ENGINE OPERATION CHARACTERISTICS

Nazaruddin Sinaga

Vehicle Road Load Requirement

$$P_b = \frac{1}{\eta_T}(F_R + F_D + F_a + F_C)S_V$$

P_b = Required engine brake power output

η_T = Transmission efficiency

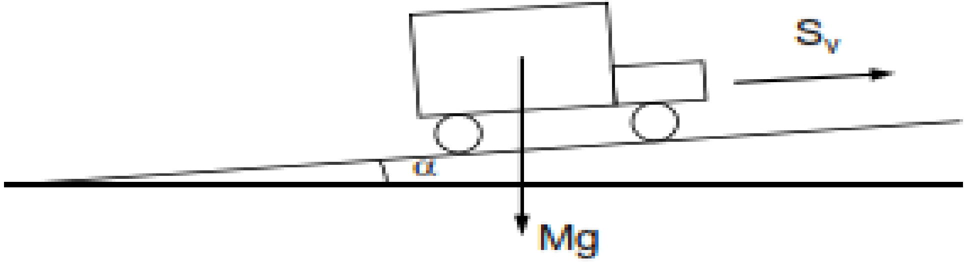
F_R = Rolling frictional force (= C_R Mg cos(α) ; $C_R \sim 0.015$)

F_D = Aerodynamic drag force (= 0.5 $\rho_a S_v^2 C_D A_v$; $C_D \sim 0.3$)

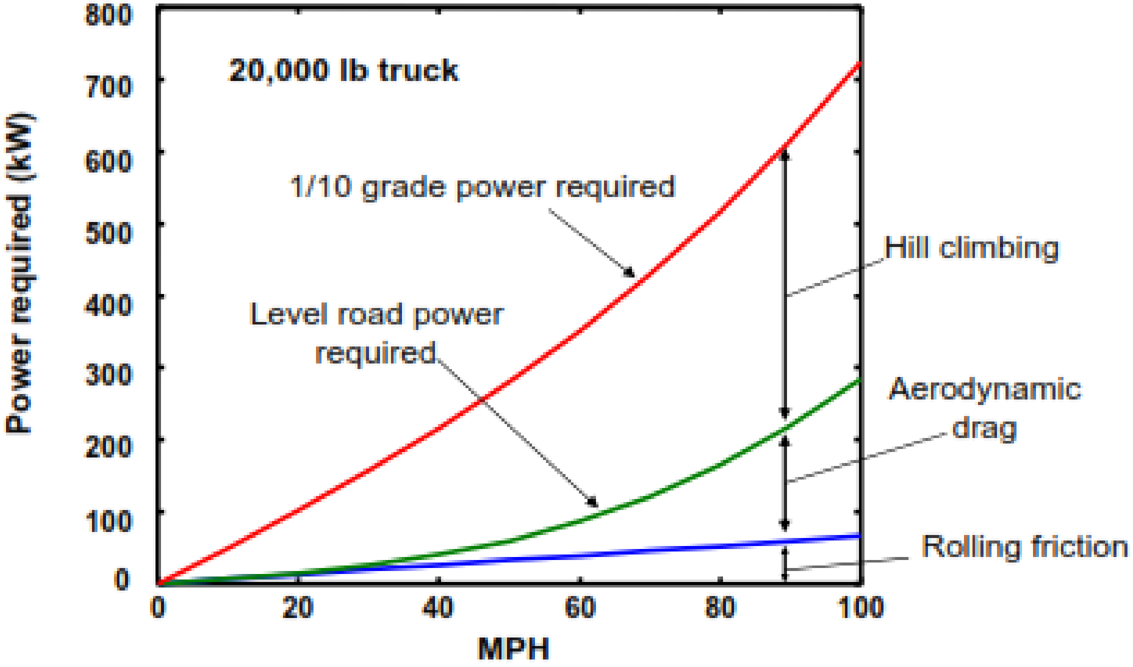
F_a = Force to provide acceleration (= Ma)

F_C = Force for climbing incline = (Mg sin(α)) ; negative for downhill

S_v = Vehicle speed



Truck Road Load Requirement



Vehicle Road Load Requirement

Vehicle speed and engine rpm are related

$$S_V = \frac{N \pi d}{G.R.}$$

Sv = Vehicle speed

N = Engine revolution per second (= RPM / 60)

G.R. = Overall gear ratio

d = External diameter of tire

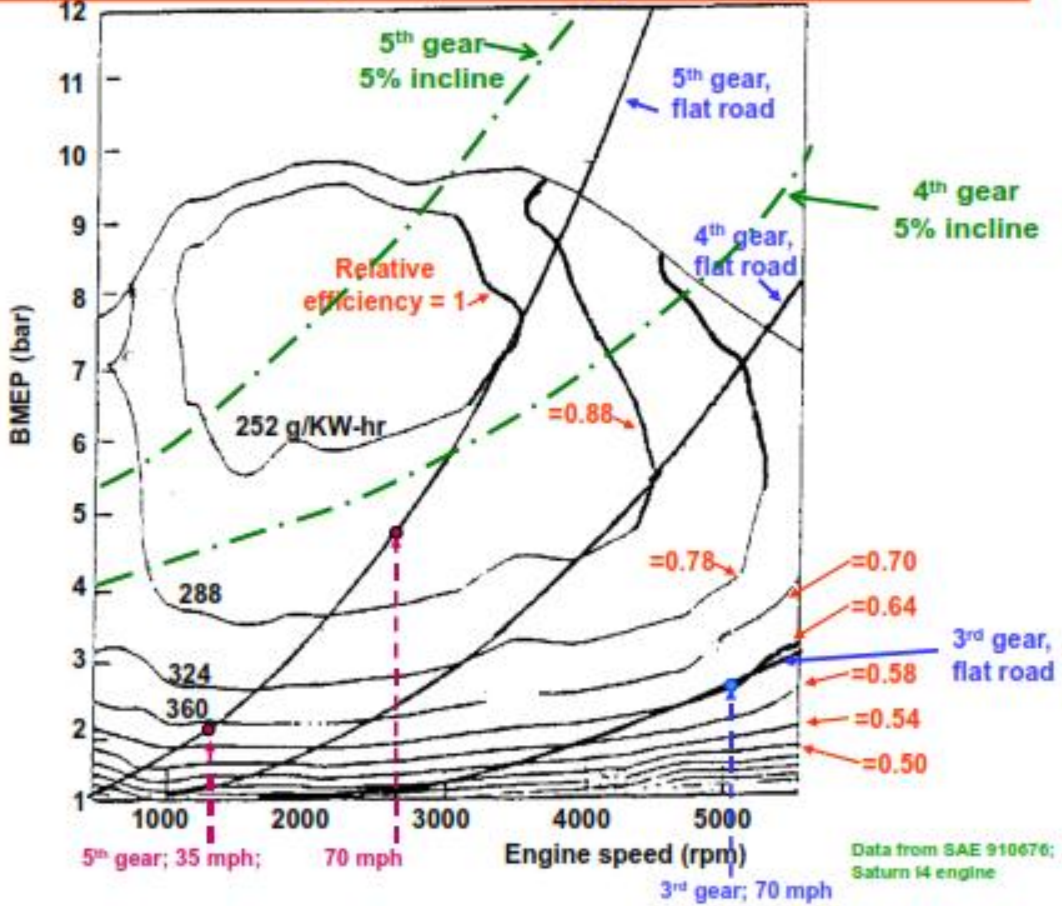
BMEP of engine

$$BMEP = \frac{P_b}{V_D N / n_R}$$

V_D = Engine displacement

η_R = 1 for two-stroke engine; 2 for four-stroke engine

Passenger car SI engine map



Data from SAE 910676;
Saturn I4 engine

Combustion Stoichiometry

Air: Oxygen 21%, Nitrogen (nitrogen + argon) 79%

Fuel: Hydrocarbons (C_aH_b), oxygenates ($C_aH_bO_c$)

Examples:

		LHV
Gasoline	$C_nH_{1.87n}$	44 MJ/kg
Diesel fuel	$C_nH_{1.75n}$	43 MJ/kg
Natural gas (mostly methane)	$CH_{3.8}$	45 MJ/kg
Coal	$C_nH_{0.8n}$	30 MJ/kg
Methanol	CH_3OH	20 MJ/kg
Ethanol	C_2H_5OH	26 MJ/Kg

(LHV = Energy released per unit mass of fuel without recovery of the heat of vaporization of the water vapor in the combustion products)

Stoichiometric Combustion

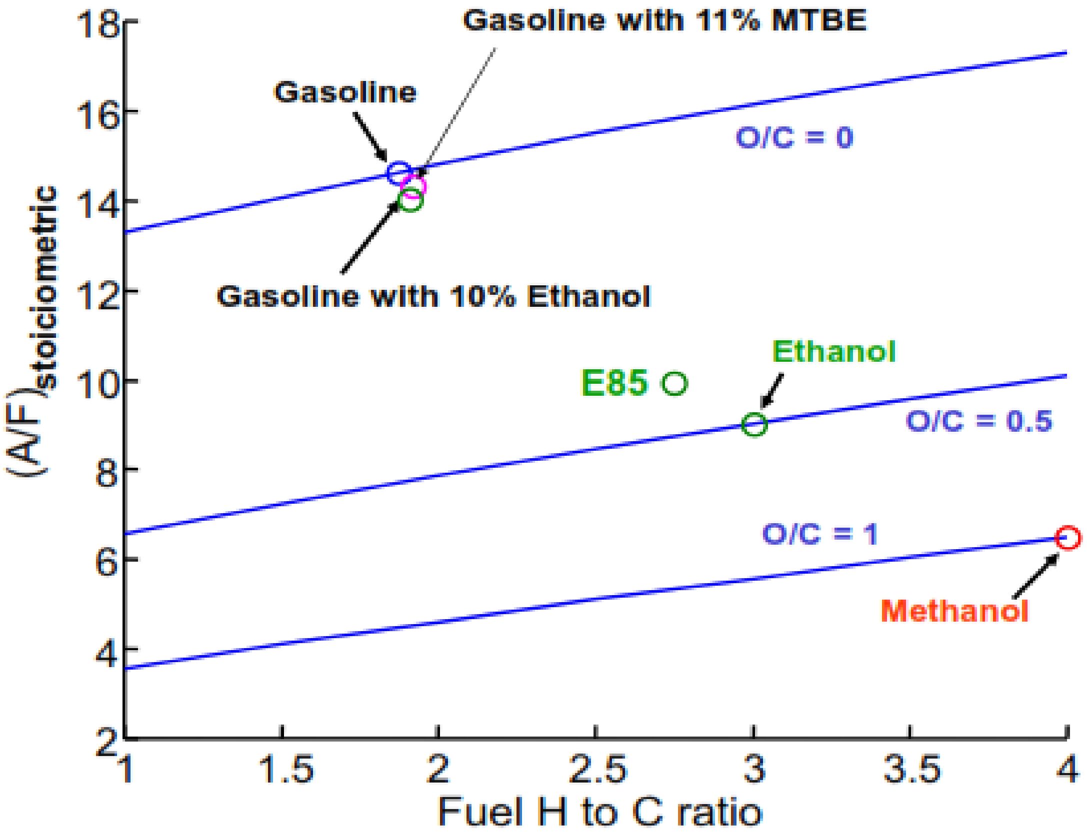
$$C_aH_bO_c + \frac{1}{2}\left(2a + \frac{b}{2} - c\right)(O_2 + 3.773\ N_2)$$

$$\rightarrow aCO_2 + \frac{b}{2}H_2O + \frac{1}{2}\left(2a + \frac{b}{2} - c\right)x3.773\ N_2$$

For typical petroleum based fuel (c=0):

$(A/F)_{stoich} \sim 14.6$ (range 14.2 to 15)

Stoichiometric requirement for different fuels



Lean and rich combustion

Fuel-lean combustion

- major products: CO_2, H_2O, O_2, N_2
- minor products: HC, CO, H_2, NO

Fuel-rich combustion

- major products: CO_2, H_2O, CO, H_2, N_2
- minor products: HC, O_2, NO

Equivalence ratio: Normalized A/F or F/A ratios:

Fuel-air equivalence ratio, Φ

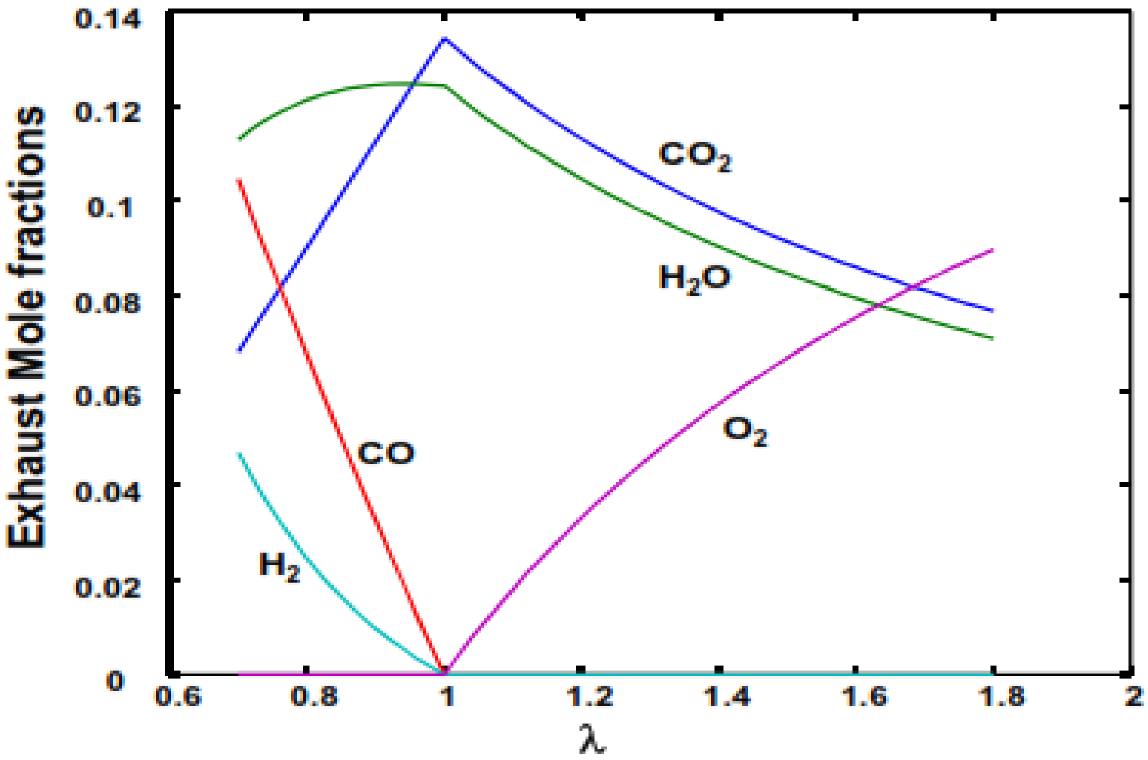
$$\Phi = \frac{F/A}{(F/A)_{\text{stoichiometric}}}$$

Relative air-fuel ratio λ

$$\lambda = \frac{A/F}{(A/F)_{\text{stoichiometric}}}$$

$$\lambda = \frac{1}{\Phi}$$

Exhaust composition (fuel $CH_{1.85}$)



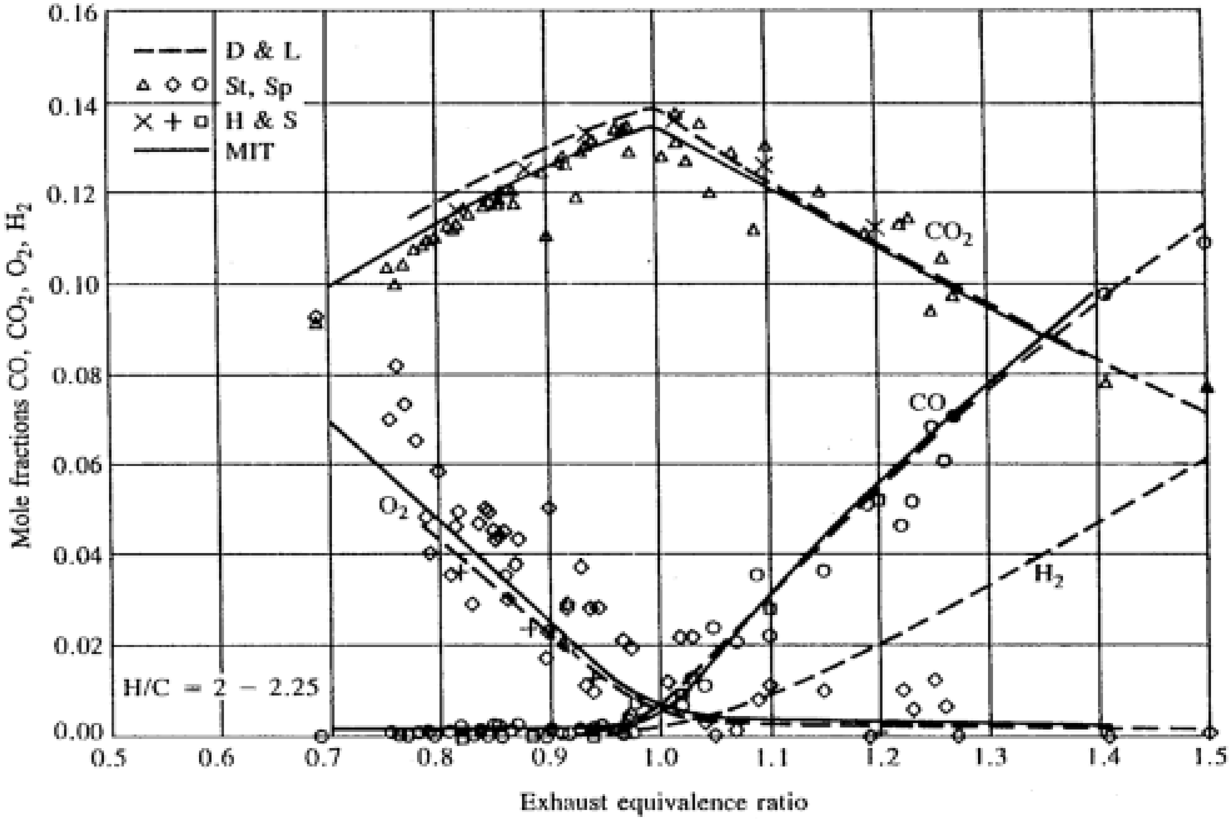


Figure 4-20 Spark-ignition engine exhaust gas composition data in mole fractions as a function of fuel/air equivalence ratio. Fuels: gasoline and isooctane, H/C 2 To 2.25. (From D'Alleva and Lovell,[24] Stivender,[25] Harrington and Shishu,[26] Spindt,[27] and data from the author's laboratory at MIT.)

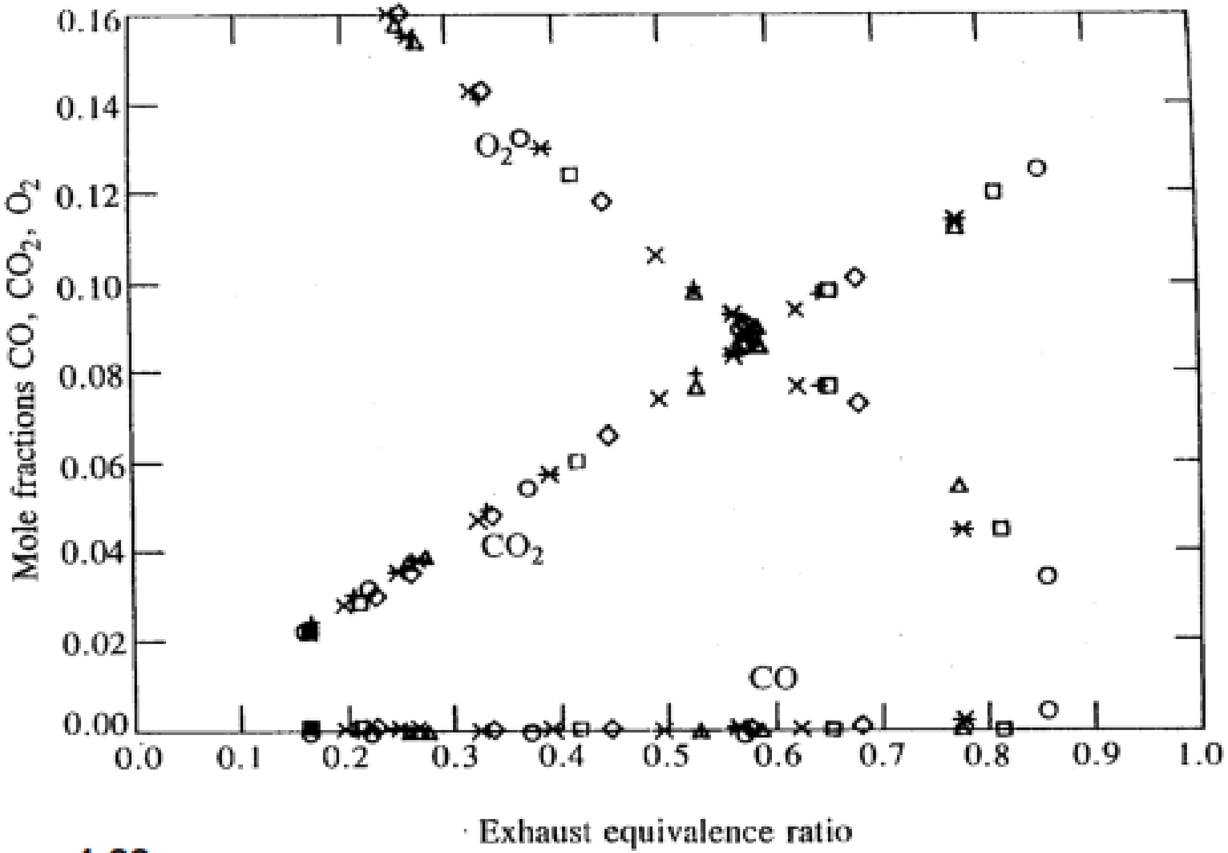
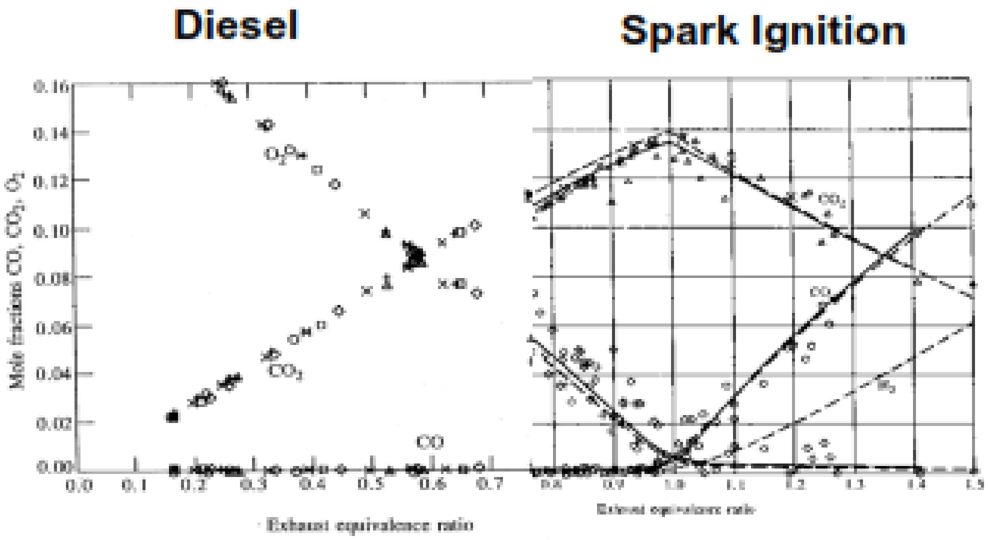


Figure 4-22

Exhaust gas composition from several diesel engines in mole fractions on a dry basis as a function of fuel/air equivalence ratio.[31]

Φ dependence of exhaust major species

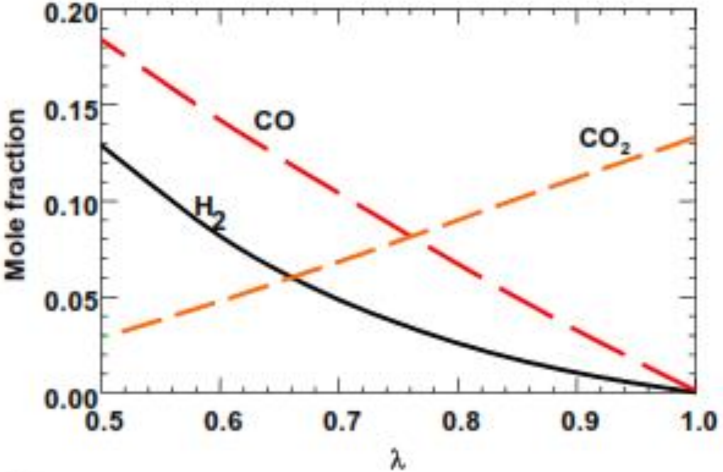


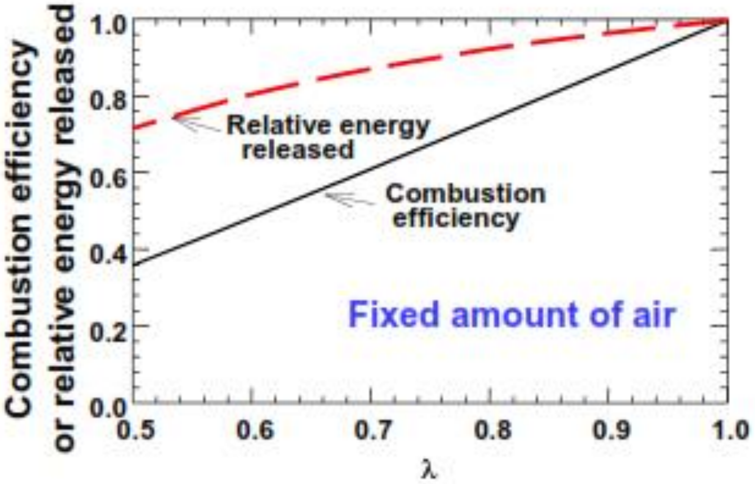
Diesel

Spark Ignition

Superposition of Figures 4-20 and 4-22

Gasoline fuel-rich combustion





For fuel rich combustion, empirically

$$\frac{[H_2O][CO]}{[H_2][CO_2]} = 3.5 \text{ to } 3.7$$

where [ ] denotes molar concentration

Value corresponds to equilibrium composition of water-shift reaction at ~ 1740°K

$H_2O + CO \leftrightarrow H_2 + CO_2$

Equilibrium combustion products: Dissociation effects

P=30 atmospheres

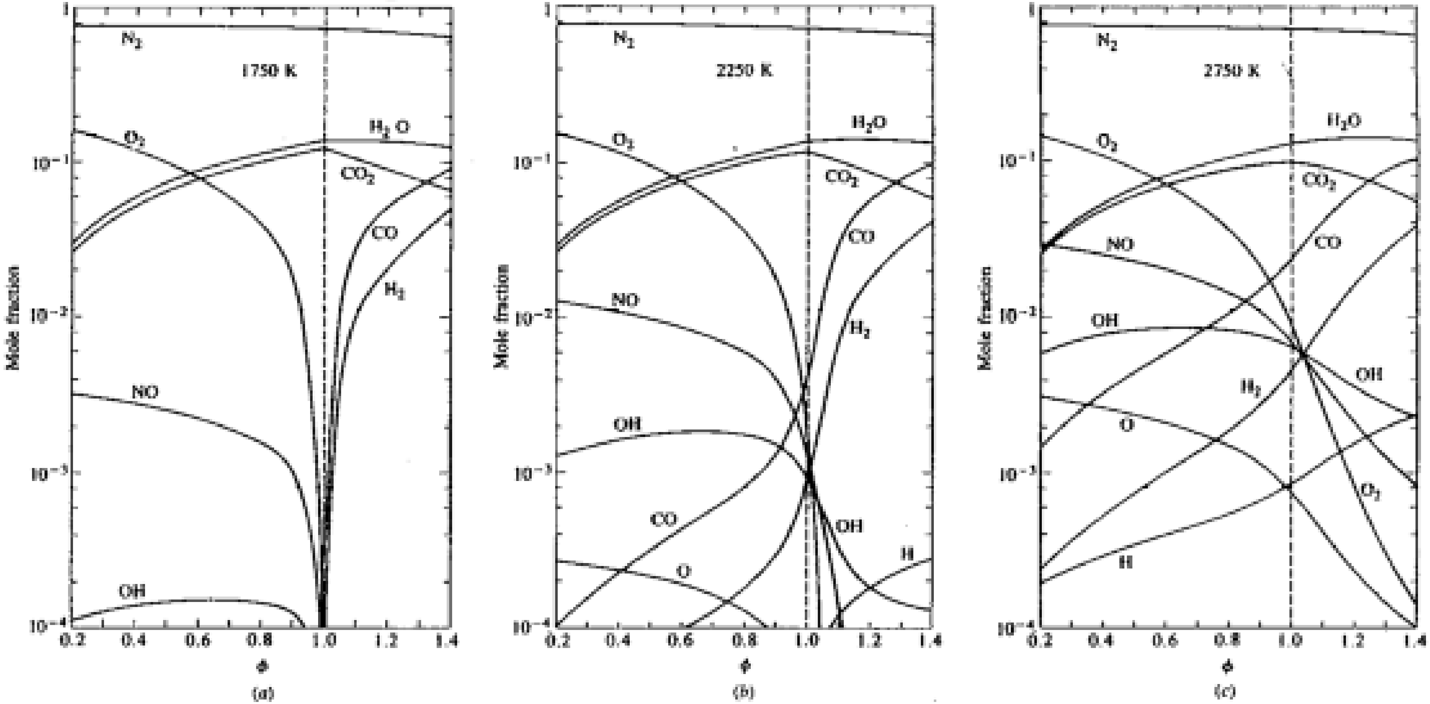


FIGURE 3-10
Mole fractions of equilibrium combustion products of isooctane-air mixtures as a function of fuel/air equivalence ratio at 30 atmospheres and (a) 1750 K; (b) 2250 K; and (c) 2750 K.

Engine Cycles

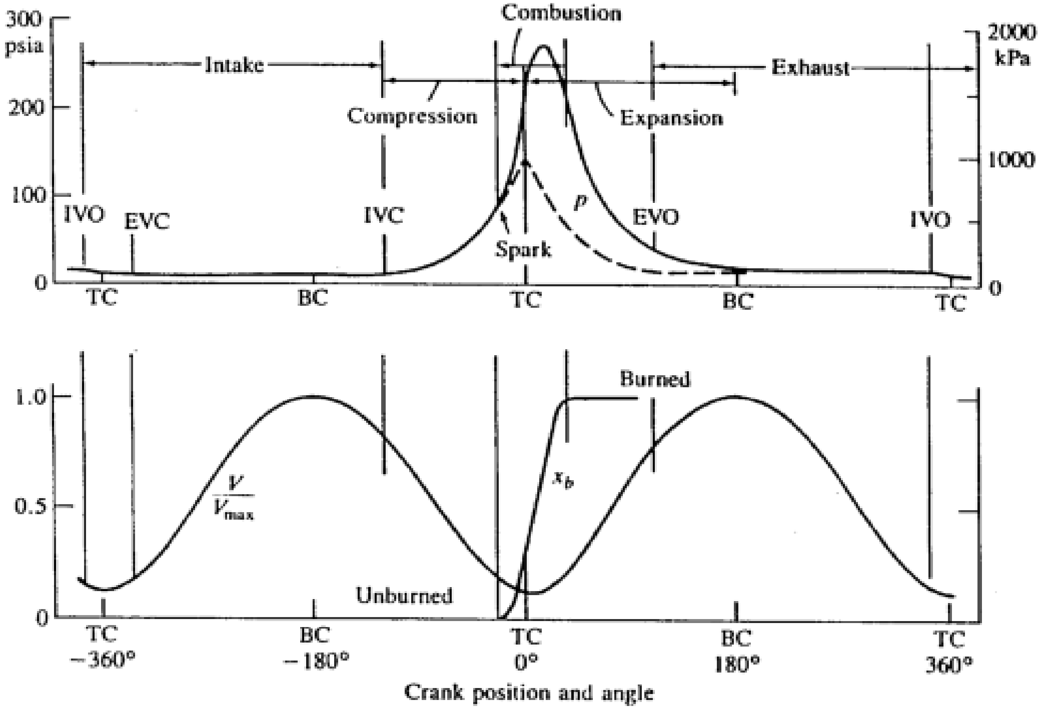
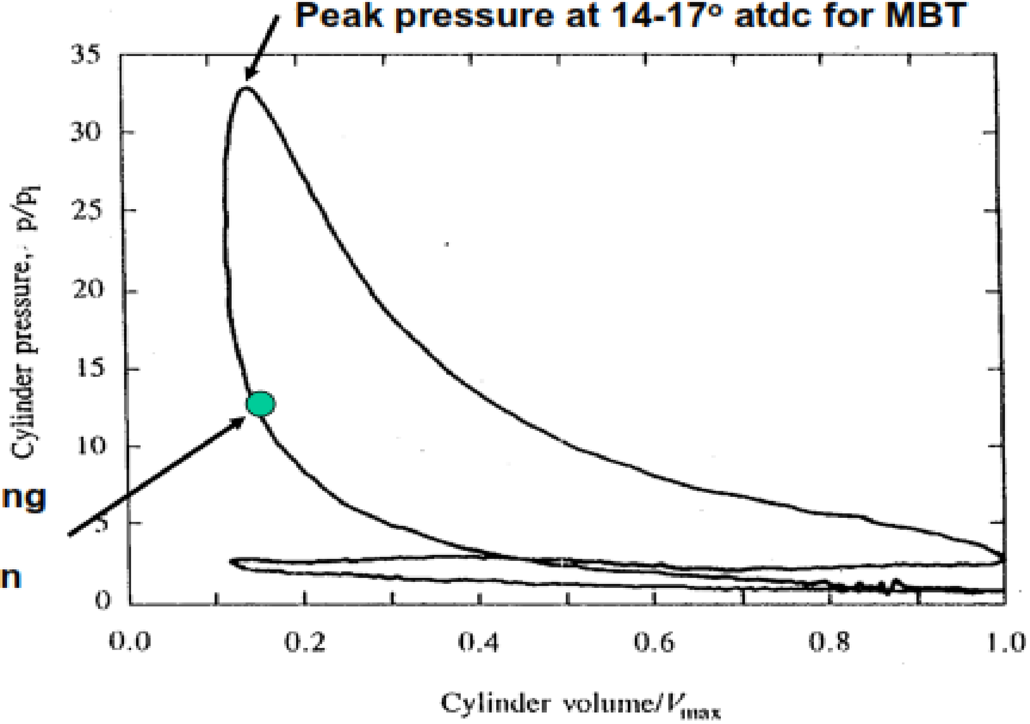


Figure 1-8

Sequence of events in four-stroke spark-ignition engine operating cycle. Cylinder pressure *p* (solid line, firing cycle; dashed line, motored cycle), cylinder volume V/V_{max}, and mass fraction burned x_b are plotted against crank angle.

Pressure-volume diagram



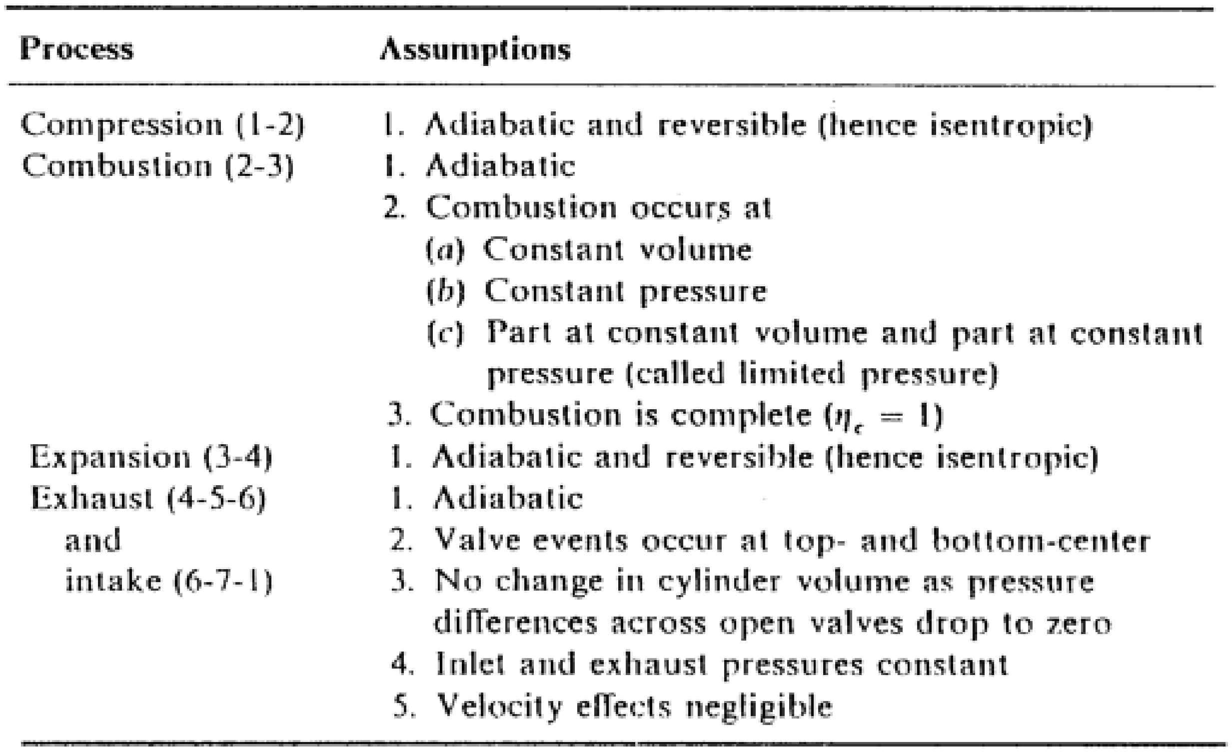
Peak pressure at 14-17° atdc for MBT

Optimal spark timing is a function of operating condition

Fig. 5-1 Pressure-volume diagram of firing SI engine; compression ratio=8.4, 3500 rpm, intake pressure = 0.4 bar, Net IMEP = 2.9 bar

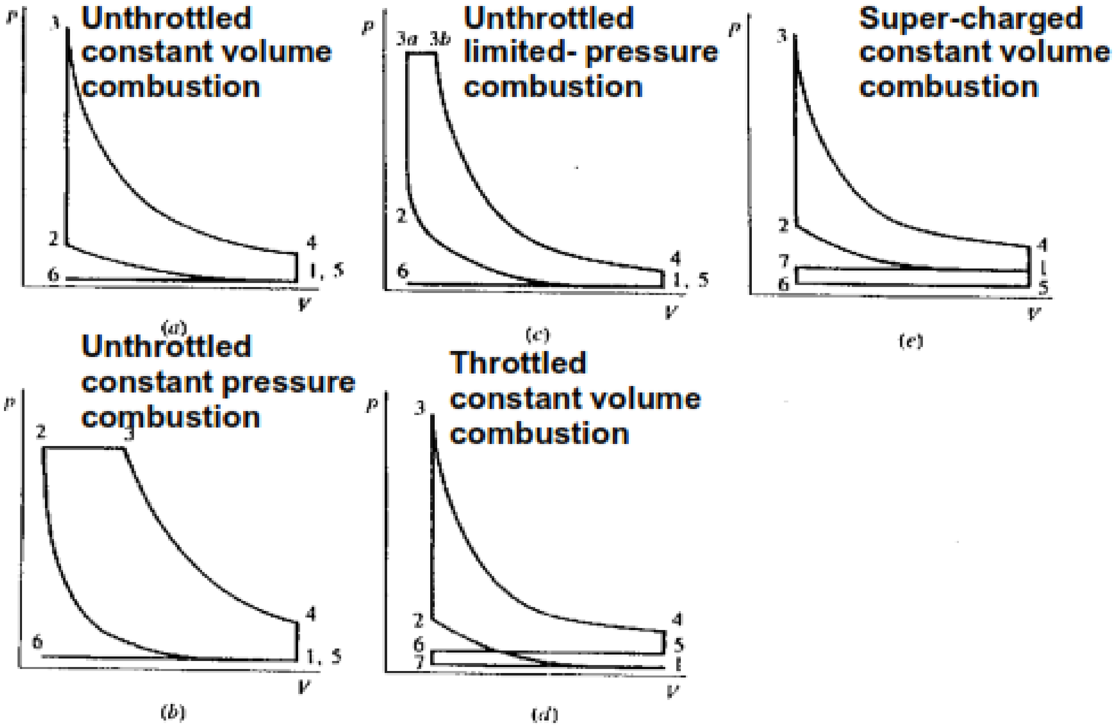
Ideal models of engine processes

Table 5.1



Process	Assumptions
Compression (1-2)	1. Adiabatic and reversible (hence isentropic)
Combustion (2-3)	1. Adiabatic 2. Combustion occurs at (*a*) Constant volume (*b*) Constant pressure (*c*) Part at constant volume and part at constant pressure (called limited pressure) 3. Combustion is complete ($\eta_c = 1$)
Expansion (3-4)	1. Adiabatic and reversible (hence isentropic)
Exhaust (4-5-6) and intake (6-7-1)	1. Adiabatic 2. Valve events occur at top- and bottom-center 3. No change in cylinder volume as pressure differences across open valves drop to zero 4. Inlet and exhaust pressures constant 5. Velocity effects negligible

Different ideal cycles



Unthrottled constant volume combustion

Unthrottled limited- pressure combustion

Super-charged constant volume combustion

Unthrottled constant pressure combustion

Throttled constant volume combustion

Fig 5.2 Pressure-volume diagrams of ideal cycles

Ideal constant volume combustion cycle fuel conversion efficiency

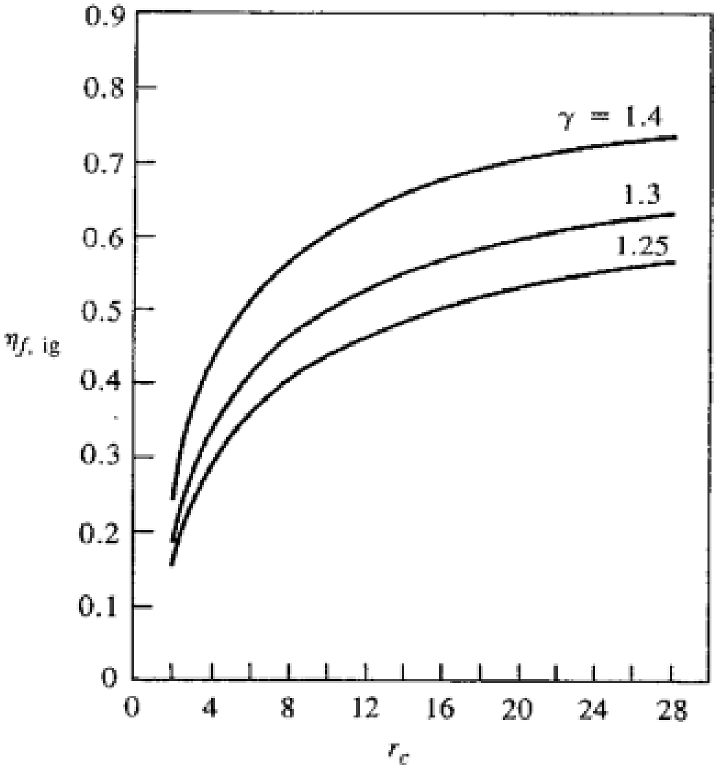


Fig. 5-5

Ideal efficiency

$$\eta_{f,ig} = 1 - \frac{1}{r_c^{\gamma-1}}$$

γ = specific heat ratio

Comparison of fuel conversion efficiency

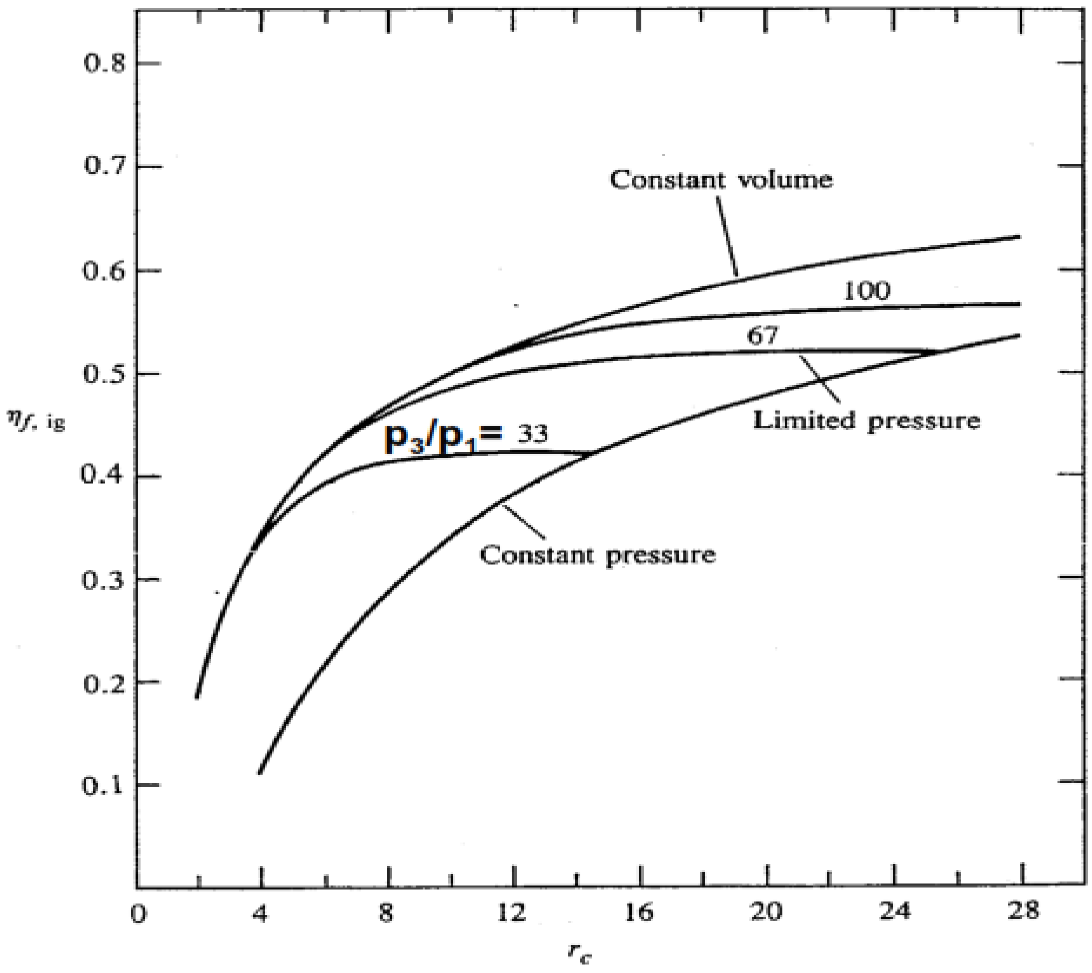


Fig. 5-7 Fuel conversion efficiency as a function of compression ratio for constant-volume, constant-pressure, and limited pressure ideal gas cycles.

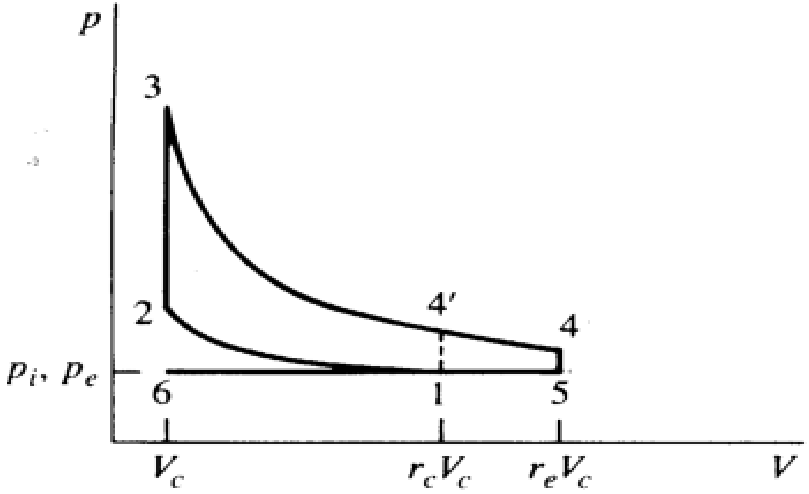
Factors affecting fuel conversion efficiency

These ideal engine cycle analysis results show that expansion ratio r_c and gas composition (through γ the ratio of specific heats) both affect the cycle's fuel conversion efficiency because:

1. The expansion ratio (which may or may not equal to the compression ratio) determines how much work is extracted over the expansion stroke.
2. The higher the value of γ the more the temperature falls during expansion, the larger the energy change and hence the larger the expansion stroke work.
3. The compression stroke work is of order one-sixth of the expansion stroke work so expansion stroke work effects dominate.

Miller cycle

- Late intake valve closing
  - Effective compression ratio is less than expansion ratio
- Advantages
  - Lower compression temperature
    - ➢ Better knock tolerance
    - ➢ Lower NOx emission



- Drawback
  - Reduced trapped charge mass: loss in max power
  - Compensated for by turbo-charging or hybrid operation

Effects of compression ratio

- Theoretical efficiency η_f increases with CR
- SI engine CR limited by knocking to 12 (13 with direct injection
- Practical η_f values decreases at high CR
  - ~~Heat transfer effect~~
  - Crevice effect
  - Dissociation effect
  - Friction
- Other considerations for diesel engines
  - Peak pressure
  - NOx emissions
  - Startability

Practical diesel engines have CR between 14 and 22

Effect of compression ratio on fuel conversion efficiency

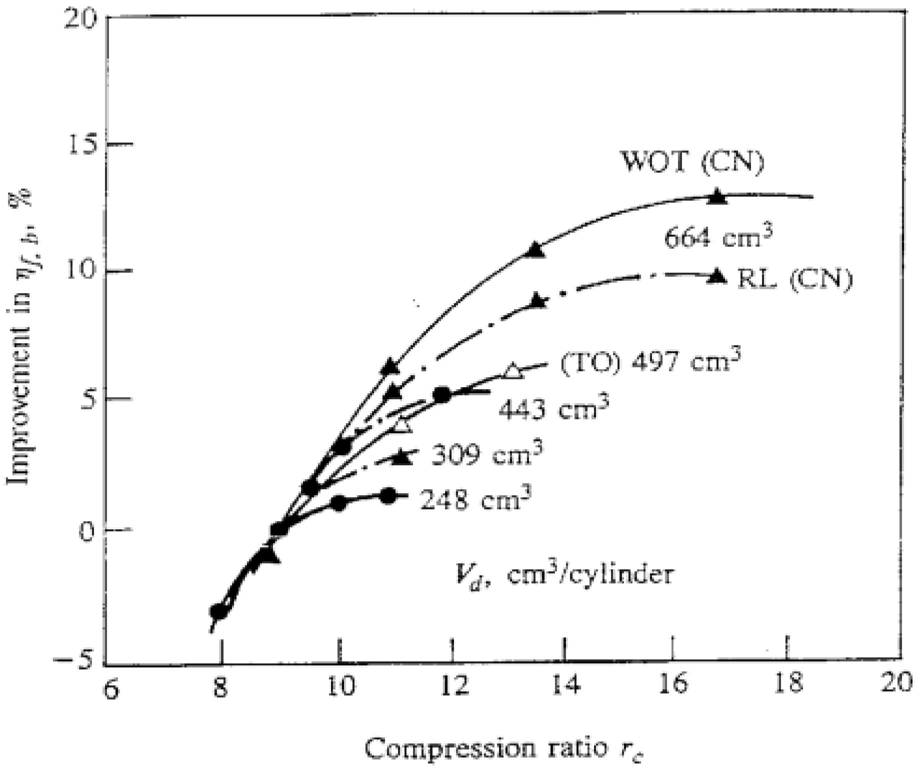


FIGURE 15-14
Relative brake fuel conversion efficiency improvement with increasing compression ratio of spark-ignition engines of different displaced volume per cylinder at part throttle (except top curve at WOT).[19] RL road load. CN,[17] TO.[10]

COMBUSTION AND THERMOCHEMISTRY

Combustion Stoichiometry

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Fuel: Hydrocarbons (C_aH_b), oxygenates ($C_aH_bO_c$)

Examples:

		LHV
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(LHV = Energy released per unit mass of fuel without recovery of the heat of vaporization of the water vapor in the combustion products)

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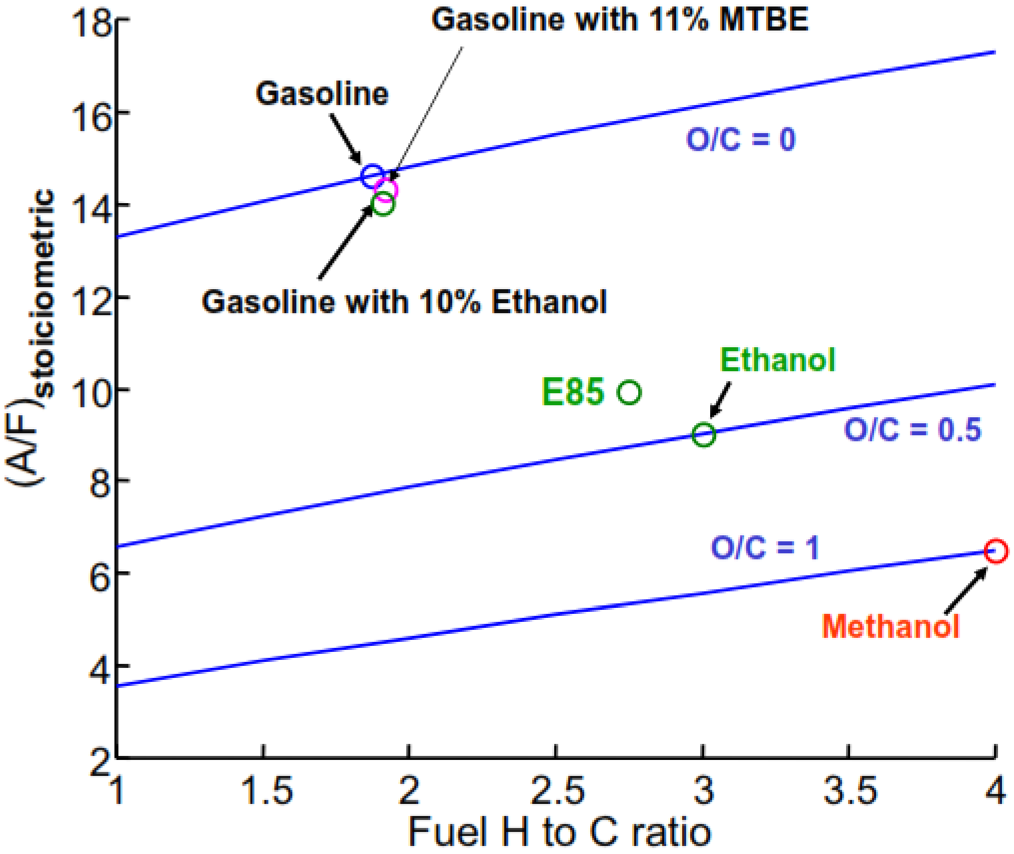
Stoichiometric requirement for different fuels



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- major products: CO_2, H_2O, O_2, N_2
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Relative air-fuel ratio λ

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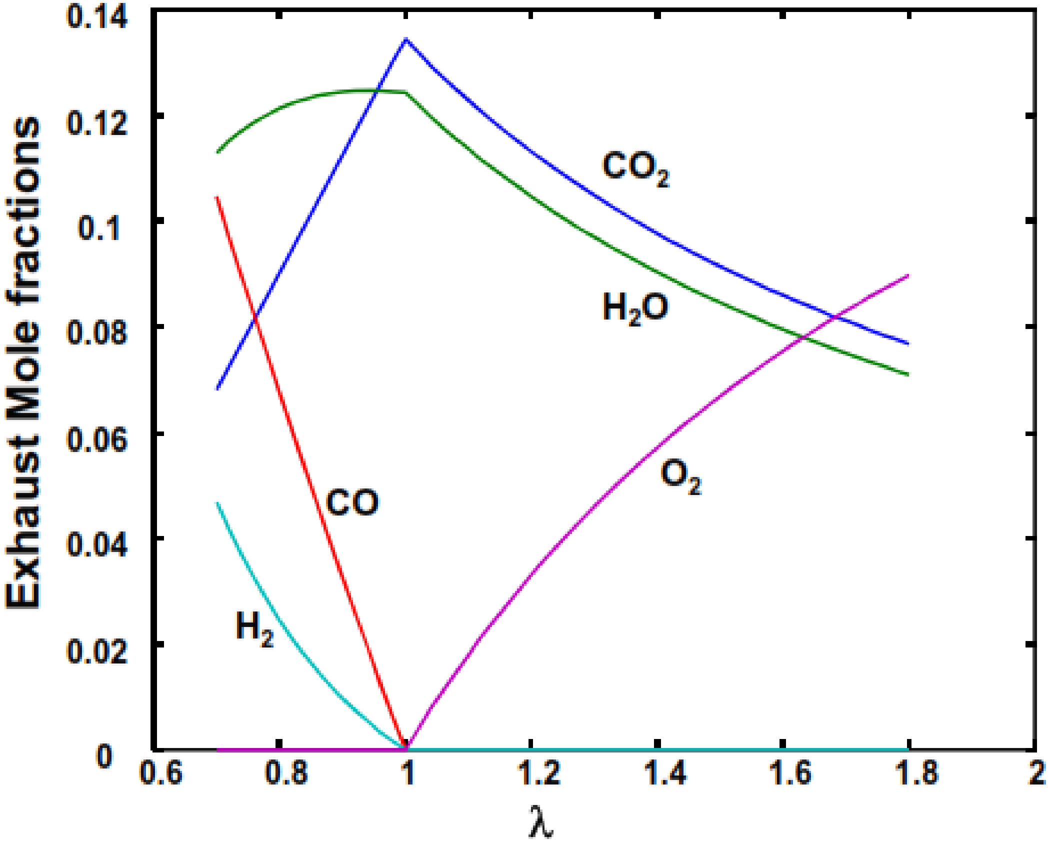
$$\lambda = \frac{1}{\Phi}$$

Exhaust composition (fuel $CH_{1.85}$)



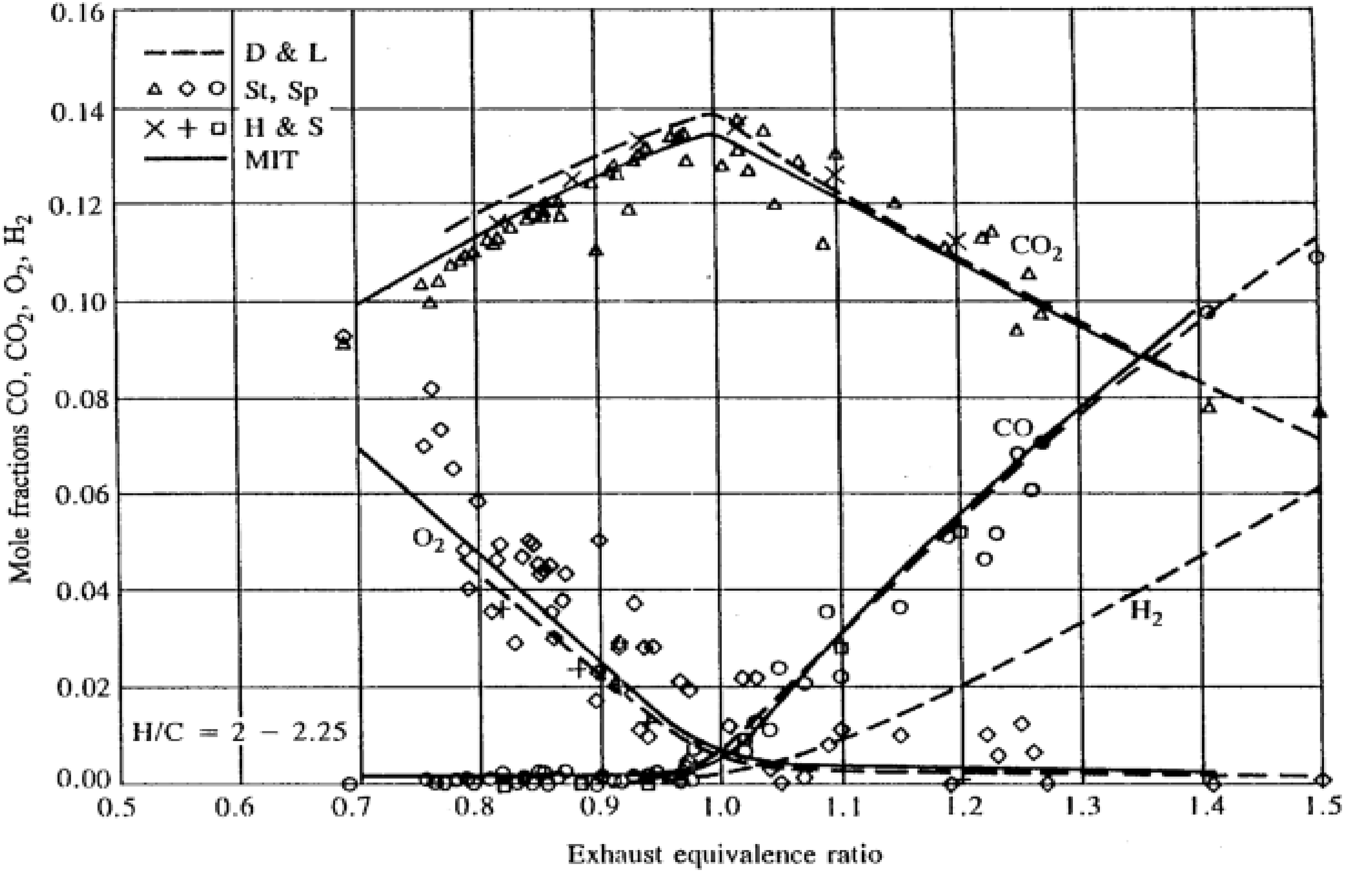


Figure 4-20 Spark-ignition engine exhaust gas composition data in mole fractions as a function of fuel/air equivalence ratio. Fuels: gasoline and isooctane, H/C 2 To 2.25. (From D'Alleva and Lovell,[24] Stivender,[25] Harrington and Shishu,[26] Spindt,[27] and data from the author's laboratory at MIT.)

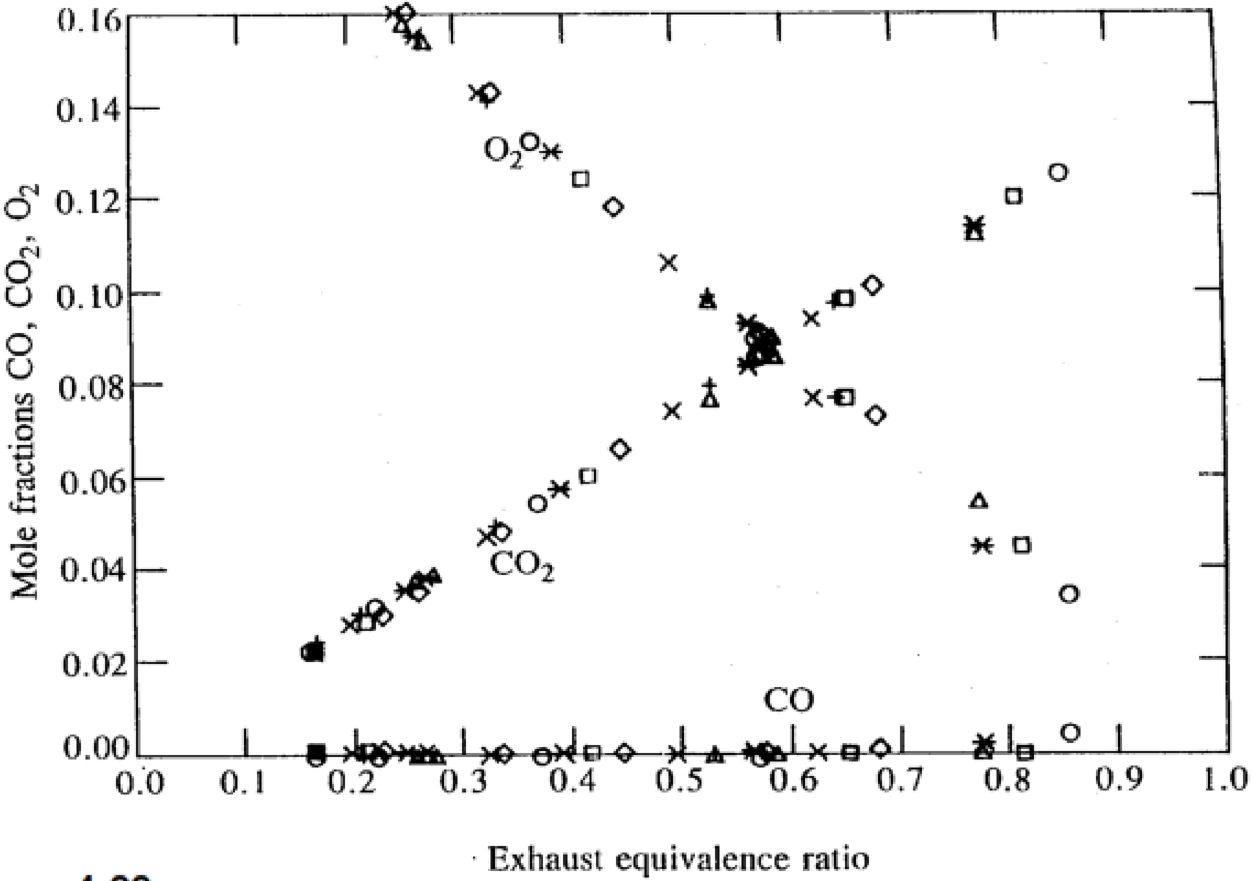
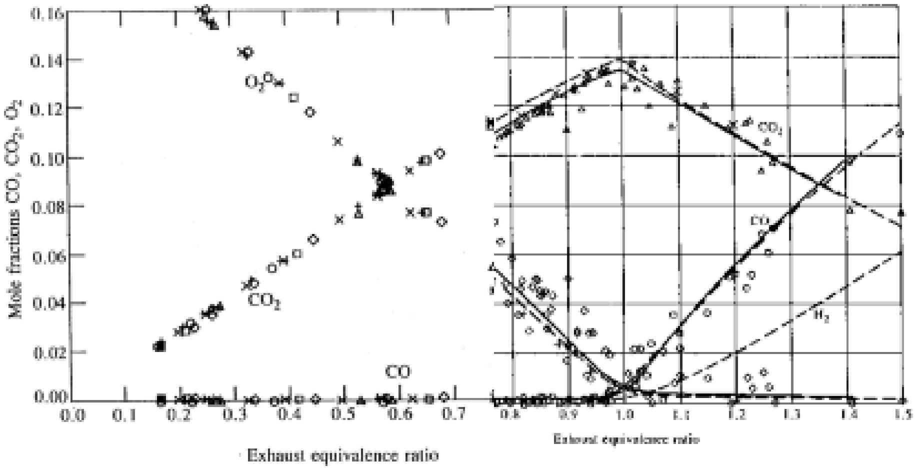


Figure 4-22

Exhaust gas composition from several diesel engines in mole fractions on a dry basis as a function of fuel/air equivalence ratio.[31]

Φ dependence of exhaust major species

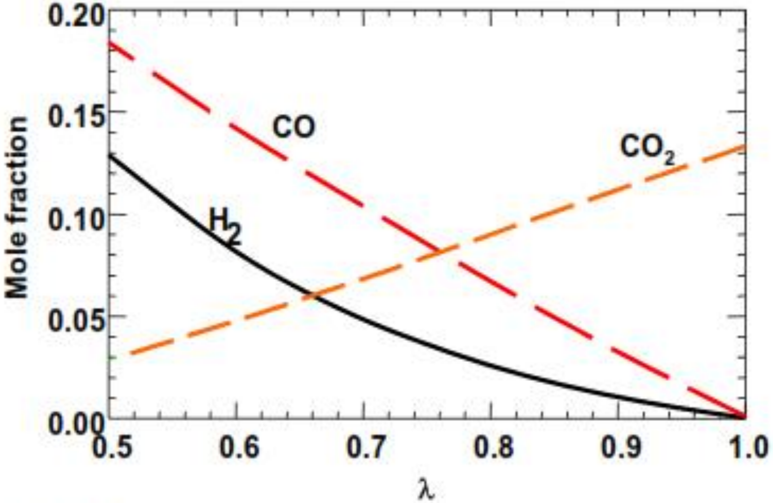


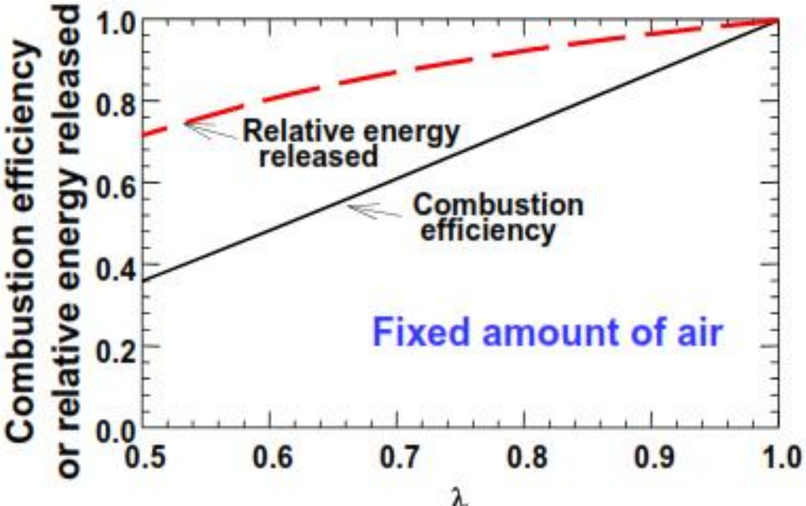
Diesel

Spark Ignition

Superposition of Figures 4-20 and 4-22

Gasoline fuel-rich combustion





For fuel rich combustion, empirically

$$\frac{[H_2O][CO]}{[H_2][CO_2]} = 3.5 \text{ to } 3.7$$

where [ ] denotes molar concentration

Value corresponds to equilibrium composition of water-shift reaction at ~ 1740°K

$H_2O + CO \leftrightarrow H_2 + CO_2$

Equilibrium combustion products: Dissociation effects

P=30 atmospheres

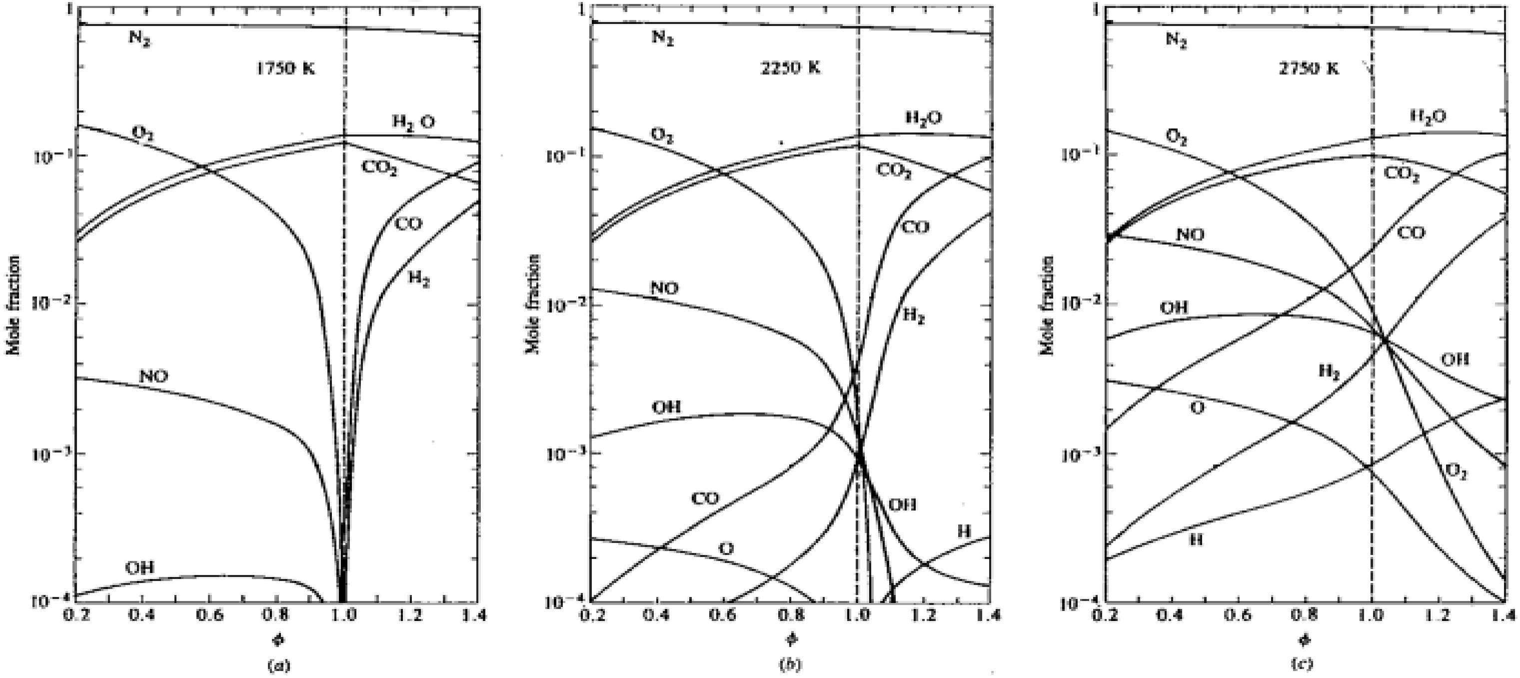
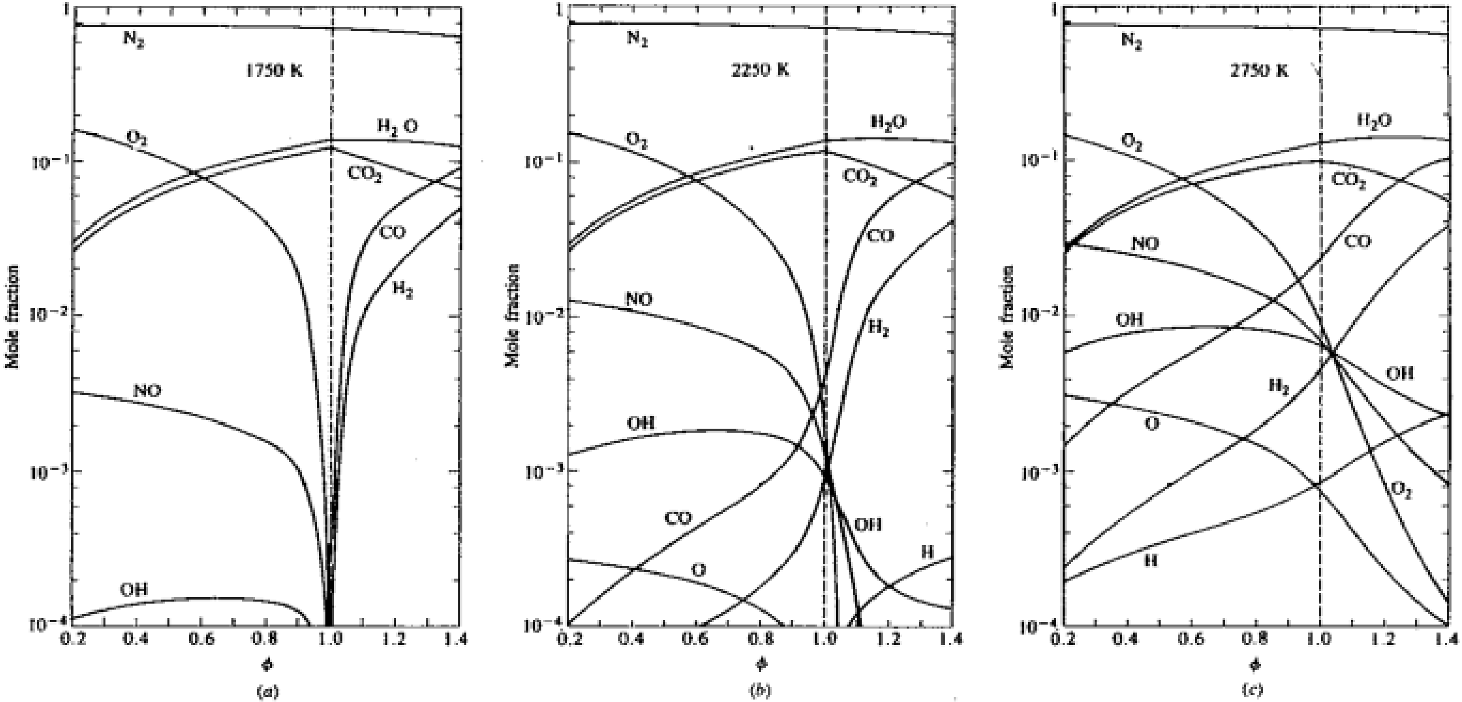


FIGURE 3-10
Mole fractions of equilibrium combustion products of isooctane-air mixtures as a function of fuel/air equivalence ratio at 30 atmospheres and (*a*) 1750 K; (*b*) 2250 K; and (*c*) 2750 K.

GAS PROPERTIES AND FUEL - AIR CYCLE: CYCLE SIMULATION (PDF)

Equilibrium combustion products: Dissociation effects

P=30 atmospheres



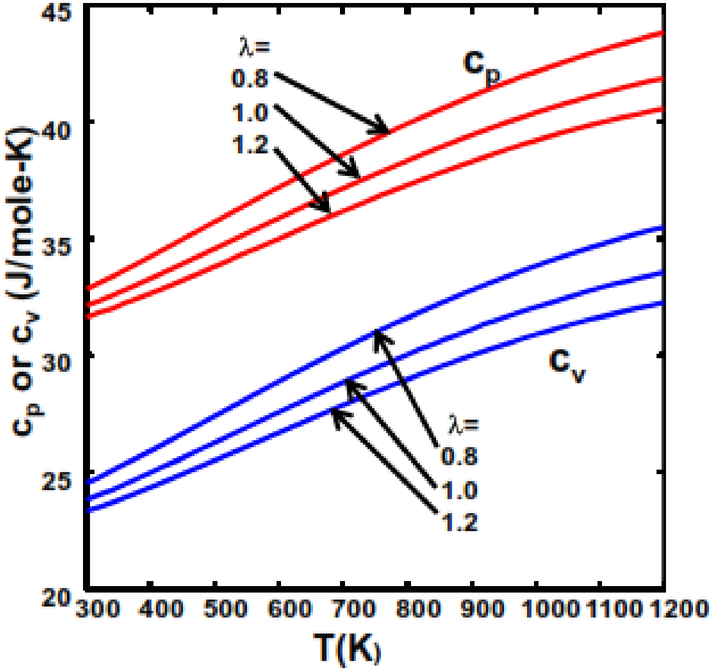
(a) (b) (c)

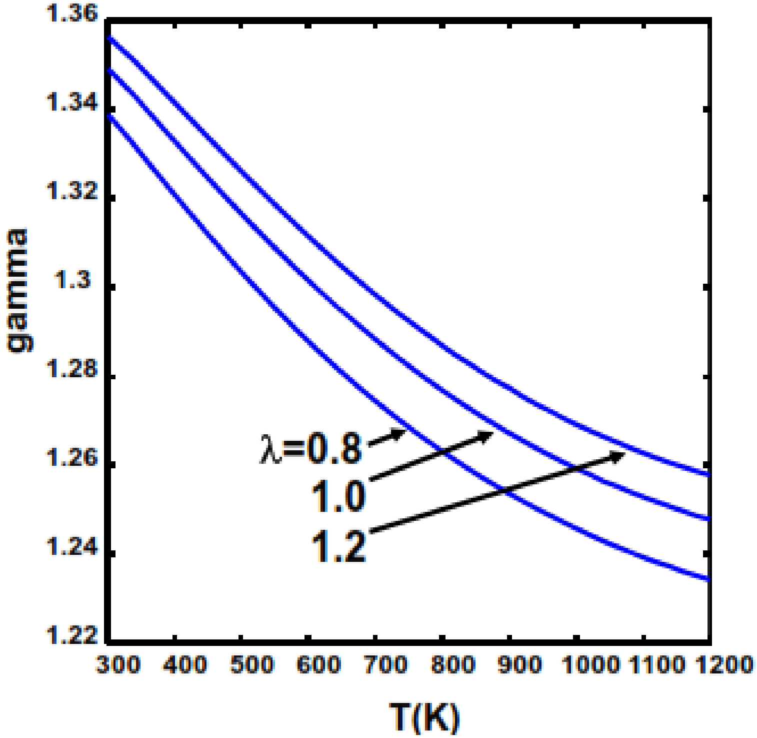
FIGURE 3-10
Mole fractions of equilibrium combustion products of isooctane-air mixtures as a function of fuel/air equivalence ratio at 30 atmospheres and (*a*) 1750 K; (*b*) 2250 K; and (*c*) 2750 K.

Thermodynamic model of engine charge for heat release process

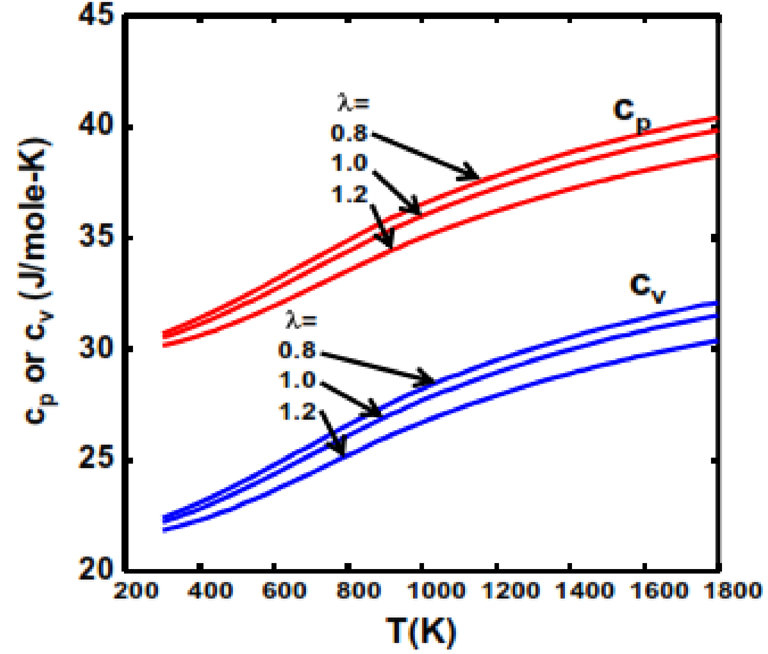
- Unburned gas
  - Ideal gas of frozen composition
- Burned gas
  - At high temperature (T>1740K), as equilibrium mixture
  - At low temperature (T< 1740K), as frozen mixture

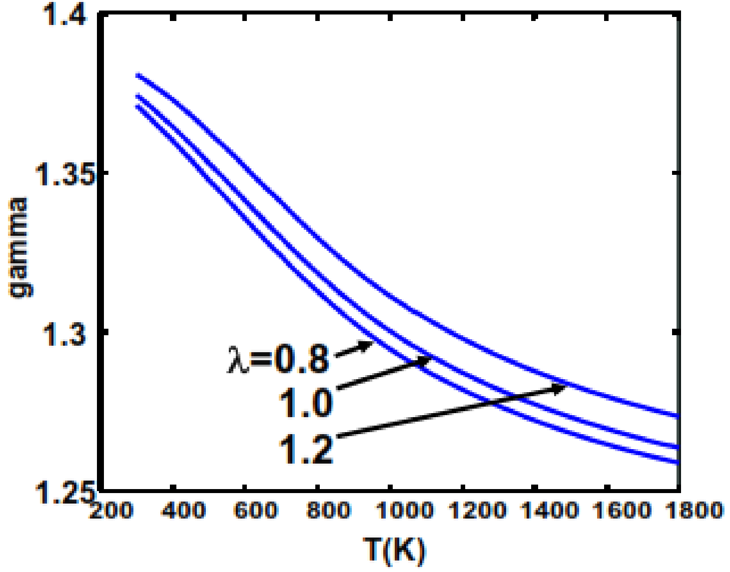
Unburned gas properties for gasoline ($CH_{1.85}$)/air





Burned gas properties for gasoline ($CH_{1.85}$)/air





Composition frozen at 1740K

Fuel-air cycle results

In the Fuel-Air Cycle, the engine processes are still modeled as ideal but the properties of the working fluid (fuel/air/residual gas mixture before combustion, and burned gases in chemical equilibrium after combustion) are described accurately.

The results from this improved cycle analysis model are useful for estimating, approximately, the effects of compression ratio, fuel/air equivalence ratio, and mixture inlet conditions on engine efficiency and performance. The following approximate relationships are useful.

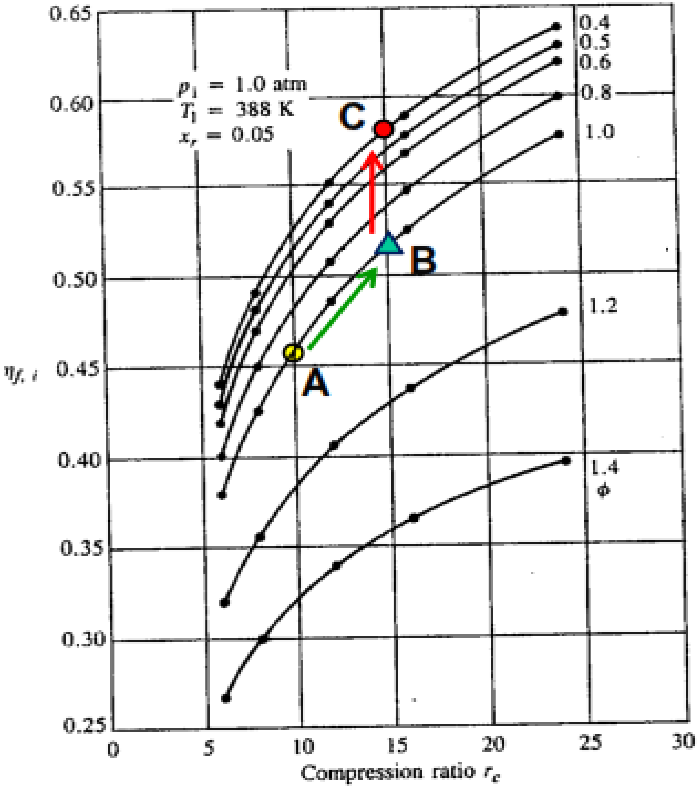
1. The maximum indicated fuel conversion efficiency of an actual engine is about 0.85 times the efficiency of the equivalent fuel-air cycle.

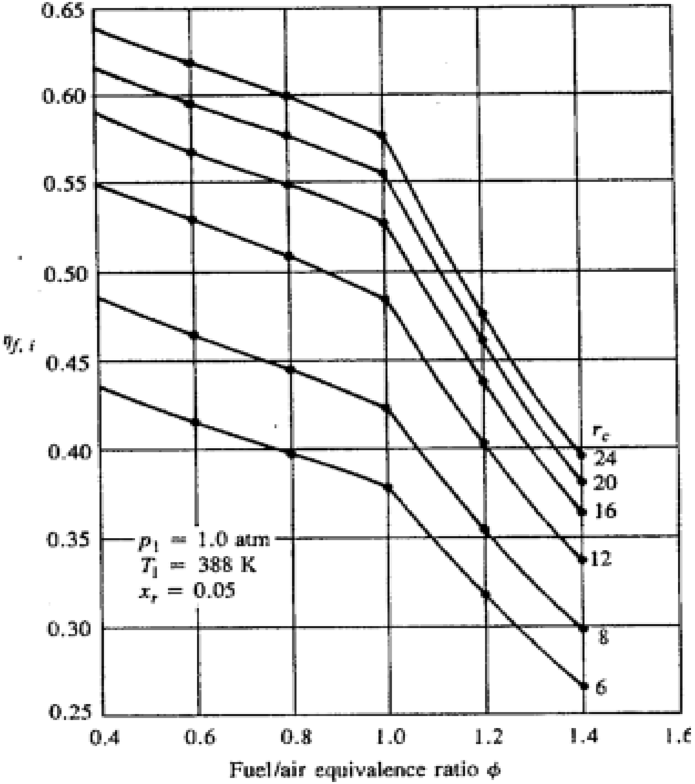
2. Results from change of engine operating condition can be interpreted in terms of percentage change in output values

   Computer codes which accurately simulate the real engine cycle have now been developed and are widely used.

Fuel-air cycle results: $\eta_{f,i}$

Fuel: octene; p_1 = 1 atm, T_1=388 K, x_r=0.05 (Fig. 5.9)



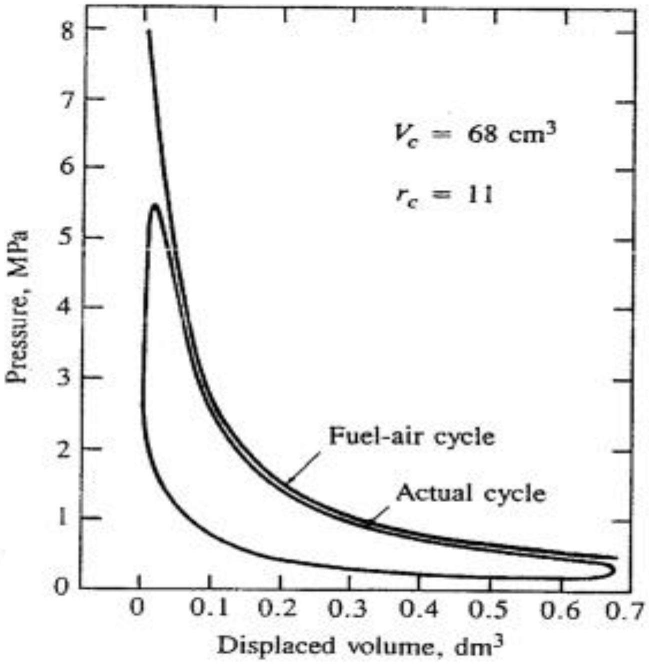


A=SI engine at stoichiometric with r_c=10; C=Diesel at A/F=36 (Φ=0.4) with r_c=15

Real Cycle Effects

1. **Combustion efficiency** $\eta_c = 1 - \frac{\text{exhaust chemical energy as CO, H}_2\text{, HC, soot}}{\text{chemcial energy in inducted fuel}}$

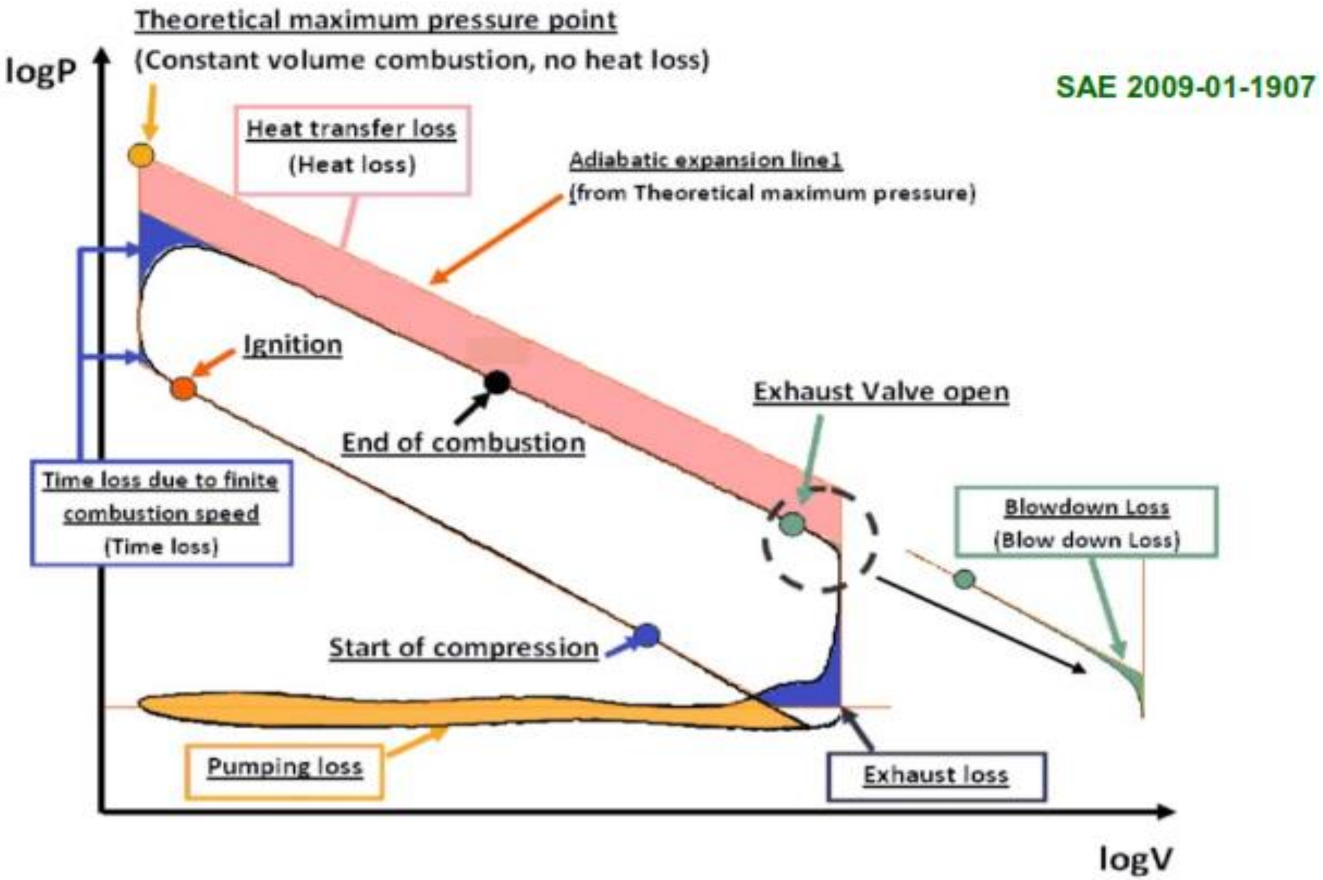
2. **Heat loss, finite combustion time, actual valve timing**



SI engine:
H_2 and CO ~ 1 to 2% of fuel energy
HC ~ 1% of fuel energy
η_c ~ 97-98%
Diesel engine
Very little unburned gas
η_c ~ 99%

Fig. 5-18
Pressure-volume diagram for actual SI engine compared with that for equivalent fuel-air cycle; $r_c = 11$.

Deconstruction of cycle losses



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