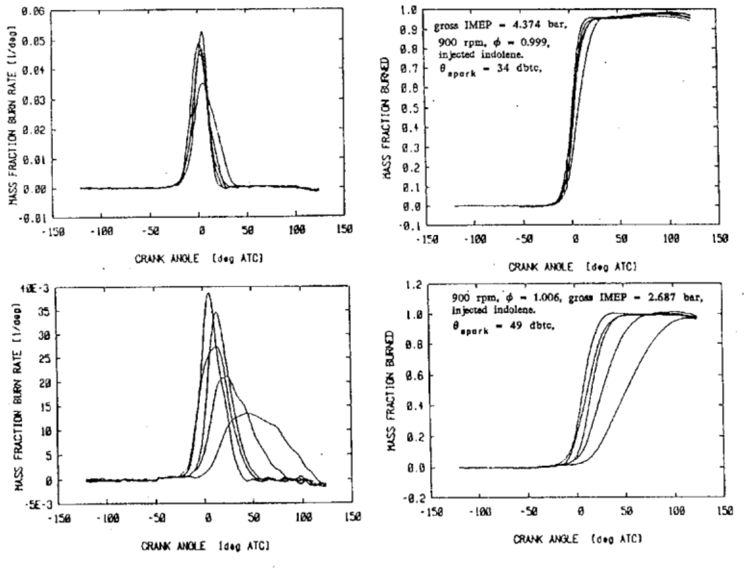


Fig. 9-31

Measured cylinder pressure and calculated gross heat-release rate for ten cycles in a single-cylinder SI engine operating at 1500 rpm, Φ = 1.0, MAP = 0.7 bar, MBT timing 25°BTC

Cycle-to-cycle change in combustion phasing



SI ENGINE CYCLE-TO-CYCLE VARIATIONS

Phases of combustion

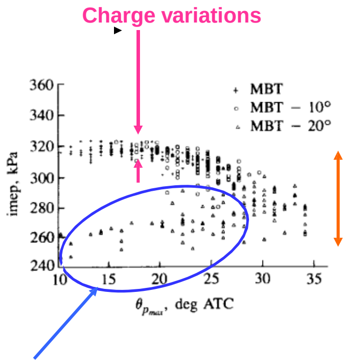
1. Early flame development
2. Flame propagation
3. Late stage of burning

Factors affecting SI engine cycle-to-cycle variations:

- (a) Spark energy deposition in gas (1)
- (b) Flame kernel motion (1)
- (c) Heat losses from kernel to spark plug (1)
- (d) Local turbulence characteristics near plug (1)
- (e) Local mixture composition near plug (1)
- (f) Overall charge components - air, fuel, residual (2, 3)
- (g) Average turbulence in the combustion chamber (2, 3)
- (h) Large scale features of the in-cylinder flow (3)
- (i) Flame geometry interaction with the combustion chamber (3)

Cycle distributions

Fig. 9-33 (b)

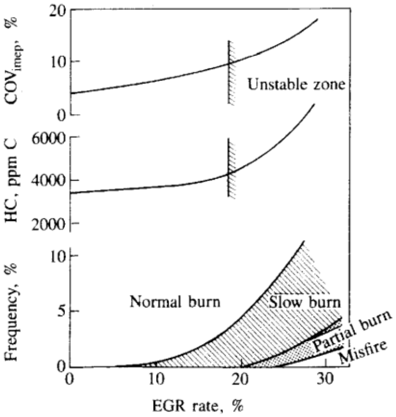


Charge variations

Charge and combustion duration variations

Very Slow-burn cycles

Fig. 9-36 (b)



Partial burn – substantial combustion inefficiency (10-70%)

Misfire – significant combustion inefficiency (>70%)

(No definitive value for threshold)

Knock

Processes

- Auto-ignition
- Rapid heat release
- Pressure oscillation

Consequences

- Audible noise
- Damage to combustion chamber in severe knock

How to “burn” things?

Reactants → Products

Premixed

- Premixed flame
  - Examples: gas grill, SI engine combustion
- Homogeneous reaction
  - Fast/slow reactions compared with other time scale of interest
  - Not limited by transport process
- Detonation
  - Pressure wave driven reaction

Knock → Homogeneous reaction, Detonation

Non-premixed

- Diffusion flame
  - Examples: candle, diesel engine combustion

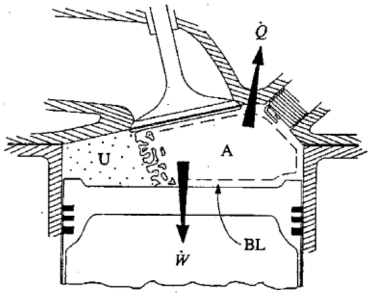
SI engine Combustion

Normal combustion

- Spark initiated premixed flame

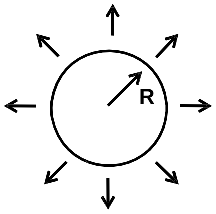
Abnormal combustion

- Pre-ignition (“diesel”)
  - – Ignition by hot surfaces or other means
- End gas knock (“spark knock”)
  - – Compression ignition of the not-yet-burned mixture (end gas)
  - – Affected by spark timing



Heat release rate and pressure wave

- When acoustic expansion is not fast enough to alleviate local pressure buildup due to heat release, pressure wave develops



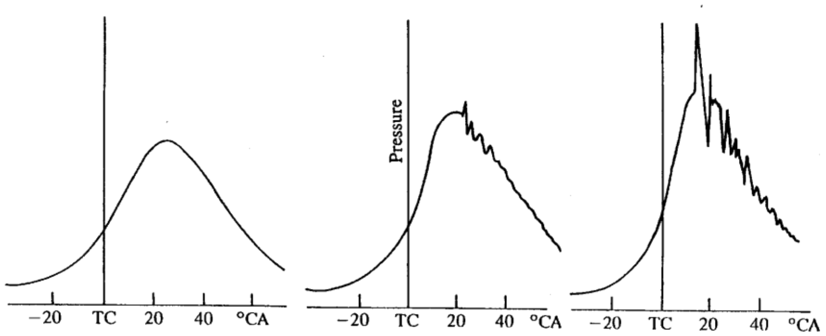
$\dot{q}$ = Heat release per unit volume over sphere of radius R

a = Sound speed

Critirion for setting up pressure wave:

$$\dot{q} \geq \frac{3\gamma}{\gamma - 1}\frac{ap}{R}$$

Pressure oscillations observed in engine knock



(*a*) Normal combustion, spark 28°BTC

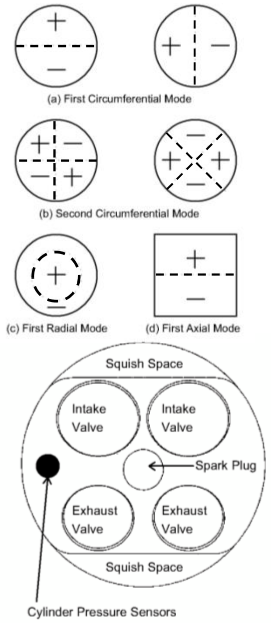
(*b*) Slight knock, spark 28°BTC

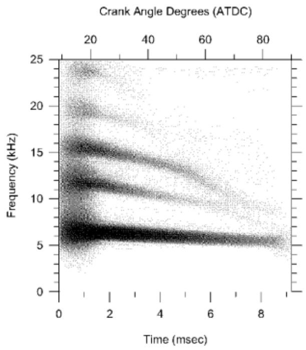
(*c*) Intense knock, spark 32°BTC

Single cylinder engine, 381 cc displacement; 4000 rpm, WOT

Fig. 9-59

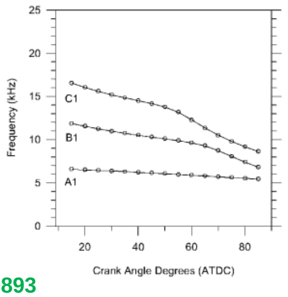
Acoustic modes





Spectrogram of 4 valve engine knock pressure data

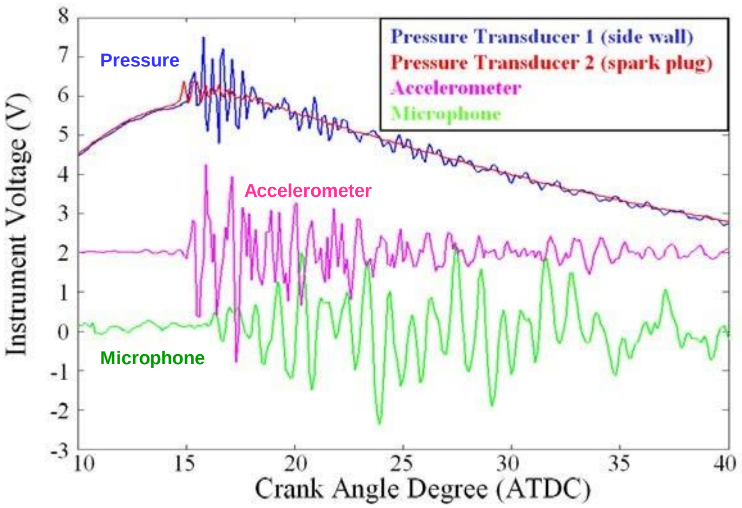
(2L I-4 engine; CR=9.6)



Calculated acoustic frequency of modes by FEM

SAE Paper 980893

Steps to Audible Knock



Auto-ignition
↓
Pressure oscillation
↓
Block vibration
↓
Audible noise

Heavy Knock/ detonation

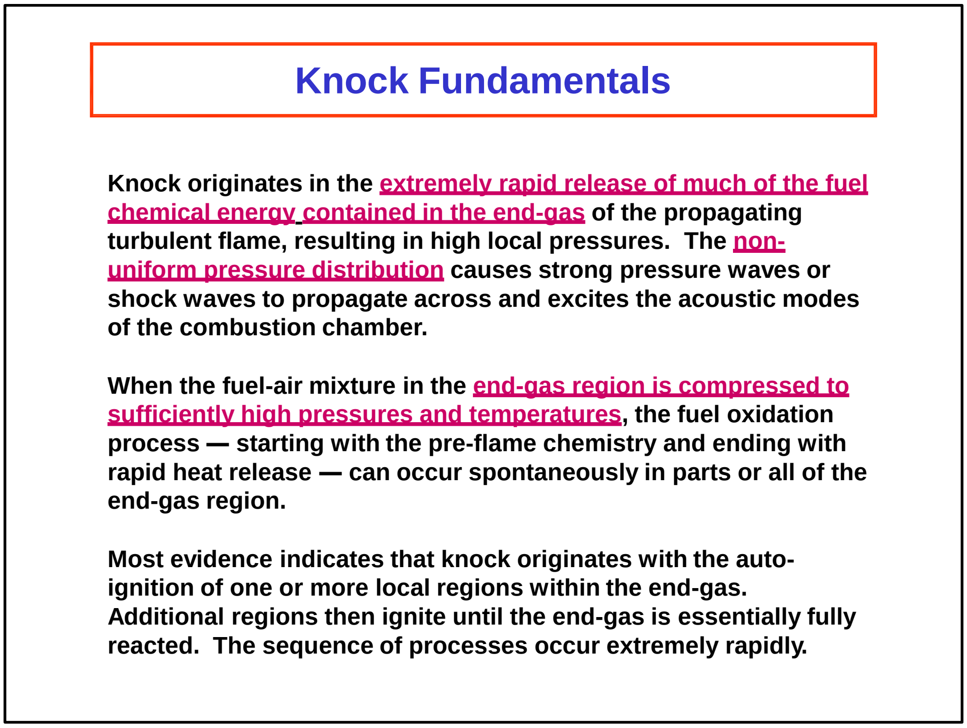
- Rapid combustion of stoichiometric mixture at compressed condition
  - Approximately constant volume
  - Local P ~ 100 to 150 bar
  - Local T > 2800^{o}K
- High pressure and high temperature lead to structural damage of combustion chamber

Knock damaged pistons

From Lichty, *Internal Combustion Engines*

From Lawrence Livermore website

Image courtesy of Lawrence Livermore National Laboratory.



Knock Fundamentals

Knock originates in the extremely rapid release of much of the fuel chemical energy contained in the end-gas of the propagating turbulent flame, resulting in high local pressures. The non-uniform pressure distribution causes strong pressure waves or shock waves to propagate across and excites the acoustic modes of the combustion chamber.

When the fuel-air mixture in the end-gas region is compressed to sufficiently high pressures and temperatures, the fuel oxidation process — starting with the pre-flame chemistry and ending with rapid heat release — can occur spontaneously in parts or all of the end-gas region.

Most evidence indicates that knock originates with the auto-ignition of one or more local regions within the end-gas. Additional regions then ignite until the end-gas is essentially fully reacted. The sequence of processes occur extremely rapidly.

Knock chemical mechanism

CHAIN BRANCHING EXPLOSION

Chemical reactions lead to increasing number of radicals, which leads to rapidly increasing reaction rates

Chain Initiation

$$RH + O_2 \Rightarrow \dot{R} + H\dot{O}_2$$

Chain Propagation

$$\dot{R} + O_2 \Rightarrow R\dot{O}_2, \text{etc.}$$

Formation of Branching Agents

$$R\dot{O}_2 + RH \Rightarrow ROOH + \dot{R}$$

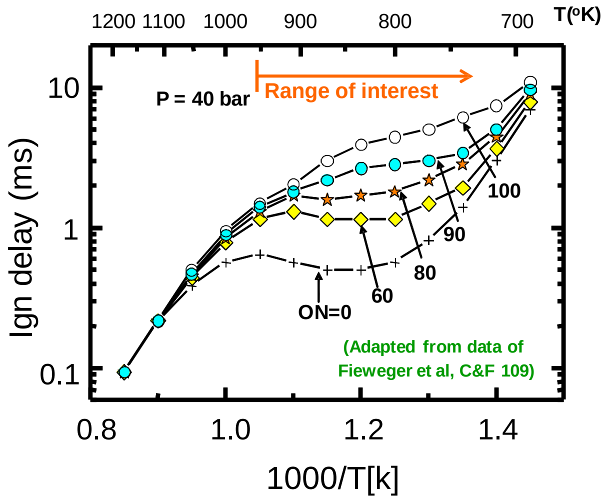
$$R\dot{O}_2 \Rightarrow R'CHO + R'\dot{O}$$

Degenerate Branching

$$ROOH \Rightarrow R\dot{O} + \dot{O}H$$

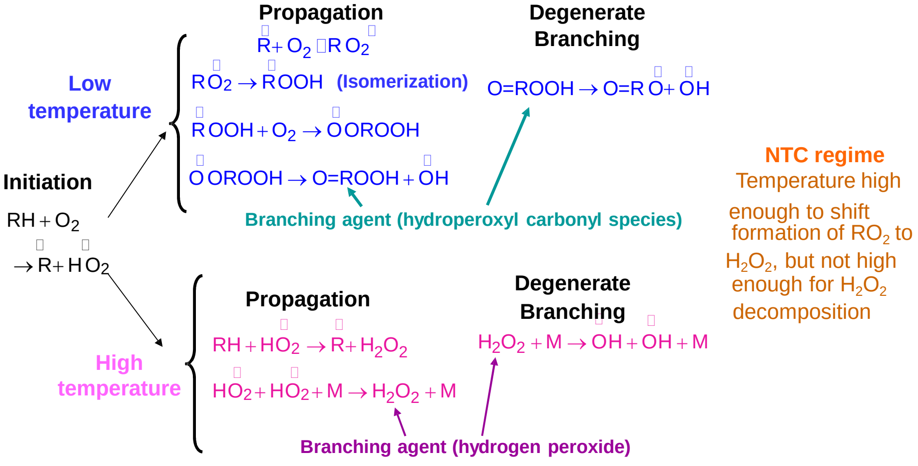
$$R'CHO + O_2 \Rightarrow R'\dot{C}O + H\dot{O}_2$$

Ignition delay for primary reference fuels



(Adapted from data of Fieweger et al, C&F 109)

Ignition delay kinetics



Low temperature

Propagation

$\dot{R}+O_2 \rightleftharpoons R\dot{O}_2$

$\dot{R}O_2 \to \dot{R}OOH$ **(Isomerization)**

$\dot{R}OOH + O_2 \to \dot{O}OROOH$

$\dot{O}OROOH \to O{=}ROOH + \dot{O}H$

Degenerate Branching

$O{=}ROOH \to O{=}R\dot{O} + \dot{O}H$

Branching agent (hydroperoxyl carbonyl species)

Initiation

$RH + O_2 \to \dot{R} + H\dot{O}_2$

High temperature

Propagation

$RH + H\dot{O}_2 \to \dot{R} + H_2O_2$

$H\dot{O}_2 + H\dot{O}_2 + M \to H_2O_2 + M$

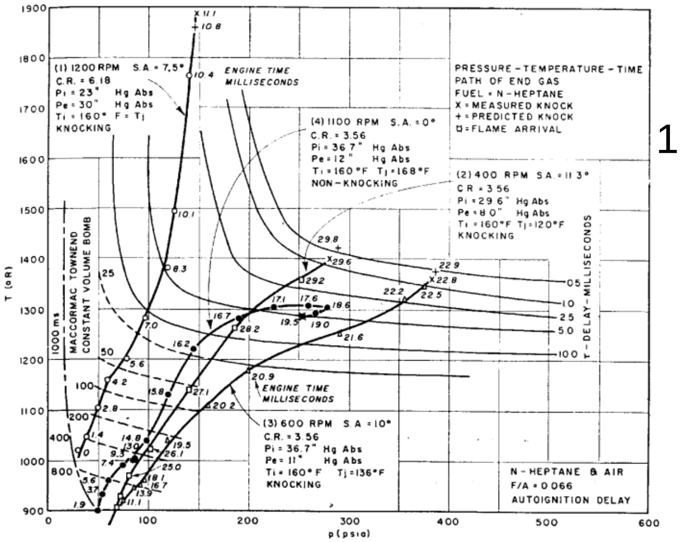
Degenerate Branching

$H_2O_2 + M \to \dot{O}H + \dot{O}H + M$

Branching agent (hydrogen peroxide)

NTC regime
Temperature high enough to shift formation of RO_2 to H_2O_2, but not high enough for H_2O_2 decomposition

Livengood and Wu integral



$$1=\int^{t_{ign}} \frac{dt}{\tau(p(t),T(t))}$$

5th Combustion Symposium, 1954

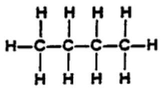
FUEL FACTORS

- The auto-ignition process depends on the fuel chemistry.
- Practical fuels are blends of a large number of individual hydrocarbon compounds, each of which has its own chemical behavior.
- A practical measure of a fuel's resistance to knock is the octane number. High octane number fuels are more resistant to knock.

Types of hydrocarbons

(See text section 3.3)

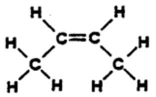
PARAFFINS



Butane

The carbon atoms in paraffins are held together chemically by single bonds. Paraffins have the general formula C_nH_{2n-2} with "n" indicating the number of carbon atoms.

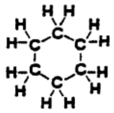
OLEFINS

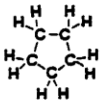


cis-2-Butene

Olefins are similar to paraffins, but they have two fewer hydrogen atoms and contain one double bond between two of the carbon atoms. Olefins have the general formula C_nH_{2n}. They rarely occur naturally in crude oil, but are formed in the refining process. Olefins may also be cyclic, resembling a naphthene with a double bond.

NAPHTHENES

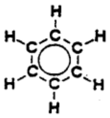




Cyclohexane Cyclopentane

Naphthenes are also called "cycloparaffins," because the carbon atoms are arranged in a ring structure — usually of five or six carbon atoms. If all the carbon atoms are held together by single bonds, naphthenes have the same general formula as olefins, C_nH_{2n}.

AROMATICS

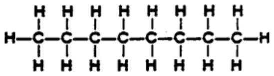


Benzene

Aromatics are odorous, ring-type hydrocarbons. The carbon atoms are joined by "aromatic" bonds, which are actually hybrids of single and double bonds.

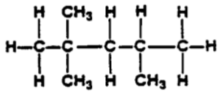
ISOMERS

There is one more thing you should know about paraffins and olefins. Paraffins with four or more carbon atoms can exist in more than one form. Butane, with four carbon atoms, is the simplest member of the paraffins in which it is possible to form two or more distinctly different chemical structures using the same number of hydrogen and carbon atoms. These variations are called isomers. For example, normal octane is a straight-chain hydrocarbon. It has 8 carbon and 18 hydrogen atoms and it looks like this:



n-Octane

Isooctane is one of the isomers of octane. It also has eight carbon and eighteen hydrogen atoms, but they form a branched chain, as in this example:



2,2,4-Trimethylpentane

Different isomers do not have the same properties. Isooctane is less likely to knock than normal octane — it has an RON of 100, compared to only 25 for its straight-chain cousin. Not surprisingly, it has become a standard for rating the performance of a gasoline.

Knock tendency of individual hydrocarbons

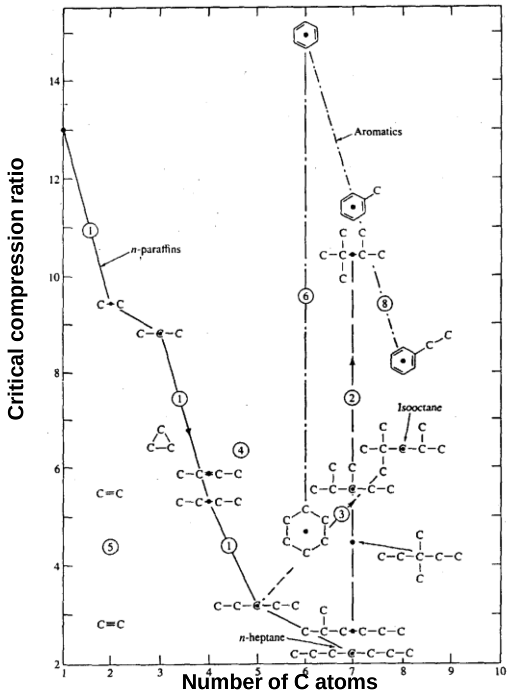


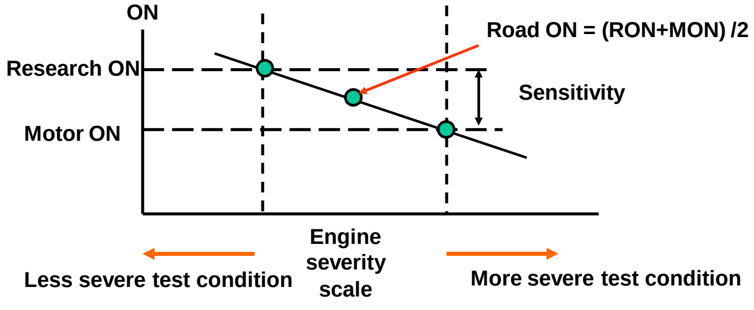
Fig 9-69

Critical compression ratio for incipient knock at 600 rpm and 450 K coolant temperature for hydrocarbons

Fuel anti-knock rating

(See table 9.6 for details)

- Blend primary reference fuels (iso-octane and normal heptane) so its knock characteristics matches those of the actual fuel.
- Octane no. = % by vol. of iso-octane
- Two different test conditions:
  - Research method: 52^{o}C (125^{o}F) inlet temperature, 600 rpm
  - Motor method: 149^{o}C (300^{o}F) inlet temperature, 900 rpm



ON

Research ON

Motor ON

Road ON = (RON+MON) /2

Sensitivity

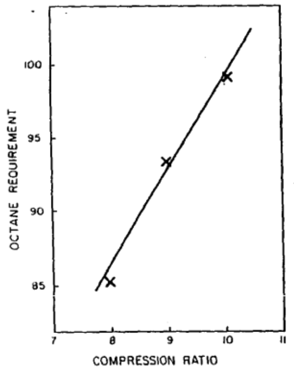
Engine severity scale

Less severe test condition

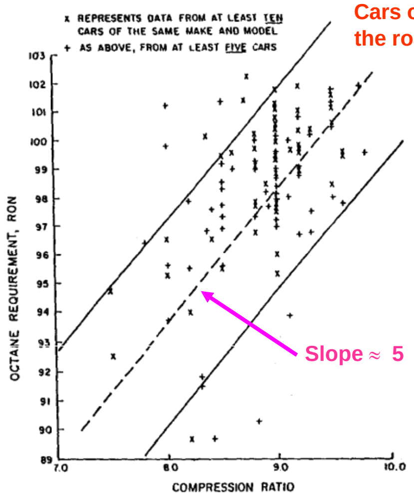
More severe test condition

Octane requirement

Engine on test stand



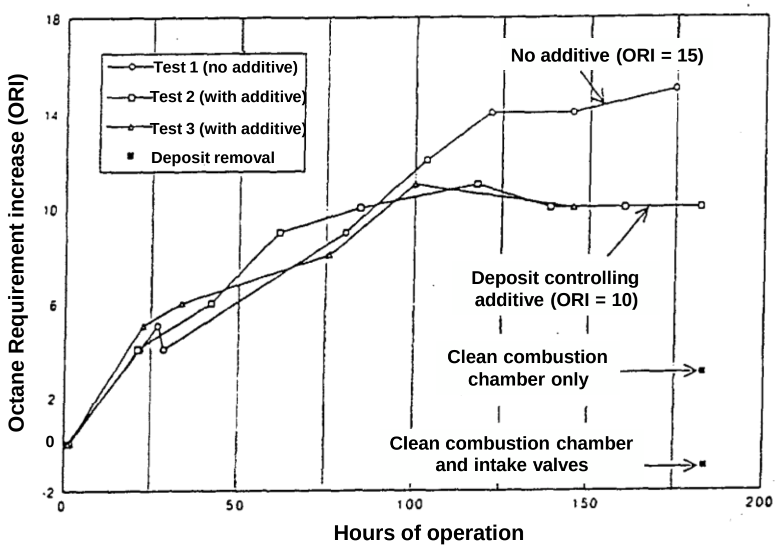
Cars on the road



Slope ≈ 5

From Balckmore and Thomas, *Fuel Economy of the Gasoline Engine*, Wiley 1977.

Octane Requirement Increase



No additive (ORI = 15)

Deposit controlling additive (ORI = 10)

Clean combustion chamber only

Clean combustion chamber and intake valves

Hours of operation

ACS Vol. 36, #1, 1991

ONR with change of engine parameters

Parameters	Octane Number Requirement (ONR)	Range Tested
Spark Advance	Increase 1 ONR / 1° knock limited spark advance	0 - 30° CA
Intake Air Temperature	Increase 1 ONR / 7°C	20 - 90°C
Air-Fuel Ratio (AFR)	Peaks around 5% rich of stoichiometric, Decreases 2 ONR / 0.1 λ	0.8 - 1.6 λ
Dilution: Cooled EGR	Decrease 3-4 ONR / 10% mass diluent	0 - 20% mass diluent
Manifold Absolute Pressure	Increase 3-4 ONR / 10 kPa	85 - 135 kPa
Compression Ratio	Increase 5 ONR / CR	5 - 12 CR
Exhaust Back Pressure	Increase 1 ONR / 30 kPa	0 - 65 kPa
Coolant Temperature	Increase 1 ONR / 10°C	70 - 110°C
Altitude	Decrease 1.4 ONR/300 m Decrease 2.5 ONR/300 m	0 - 1800m 1800 - 3600m
Humidity	Decrease 1 ONR when increasing relative humidity from 40% to 50% at 30°C	-
Engine Deposits	Increase 6-9 ONR over life of engine	0 - 250000km
Excessive Oil Consumption	Increase up to 12 ONR depending on driving style	-
Type of Fuel Injection	Decrease 4 ONR when DI used over PFI	-
Increasing Squish	Decrease up to 5 ONR as squish area increases	0 - 67% squish area
Combustion Chamber Shape	Decrease up to 15 ONR from cylindrical to modern type chamber	7.8 - 11 CR
Hydrogen (H_2) Addition	Decrease 1 ONR / 1% H_2 added	0 - 12% H_2 added

From SAE Paper 2012-01-1143

Knock control strategies

1. Provide adequate cooling to the engine
2. Use intercooler on turbo-charged engines
3. Use high octane gasoline
4. Anti-knock gasoline additives
5. Fuel enrichment under severe condition
6. Use knock sensor to control spark retard so as to operate close to engine knock limit
7. Fast burn system
8. Gasoline direct injection

Anti-knock Agents

Alcohols

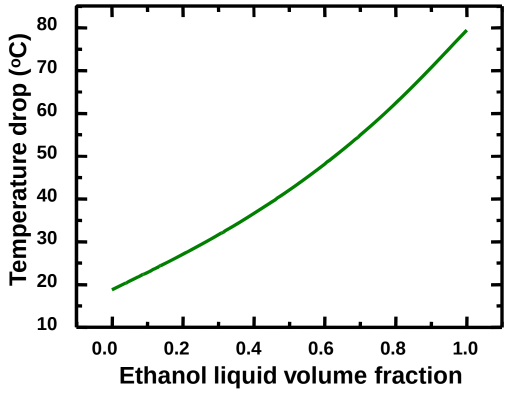
Methanol	CH_3OH
Ethanol	C_2H_5OH
TBA (Tertiary Butyl Alcohol)	$(CH_3)_3COH$

Ethers

MTBE (Methyl Tertiary Butyl Ether)	$(CH_3)_3COCH_3$
ETBE (Ethyl Tertiary Butyl Ether)	$(CH_3)_3COC_2H_5$
TAME (Tertiary Amyl Methyl Ether)	$(CH_3)_2(C_2H_5)COCH_3$

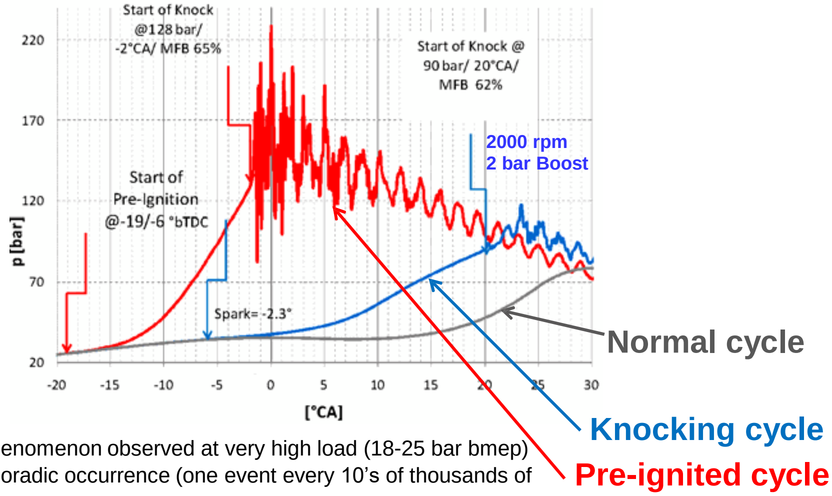
Adiabatic cooling of gasoline/ ethanol mixture

Preparing a stoichiometric mixture from air and liquid fuel



Note that Evaporation stops when temperature drops to dew point of the fuel in vapor phase

Sporadic Pre-ignition (super-knock)



- Phenomenon observed at very high load (18-25 bar bmep)
- Sporadic occurrence (one event every 10's of thousands of cycles)
- Each event may be one or more knocking cycles
- Mechanism not yet defined (oil, deposit, …?)

SI Engine Knock

1. Knock is most critical at WOT and at low speed because of its persistence and potential for damage. Part-throttle knock is a transient phenomenon and is a nuisance to the driver.

2. Whether or not knock occurs depends on engine/fuel/vehicle factors and ambient conditions (temperature, humidity). This makes it a complex phenomenon.

3. To avoid knock with gasoline, the engine compression ratio is limited to approximately 12.5 in PFI engines and 13.5 in DISI engines. Significant efficiency gains are possible if the compression ratio could be raised. (Approximately, increasing CR by 1 increases efficiency by one percentage point.)

4. Feedback control of spark timing using a knock sensor is used so that SI engine can operate close to its knock limit.

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