LECTURE NOTES

on

INTERNAL COMBUSTION ENGINES
SI AN INTEGRATED EVALUATION

Integrated Master Course on
Mechanical Engineering

Tiago Farias
Mechanical Engineering Department

November 2015
Approach SI _ indirect injection & DI

• Review how it works
• What can we control: mass flow, spark ignition and lambda...
  • Air is power
  • Burn all the fuel versus burn all the air
• How to maximize mass air / volumetric efficiency?
• How to guarantee perfect combustion
• What lambda?
  • To maximize power
  • To minimize fuel consumption
• What spark advance?
  • To maximize efficiency and power
  • To avoid knocking
• How to address emission regulations:
  • Does it affect lambda?
2. Working Cycle

Four-stroke cycle

- Intake: Air-fuel mixture is drawn in.
- Compression: Air-fuel mixture is compressed.
- Power: Explosion forces piston down.
- Exhaust: Piston pushes out burned gases.

© 2007 Encyclopædia Britannica, Inc.
Comparação entre os ciclos Otto indicado e

\[ \psi = \frac{\eta_i}{\eta_{id}} \]
Spark Ignition engine (SI)

- Wedge shaped
- Squish area
- Four-valve Pentroof chamber

Piston-crown Combustion Chamber

Wedge shaped

Four-valve Pentroof chamber
Fuel injection (SI engines)

CARBURETOR

SINGLE-PORT INJECTION

MULTI-PORT INJECTION

DIRECT INJECTION (DI)

1 – Piston
2 – Exhaust channel
3 – Spark plug
4 – Exhaust valve
5 – Intake valve
6 – Indirect injector
7 – Intake channel
8 – Direct injector

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8. Fuel Injection

**CARBURETOR**

1. float regulates amount of gasoline entering chamber
2. nozzle discharges gasoline
3. air enters air horn
4. air and gasoline mix at narrow passage (venturi)
5. throttle valve determines speed of engine by controlling amount of air-fuel mixture release

**Port-Fuel INJECTION**

**Direct INJECTION**

---

### Differences between Gasoline Direct Injection and traditional Port Fuel Injection

<table>
<thead>
<tr>
<th></th>
<th>DI</th>
<th>Port Fuel Injection</th>
</tr>
</thead>
<tbody>
<tr>
<td>Where fuel is applied</td>
<td>Combustion chamber</td>
<td>Intake port</td>
</tr>
<tr>
<td>Fuel rail pressure</td>
<td>2,200 psi (150 bar)</td>
<td>Approximately 60 psi (4 bar)</td>
</tr>
<tr>
<td>Fuel apply (crank degrees)</td>
<td>Approximately 310 degrees</td>
<td>Up to 720 degrees</td>
</tr>
<tr>
<td>Ignition</td>
<td>Spark plug-based</td>
<td>Spark plug-based</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>Higher by approximately 10 percent</td>
<td>Limited by fuel application</td>
</tr>
<tr>
<td>Cam phasing</td>
<td>Mandatory</td>
<td>Recommended</td>
</tr>
<tr>
<td>Intake air/fuel temperature</td>
<td>Lower from vaporizing fuel</td>
<td>Limited by fuel application</td>
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\[ P_e \propto \frac{n}{n_r} c d^2 \ell \eta_v r_s \rho_o \frac{1}{\lambda} \frac{PCI}{1+x} \eta_id \psi \eta_m \]

Ideal Otto Cycle

Fig. 3-42 - Diagramas \( p-v \) e \( T-s \) de um ciclo Otto
What about the $\eta_{id}$? SI
Gasoline direct injection control system (Continental)
Approach SI _ indirect injection & DI

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Variable – length manifold take advantage of the dynamics of pressure waves in the intake and exhaust manifolds at high-speed engines to increase intake pressure without the use of a compressor.
effect of intake pipe length on the volumetric efficiency at different engine speeds for a naturally aspirated spark ignition racing engine
3-stage variable length intake manifold
Volumetric efficiency

\[ \eta_v = \frac{\dot{m}_a}{r_s \rho_o V_c} \quad r_s = \frac{\rho_{\text{comp}}}{\rho_o} \quad \eta_v = \frac{\dot{m}_a}{r_s \rho_o V \frac{n}{n_r}} \]
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SI engines combustion stages:
• Ignition and flame development
• Flame propagation
• Flame termination

Types of combustion
• Controlled combustion which is initiated by a spark
• Uncontrolled combustion which is initiated by hot spot
• Abnormal combustion which is known as “auto-ignition” causing engine knock

Ignition
1. Near the end of the compression stroke the cylinder contains a homogeneous FA mixture
2. The spark fires and ignites the mixture in the vicinity forming a thin flame front
3. Combustion spreads into the mixture
4. Rate of flame propagation depends on temperature and pressure of the flame front and the surrounding envelope
Flame is detected at about 6 degrees of crank rotation after spark plug firing (“delay period”)

At the end of the Delay Period 5-10 % of AF mass has been burned, the combustion process is then well established and the flame front accelerates fast

Indicator diagram shows:
- rate of pressure rise (detonation)
- ignition lag or delay period
- losses occurring in the induction and exhaust strokes

I – Ignition lag
II - Flame Propagation
III - Afterburning

Rate of pressure rise starts to decrease
The 2nd stage starts at the first measurable rise of pressure.

- Most of the fuel-air mixture is burned in this stage.
- Most of the useful work is produced in this stage.

Flame propagation:
- Turbulence-swirl-squish interaction enhances expansion of the flame front and the rate of pressure rise.
- Flame speed linear with engine speed.

- Volume increase of the burned gases compresses the unburned gases, and therefore.
- Heat transfer (radiation, conduction, convection) further increase the unburned gases temperature and pressure.

- 3/8 of AF mixture is completely burned at TDC.
- Temperature and pressure of the gases reaches maximum values just after TDC (5-10)°.
- Most of the fuel-air mixture is completely burned at about 15° after TDC.
**Hemispherical (or pent-roof)**

Air-fuel mixture enters on one side, and exhaust gases exit on the other, thus providing cross-flow.
- Helps the engine breathe because it provides room for relatively large valves and ports.
- Reduces the flame travel distance and provides rapid and effective combustion by allowing to locate the spark plug at the centre of the chamber.

**Bath-tub**

Valves are mounted vertically and side by side allowing a relatively simple valve-operating mechanism to be used; the plug is in one side, creating a short flame path from the spark-plug.
- Turbulence is assisted by the shape of chamber
- Strong squish is produced due to the fact that the cross section is smaller than the cylinder

**Wedge-shaped**

The plug is at the thick end of the wedge; the valves are in line and inclined from the vertical.
- Reduces damage caused by detonation because the flame is directed toward the small end of the wedge.
- Less fuel is left unburned after combustion, which reduces hydrocarbon exhaust emissions because, having a smaller surface area than the others, the area where fuel droplets can condense due to heat transfer is reduced.
Spark ignition advance (0° to 40°)

http://www.youtube.com/watch?v=_KEbXwvTxkg
Engine management maps

![Graph showing engine management maps with load and engine speed axes.](image-url)
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Pollutant emissions

\[ \lambda = 1 \Rightarrow AF_{st} \sim 14,6 \]

Fig. 3-1 Exhaust products from combustion of hydrocarbon fuels.

- **Main pollutants:** CO$_2$, HC, CO, NO$_x$ (> 90 % NO)
Internal Combustion Engines
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Emissions curves from Toyota Motor Sales literature
Effects of A/F Ratio on Exhaust O₂

Exhaust O₂ is lowest when A/F ratio is richer than "ideal"; however, O₂ increases dramatically with leaner mixtures.
The graph illustrates the relationship between the air/fuel ratio ($\lambda$) and various engine performance metrics:

- **Fuel Consumption**: This line shows a decrease as the air/fuel ratio increases.
- **CO (Carbon Monoxide)**: The line indicates an increase with a peak at around $\lambda = 1.0$.
- **NO\textsubscript{x} (Nitric Oxides)**: This line also shows an increase, peaking at a similar point as CO.
- **HC (Hydrocarbons)**: The line for HC decreases with an increase in $\lambda$.

These metrics are critical in understanding the efficiency and emissions of internal combustion engines.
1 - Throttle position sensor
2 - Air flow sensor
3 - Fuel pressure sensor
4 - Crankshaft speed sensor
5 - Knock sensor
6 - Temperature sensor
7 - Oxygen sensor 1
8 - Oxygen sensor 2
A - Fuel Injector
B - Ignition control
Pollutant emissions and regulations of vehicles

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Instituto Superior Técnico
Emissions: Spark ignition engines

Gasoline combustion

- \( C_8H_{15} + \alpha(\text{O}_2+3,76\text{N}_2) \rightarrow x_{st}\text{CO}_2 + y_{st}\text{H}_2\text{O} + z_{st}\text{N}_2 \)
  - \( x_{st} = 8; \alpha_{st} = 11,75 \)

- \( C_8H_{15} + \alpha(\text{O}_2+3,76\text{N}_2) \rightarrow x_{re}\text{CO}_2 + y_{re}\text{H}_2\text{O} + z_{re}\text{N}_2 \)
  + a \text{HC} + b \text{CO} + c \text{NO}_x (\text{NO};\text{NO}_2) \)

\[
\text{Air/fuel}_{st} = 11,75x(2x16+3,76x2x14) / (8x12+15x1) \\
= 14,53 \text{ kgair/kg}_{\text{fuel}}
\]

\[
\text{CO}_2 \text{ emissions} = (8x12+15x1)/(8x(14+2x16) \\
= 3,17 \text{ kg/kg}_{\text{fuel}}
\]
Summary of HC, CO, and NO pollutant formation mechanisms in a spark-ignition engine.
Schematic summarizing processes important in hydrocarbon emissions. (a) After the charge is burned, unburned fuel is in the vicinity of the walls at (1), (2), and (3) because of flame quenching and absorption by oil layers. In the crevice volume (4), the flame is quenched by the cool walls. (b) During the expansion process, hydrocarbons are desorbed by the oil layers and gas in the crevice volume expands and is laid along the cylinder walls. When the exhaust valve opens, hydrocarbons from regions (1) and (2) are entrained by the flow and exit the cylinder. (c) Roll-up of hydrocarbon rich cylinder wall boundary layer into a vortex as piston moves up cylinder during exhaust stroke.
Pollutant emissions

\( \lambda = 1 \Rightarrow AF_{st} \sim 14.6 \)

*Fig. 3-1 Exhaust products from combustion of hydrocarbon fuels.*

- **Main pollutants:** \( CO_2, HC, CO, NO_x (> 90 \% \ NO) \)
Euro emissions Control
Vehicle evolution in fuel consumption and emissions

(HC, CO, NOx, PM)

Europe: NEDC standard cycle
# Vehicle evolution in fuel consumption and emissions

<table>
<thead>
<tr>
<th>Tier</th>
<th>Date</th>
<th>CO</th>
<th>HC</th>
<th>HC+NOx</th>
<th>NOx</th>
<th>PM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Euro 1†</td>
<td>1992.07</td>
<td>2.72</td>
<td>3.16</td>
<td>0.97</td>
<td>1.13</td>
<td>0.14</td>
</tr>
<tr>
<td>Euro 2, IDI</td>
<td>1996.01</td>
<td>1.0</td>
<td>-</td>
<td>0.7</td>
<td>-</td>
<td>0.08</td>
</tr>
<tr>
<td>Euro 2, DI</td>
<td>1996.01</td>
<td>1.0</td>
<td>-</td>
<td>0.9</td>
<td>-</td>
<td>0.10</td>
</tr>
<tr>
<td>Euro 3</td>
<td>2000.01</td>
<td>0.64</td>
<td>-</td>
<td>0.56</td>
<td>0.50</td>
<td>0.05</td>
</tr>
<tr>
<td>Euro 4</td>
<td>2005.01</td>
<td>0.50</td>
<td>-</td>
<td>0.30</td>
<td>0.25</td>
<td>0.025</td>
</tr>
</tbody>
</table>
| Euro 5       | 2009.09b| 0.50 | -   | 0.23   | 0.18| 0.005⁵⁻
| Euro 6       | 2014.09| 0.50 | -   | 0.17   | 0.08| 0.005⁵⁻|
| Petrol (Gasoline) | |      |     |        |     |     |
| Euro 1†      | 1992.07| 2.72 | 3.16| 0.97   | 1.13| -   |
| Euro 2       | 1996.01| 2.2  | -   | 0.5    | -   | -   |
| Euro 3       | 2000.01| 2.30 | 0.20| -      | 0.15| -   |
| Euro 4       | 2005.01| 1.0  | 0.10| -      | 0.08| -   |
| Euro 5       | 2009.09b| 1.0  | 0.10³⁻| -      | 0.06| 0.005⁵,⁶⁻|
| Euro 6       | 2014.09| 1.0  | 0.10³⁻| -      | 0.06| 0.005⁵,⁶⁻|

* At the Euro 1..4 stages, passenger vehicles > 2,500 kg were type approved as Category N₁ vehicles.
† Values in brackets are conformity of production (COP) limits.
  a - until 1999.09.30 (after that date DI engines must meet the IDI limits).
  b - 2011.01 for all models.
  c - and NMHC = 0.068 g/km.
  d - applicable only to vehicles using DI engines.
  e - proposed to be changed to 0.003 g/km using the PMP measurement procedure.

www.dieselnet.com (g/km)
FTP 75

Cold start phase 0-505 s

Transient phase 505-1369 s

Hot start phase 0-505 s

Time, s

Speed, mile/h
Emissão de gases de escape no ciclo de testes EUA FTP 75

<table>
<thead>
<tr>
<th></th>
<th>Fase fria</th>
<th>Fase estabilizada</th>
<th>Motor desligado</th>
<th>Fase quente</th>
</tr>
</thead>
<tbody>
<tr>
<td>v (km/h)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>HC (%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CO (%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NOx (%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Tempo**

0  125  500  1000  1500  2000  2500 s

BOSCH
Emissions control

3 way catalyst converter

Fuel without lead
Catalyst efficiency

\[ \eta_{\text{cat}} \]

- Temperature Exhaust gases
- Age of the catalysts (kms)
Figura 5.9

1: Emissões antes do catalizador
2: Emissões depois do catalizador
3: Sinal da sonda lambda

Figura 5.8
Esquema de instalação de um catalizador de 3 vias de anel fechado
Emissions versus engine management
Pollutant emissions

\[
\lambda = 1 \implies AF_{st} \sim 14.6
\]

- **Main pollutants:** $\text{CO}_2$, $\text{HC}$, $\text{CO}$, $\text{NO}_x$ ($> 90 \% \text{ NO}$)

*Fig. 3-1 Exhaust products from combustion of hydrocarbon fuels.*
Emissions are also affected by SI advance

Diagram showing the relationship between angle of camshaft and emissions.
NOx

FIGURE 11-9
Variation of exhaust NO concentration with A/F and fuel/air equivalence ratio. Spark-ignition engine, 1600 rev/min, $\eta = 50$ percent, MBT timing.¹²
NOx

**FIGURE 11-13**
Variation of exhaust NO concentration with spark retard. 1600 rev/min, $\eta_s = 50$ percent; left-hand end of curve corresponds to MBT timing for each $A/F$.12
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SI engines

Engine maps

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Engine Characteristics Curves:

(WOT – Wide Open Throttle)
- Effective Power ($P_e$) vs rpm
- Effective Torque ($B_e$) vs rpm
- Specific Fuel Consumption vs rpm

Engine Maps:

Specific Fuel Consumption vs rpm and $P_e$ (or $B_e$)
Power vs rpm vs $P_e$ (or $B_e$)
Fig. 3-56 - Andamento típico das curvas de potência indicada, potência efectiva, binário efectivo, e potência de perdas mecânicas num motor de combustão interna. Identificação das velocidades de rotação mínima, de binário efectivo máximo, potência efectiva máxima, máxima admissível, e de embalamento.
- Cilindrada externa: 3.398 c.c.
- Diâmetro x curso: 52 x 65.5 mm.
- Cilindrada unitária: 307 c.c.
- Relação volumétrica de compressão: 2,8:1.
- Pressão máxima de sobressaturação: 1,1 bar.
- Ângulo inserção entre as válvulas: 33° 30 min.
- Diagrama de distribuição: avanço da abertura na admissão AAA.
- Potência máxima: 478 CV (351.5 kW) DIN às 7 000 r.p.m.
-binário máximo: 56,8 m/kg (677 Nm) DIN às 4 000 r.p.m.
- Regime máximo de utilização: 7 700 r.p.m.
- Potência específica: 183 CV/litro (120 kW/litro).
- Velocidade linear média dos eixos em regime de potência máxima: 19,2 m/seg às 7 000 r.p.m.
Fig. 3-60 - Zonas no diagramas \( p_e vs n \) onde \( C_e \) é necessariamente elevado.
In SI engines it is because of mixture enrichment from the action of the economizer and because of the poorer distribution at full throttle.

Moving to the left from point A to a lower piston speeds, the bsfc increases in SI engines because of the increased heat loss per cycle, poor distribution at low manifold velocities and lowered efficiency due to automatically retarded spark used for detonation control at low engine speeds.

In CI engines it is because of the increased fuel waste (smoke) associated with high fuel/air ratios at high loads.

Moving to a lower bmep from point A, the bsfc increases due to reduced mechanical efficiency.

At very low speeds (not shown in the plot) the CI engines may also have increased bsfc because the injection equipment cannot be set to give completely satisfactory characteristics over the entire speed range.
\[ P_e \propto \frac{n}{n_r} \left( \frac{c}{d^2} \ell \eta_v r_s \rho_o \frac{1}{\lambda} \frac{PCI}{1+x} \eta_i d \psi \eta_m \right) \]

\[ C_e = \frac{C_h}{P_e} = \frac{3600 \dot{m}_{fu}}{P_e} \]
Fig. 3-63 - Isolinhas de consumo específico (a)) e de consumo horário (b))

motor de automóvel, normalmente aspirado, 4 Tempos, 4 cilindros, 2000 cm³
(fonte: simulação a partir de dados fornecidos pela FIAT Richerche)
Fig. 3-62 - Diferenças nos diagramas em colina de $Ce$ para duas versões do mesmo motor de explosão, uma sem e outra com catalisador de três vias - motor de automóvel, normalmente aspirado, 4 Tempos, 4 cilindros, 1242 cm³ (fonte: Ford)
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