2.61 Internal Combustion Engines
Spring 2008

Lecture 9

SI Engine Combustion

- Movie of SI engine combustion
- The spark discharge
- The SI combustion flame propagation process
- Heat release phasing
- Heat release analysis from pressure data

Square piston flow visualization engine

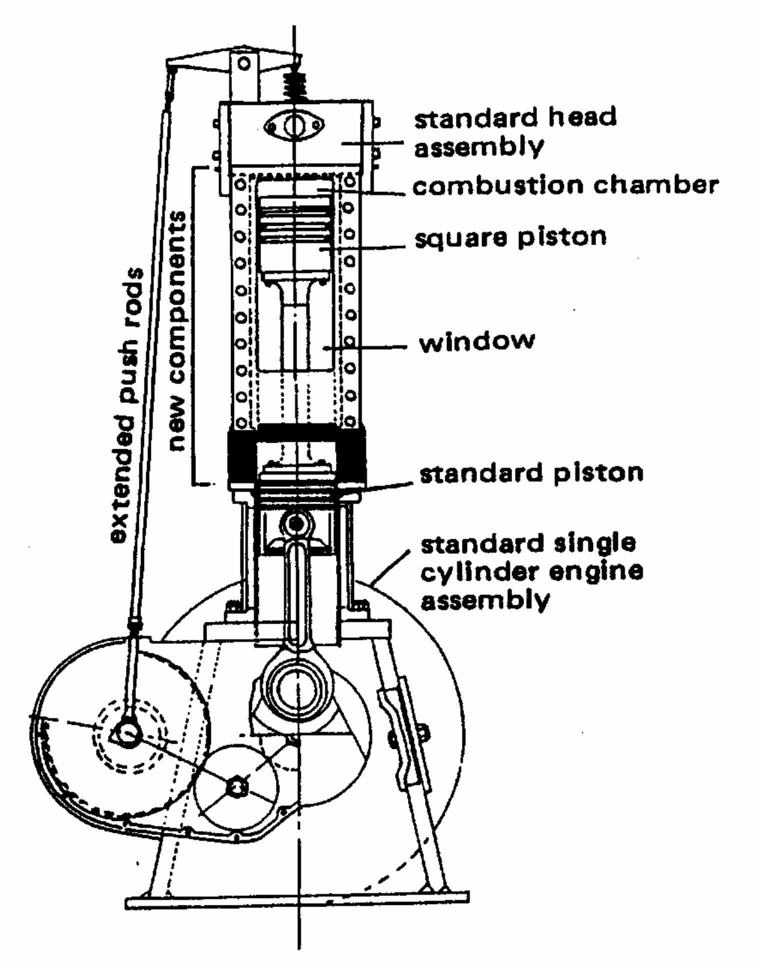

Fig. 3.1 in Constanzo, Vincent M. "A Visualization Study of Mixture Preparation Mechanisms for Port Fuel Injected Spark Ignition Engines." Master's Thesis, MIT. June 2004.

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

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$^{0.2}$

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

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Single cycle flame sequence

Flame from 4 consecutive cycles at fixed time after spark

Energy associated with Spark Discharge, Combustion and Heat Loss

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SAE Paper 880518

Schematic of SI engine flame propagation

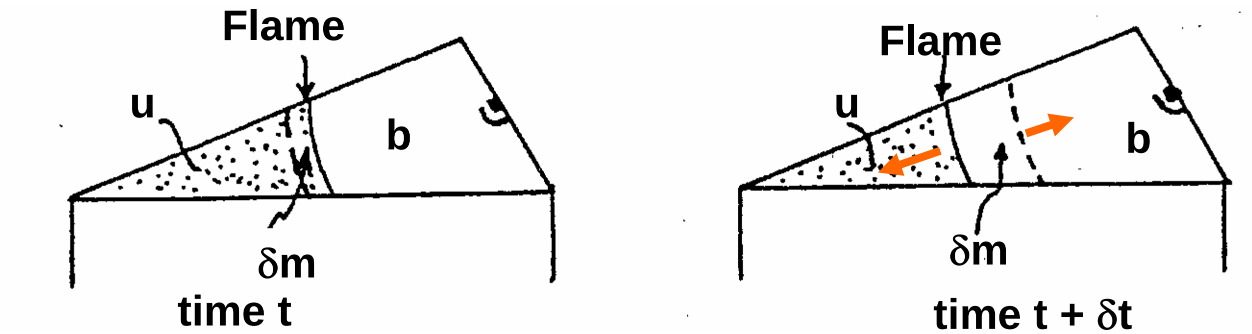
Heat transfer

Image removed due to copyright restrictions. Please see Fig. 9-4 in Heywood, John B. *Internal Combustion Engine Fundamentals*. New York, NY: McGraw-Hill, 1988.

Work transfer

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.

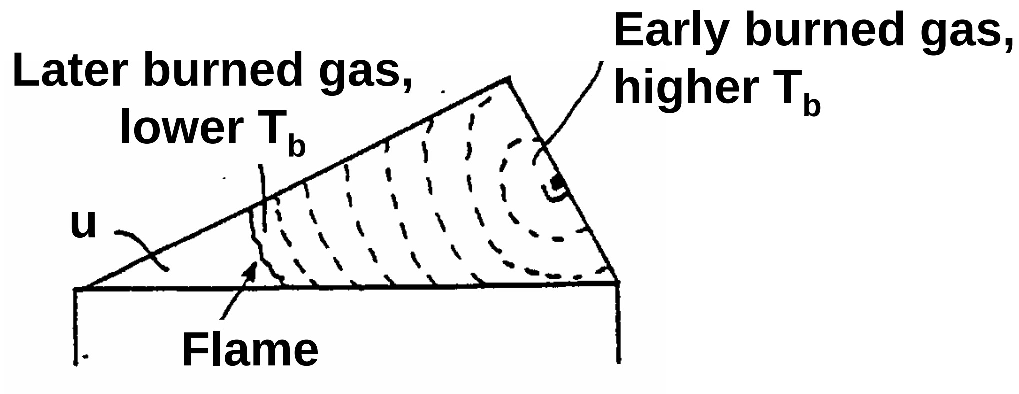
Combustion produced pressure rise



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: $\rho_b \sim ¼\, \rho_u$; $\delta V_b \sim 4\, \delta V_u$

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.

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^{o}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 (50º BTDC), MBT timing (30º BTDC), and retarded timing (10º BTDC). (b) Effect of spark advance on brake torque at constant speed and A/F, at WOT

Control of spark timing

WOT

Fig. 15-17

Fig. 15-3

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.

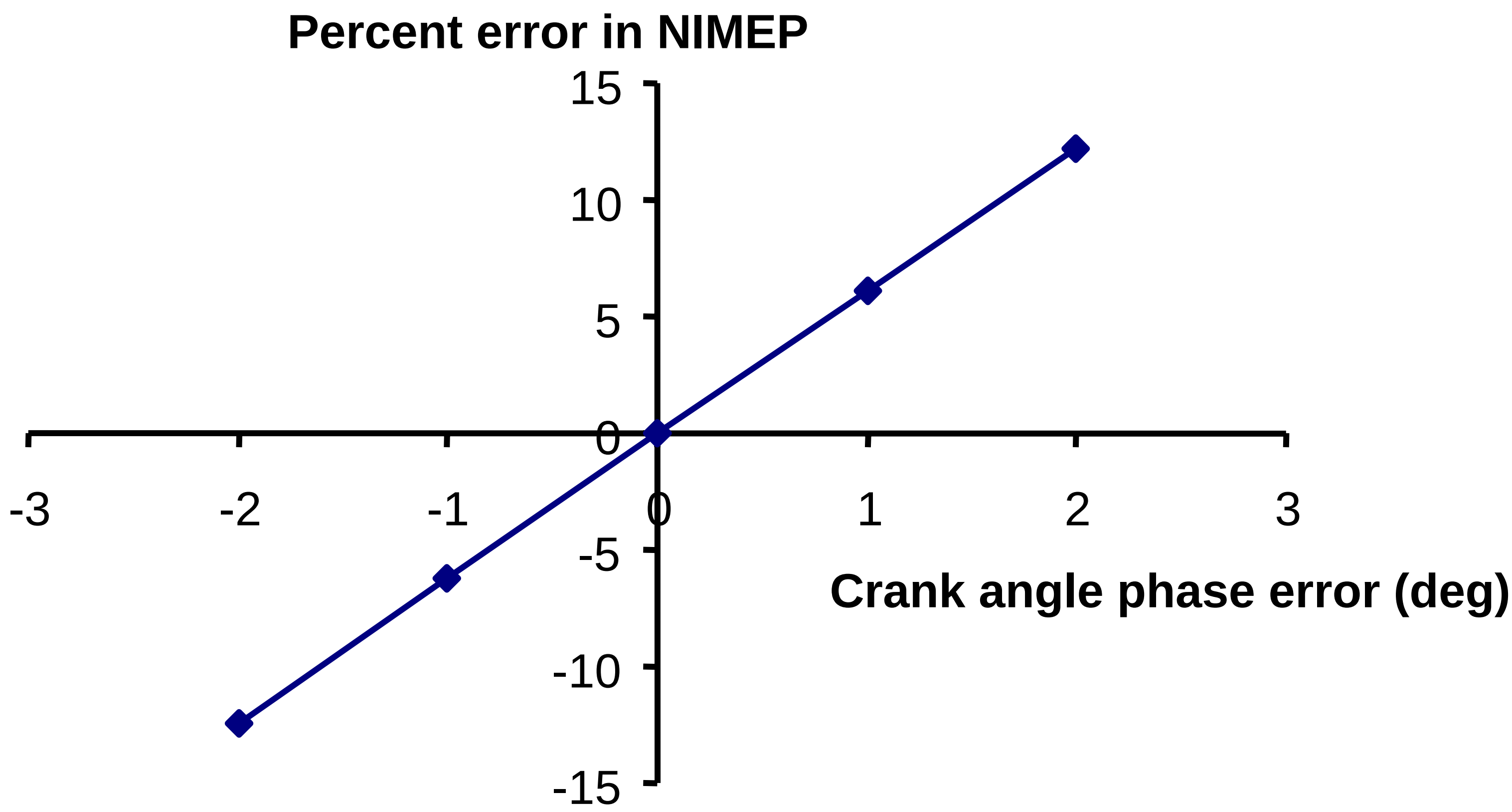
Image and data table removed due to copyright restrictions. Please see
http://www.intertechnology.com/Kistler/pdfs/Pressure_Model_6125B.pdf

6.2mm →   ←

Kistler 6125

Sensitivity of NIMEP to crank angle phase error

SI engine;1500 rpm, 0.38 bar intake pressure



Simple heat release analysis

- Single zone, perfect gas, idealized model

$$\frac{d}{d\theta}(mc_v T) = \dot{Q}_{gross} - \dot{Q}_{ht\ loss} - p\dot{V}$$

$$mc_v T = \frac{PV}{\gamma - 1}$$

whence $\dot{Q}_{gross} = \frac{\gamma}{\gamma-1} p\dot{V} + \frac{1}{\gamma-1}\dot{p}V + \dot{Q}_{ht\ loss}$

Mass fraction burned:

$$x_b = \frac{\dot{Q}_{gross}}{m_f LHV}$$

- Other effects:
  - Crevice effect
  - Blowby
  - Real gas properties
  - Non-uniformities (significant difference between burned and unburned gas)
  - Unknown residual fraction

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, Φ = 0.8, r_c = 8.7, propane fuel.

Burned mass analysis – Rassweiler and Winthrow (SAE 800131)

Image removed due to copyright restrictions. Please see Rassweiler, Gerald M., and Lloyd Withrow. "Motion Pictures of Engine Flames Correlated with Pressure Cards." *SAE Transactions* 33 (1938): 185-204. Reprinted as SAE Technical Paper 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

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

Results of heat-release analysis

P_{intake}

Fig. 9-12 Results of heat-release analysis showing the effects of heat transfer, crevices and combustion inefficiency.