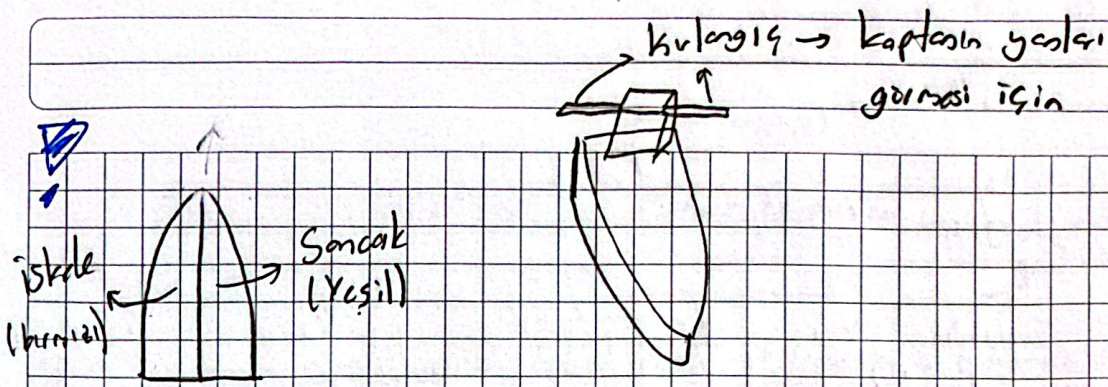




Gros Tona: Geminin tem kapsel hacimleriyle ilgili birimsiz gürmesi (GT)

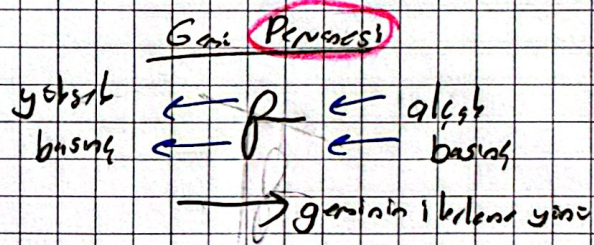
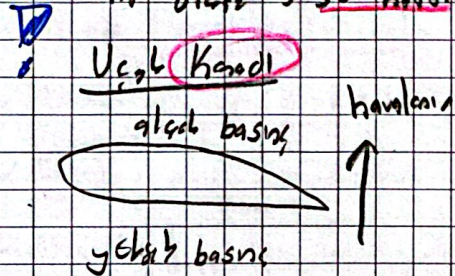
Net Tona: Geminin taşımağı ağırlığı bölün (kibar, tak, yoku kırıntı vb.) hacmidir. (NRT)

Dredweight Tona: Ham yük, yekit, su, kırıntı, yoku ve gemi ağırlığı, eşyaların toplam ağırlığı. Taşımağı ağırlığı. (dwt)

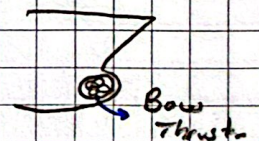
Dephmen Tona: Belirli bir durumda geminin taşıdığı su miktarı.

→ Ton olarak → gemi ağırlığı
→ m³ olarak → su haddi altındaki hacim

→ Vida gibi gemiyi itmeğe
→ çalışır.



Bow Thruster: Ön pervane, Manevra ve yavaşlama için.
Yüksek enerji harcar, jeneratör kullanır, dijitale edilebilir



Bow Thruster: Yüksek Draft → Yüksek Dönüş → Yüksek Pervane → Yüksek Pervane Çapı → Yüksek Dönüş Olmalı

EDN: Tonlar

EDN: Kırıntıya → Yüksek Pervane Çapı → Yüksek Dönüş

(Tonlar için 10m olabilir)

Gemi Pervane Kırıntı: $P = C \times n^3$???

INTRODUCTION TO MARINE PROULSION ENGINES

Dizel yakıtı, sıkıştırılmış havaya gündeştir, hava ısıyla yanar.

1) Diesel Engines

Benzinli den farklı olarak ateşleme bujisi bulunmaz. Karşın benzinden 2 kat fazla sıkıştırılır. (Sıkıştırma oranı)

→ Daha sessiz çalışır.

1) Low speed engines, 60-150 rpm

2) Medium speed engines, 450-800 rpm

3) High speed engines, 1000-3000 rpm

Genelde motor çalışırken hava nemli olduğundan benzinli motorlardaki ateşleme bujisinin elektirik akımına gerek vardır. Bu sebepten benzin motoru kullanılmaz.

Low speed engines:

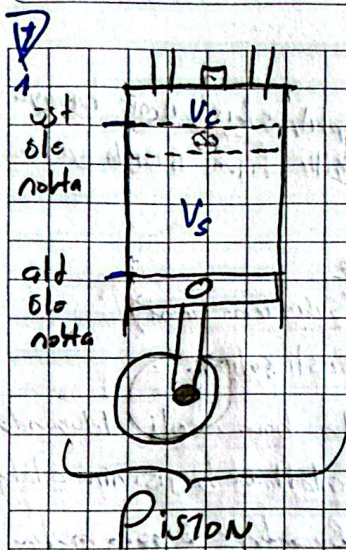
- ✓ Ideal for container ships, bulk carriers, tankers, LNG carriers
- ✓ High power output. Max available from one unit: 16000 kW - 80000 kW
- ✓ Economic, low Specific Fuel Oil Consumption (SFOC) $\sim 120 \text{ g/kWh}$
- ✓ High thermal efficiency $\sim 62\%$
- ✓ Allows direct drive of propeller
- ✓ Reliable, easily maintained
- ✓ Able to burn low quality fuels

Medium speed engines:

- ✓ Ideal for passenger ships, RoRo's
- ✓ Max power available: 8000 kW - 23850 kW
- ✓ Poor SFOC $\sim 190 \text{ g/kWh}$
- ✓ Optimum propeller speed selection through a gearbox
- ✓ Multiple engines for one shaft/propeller.
- ✓ More maintenance required due to more moving parts

High speed engines:

- ✓ Ideal for small crafts, fast ferries
- ✓ Max power available from one unit: 235 kW - 9000 kW
- ✓ High SFOC $\sim 240 \text{ g/kWh}$
- ✓ High power density



Deplasman hacmi:

$V_{\text{deplasman}} = V_c + V_s$ $\rightarrow$ Stroke hacmi

Sıkıştırma Oranı $(E \equiv r) = \frac{V_T}{V_c} = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c}$

$W = P \cdot V_s$

$V_s = \frac{\pi D^2}{4} \times L$

Comparison at the same power output?

	Power kW	rpm	Weight ton	Power density kW/ton	Density kg/m ³
Low Speed	7000	100	313	23	0.90
Medium Speed	7000	520	153	50	1.25
High Speed	7000	1300	21	300	1.9

Gas Turbines:

- ✓ Ideal for fast ferries and naval ships.
- ✓ Very high SFOC
- ✓ High power density
- ✓ Utilised in combined power systems

Steam Turbines

- ✓ Appropriate for large power output demands
- ✓ Higher SFOC than diesel
- ✓ Able to utilize coal or nuclear fuels
- ✓ Utilised in combined systems with gas turbines
- ✓ Heavy, requires large space

ENGINE CLASSIFICATION

Fuel:

- ✓ Diesel
- ✓ Gasoline
- ✓ Natural Gas

Ignition:

- ✓ Compression
- ✓ Spark

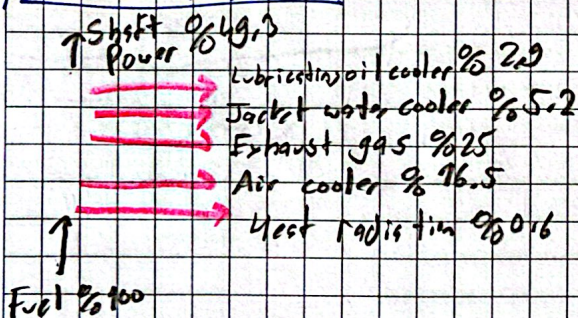
Working Cycle:

- ✓ Four stroke
- ✓ Two stroke

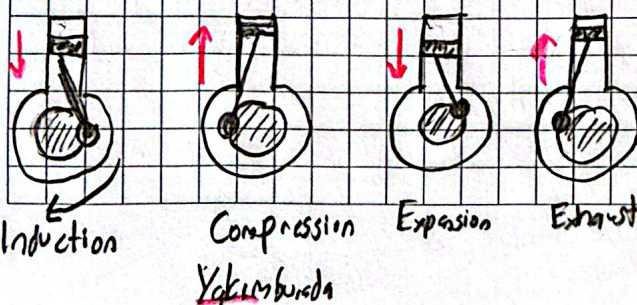
Engine Design:

- ✓ In-line
- ✓ V-Type
- ✓ Opposed

Heat Balance Diagram

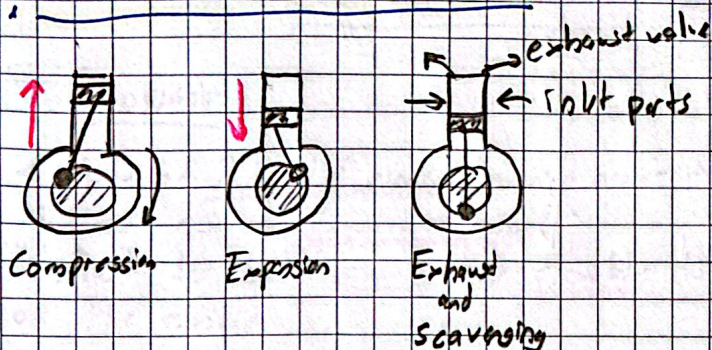


Four-Stroke Diesel Engine



TWO-STROKE DIESEL ENGINE

4 zamanlı den 935 li, 360 derece sores bite

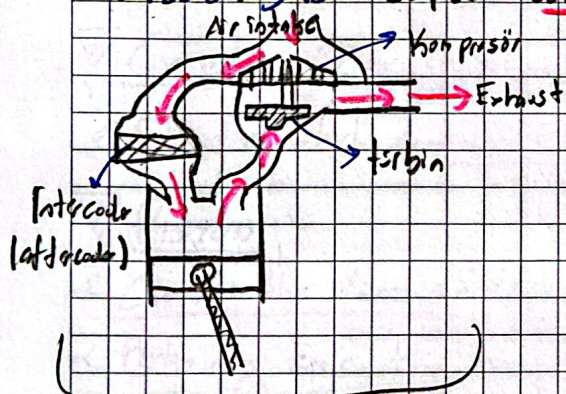


SUPERCHARGER

Used to increase power of a diesel engine by raising intake pressure, allowing more fuel to be burned per cycle. $P \uparrow$

Mechanical Superchargers: Compressors that are mechanically driven by engine crankshaft.

Turbochargers: Couples a compressor with turbine driven by the exhaust gas.



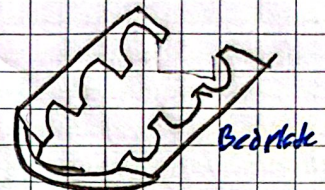
Compressor raises the temperature of the air it sucks, so an inter-cooler (soğutucu) is needed.

Turbo charger

ENGINE COMPONENTS

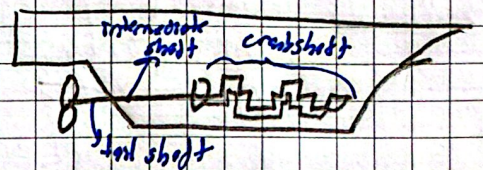
BEDPLATE

- Made of 2 parallel girders running across the length of the engine.
- Works as a foundation block.
- Must be both strong and flexible to support the weight and fluctuating forces of the engine.



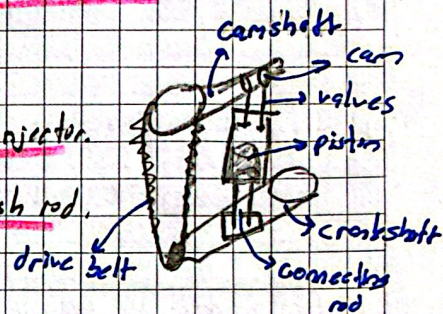
CRANKSHAFT

- Transmitting cylinder power to the propeller shaft.
- Converts the oscillating motion of connecting rod to rotary motion of the shaft.



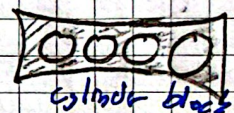
CAM SHAFT

- Operates three main valves: Inlet, Exhaust and Fuel Injector.
- Operates with the help of cam follower, rocker arm, push rod.



CYLINDER BLOCK (ENGINE BLOCK)

- Structure which contains the cylinders and other components.



PISTON

- Converts the force of expanding gases during combustion process to mechanical energy.
- Piston rings seals the combustion chamber, prevents leak.

CONNECTING ROD

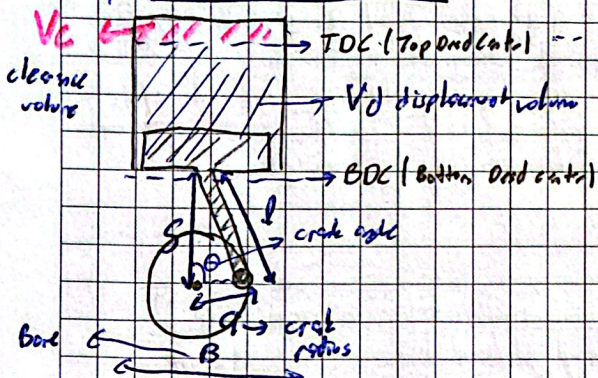
- Converts the motion of the piston to rotary motion of crankshaft.

CYLINDER HEAD

- Main function is to form the top part of combustion space and supporting the valves.

ENGINE SPECIFICATION

CYLINDER GEOMETRY



Displacement, V_d

$$V_d = \frac{\pi B^2}{4} \times L$$

Compression ratio, r_c

$$r_c = \frac{\text{Max Volume}}{\text{Min Volume}} = \frac{V_d + V_c}{V_c}$$

Mean piston speed, S_p

$$S_p = 2 \cdot L \cdot N \rightarrow \text{engine speed}$$

Two times

Cylinder volume at any crank angle:

$$V = V_c + \frac{\pi B^2}{4} (1 + \cos \theta)$$

If $B = L \rightarrow$ square engine

$L > B \rightarrow$ under square engine

$L < B \rightarrow$ over square engine

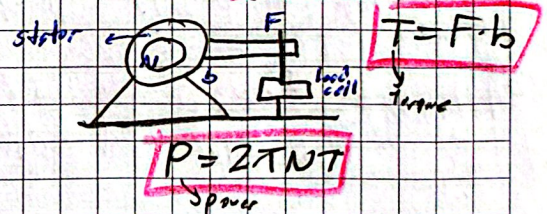
BRAKE TORQUE AND POWER

Torque is a measure of an engine's ability to do work $\rightarrow$ Measured with a dynamometer.

Power is the rate at which work is done.

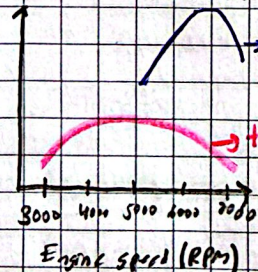
Brake power, P_B is measured at the output shaft.

$P_{\text{generated}} > P_B \rightarrow$ losses



$$P = 2\pi NT$$

Spring



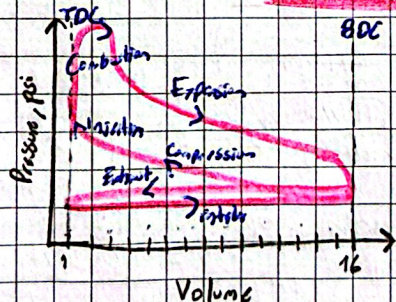
Because friction increases with engine speed.

Engine is unable to ingest a full charge of air at higher speeds.

INDICATED WORK AND POWER

$$P_i = \frac{W_{ci} \cdot N}{n_R}$$

$W_{ci} \rightarrow$ crankshaft speed
 $n_R \rightarrow$ 2 for 4-stroke
1 for 2-stroke
crank revolutions per cycle

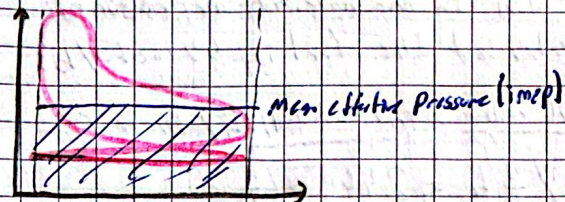


Indicated work per cycle, W_{ci}

$$W_{ci} = \oint P dV$$

MEAN EFFECTIVE PRESSURE (mep)

∝ fictitious constant pressure that would produce the same work per cycle if it acted on the piston during the power stroke.



$$mep (kPa) = \frac{P \cdot n_R}{V_d \cdot N}$$

∝ imep: indicated mean effective pressure, calculated

∝ bmep: brake mean effective pressure, calculated

Bmep:

$$Max Bmep = \frac{W_b}{V_d} = \frac{2\pi T n_B}{V_d}$$

$\rightarrow 2, 4, 6$

∝ Greater power can be generated by increasing displacement.

But this makes the engine bigger. Most modern engines are smaller but run at high speeds, often turbocharged or supercharged to increase mep.

∝ Closing the throttle $\rightarrow$ bmep ↓

∝ Higher max Bmep $\rightarrow$ more torque (at given displacement)

∝ Higher max Bmep $\rightarrow$ smaller engine (at given torque)

∝ Higher max Bmep $\rightarrow$ stresses ↑ temperatures ↑ engine life ↓

∝ 2-strokes have twice the power at 4-strokes for same bmep.

SPECIFIC FUEL CONSUMPTION (sfc)

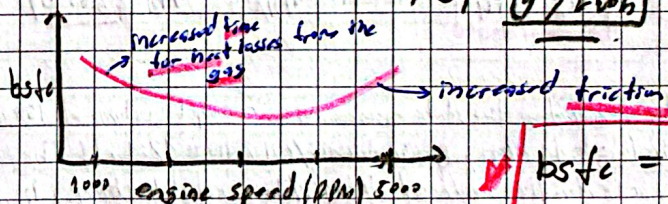
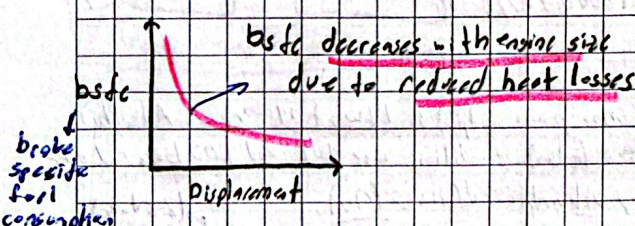
∝ Measure of how efficiently the fuel is used to produce power.

∝ Low sfc is better

$$sfc = \frac{\dot{m}_f}{P}$$

fuel mass flow rate

(mg/s) $\rightarrow$ (g/kwh)



$$bsfc = \frac{\dot{m}_f}{P_b}$$

brake power

∝ Compression ratio ↑ fuel consumption ↓ $\rightarrow$ better thermal efficiency

INDICATED, BRAKE, FRICTION POWERS

Brake

Friction

Indicated

∝ Delivered by engine shaft

∝ Difference between Brake and Indicated

∝ Delivered by the gas

∝ Measured with dynamometer

∝ Due to engine losses

$$\frac{\frac{N}{m^2}}{\frac{J}{s}} \quad \frac{N \cdot s}{J \cdot m^2}$$

~ Examples ~

Ex 1: Four stroke SI engine delivers a brake power of 442 kW with a mechanical efficiency of 85%. Measured fuel consumption is 160 kg of fuel in one hour and air consumption is 410 kg during one sixth of an hour. Heating value of the fuel is 42000 kJ/kg.

Calculate indicated power, frictional power, air-fuel ratio, indicated η_{th} , brake η_{th}

1) Variables: 4 stroke SI
 $P_b = 442 \text{ kW}$
 $\eta_m = 0.85$
 $\dot{m}_f = 160 \text{ kg/h}$
 $\dot{m}_a = 410 \text{ kg} / \frac{1}{6} \text{ h} = 2460 \text{ kg/h}$
 $HV_{fuel} = 42000 \text{ kJ/kg}$

2) $\eta_m = \frac{P_b}{P_{ig}} \rightarrow 0.85 = \frac{442 \text{ kW}}{P_{ig}} \rightarrow P_{ig} = 520 \text{ kW}$

3) $P_{ig} = P_b + P_f \rightarrow 520 \text{ kW} = 442 \text{ kW} + P_f \rightarrow P_f = 78 \text{ kW}$

4) $AF = \frac{\dot{m}_a}{\dot{m}_f} \rightarrow AF = \frac{2460 \text{ kg/h}}{160 \text{ kg/h}} = 15.375$

5) $\eta_{th} = \frac{W_u}{Q_{in}} = \frac{P_{ig}}{\dot{m}_f \cdot Q_{HV}} = \frac{520 \text{ kW}}{(160 \text{ kg/h}) \cdot (42000 \text{ kJ/kg}) \cdot (\frac{1}{3600} \text{ h/s})} = \frac{520 \text{ kW}}{1866.67 \text{ kW}} = 0.2786 \rightarrow 27.9\%$

6) $\eta_{th, b} = \frac{P_b}{\dot{m}_f \cdot Q_{HV}} = \frac{442 \text{ kW}}{(160 \text{ kg/h}) \cdot (42000 \text{ kJ/kg}) \cdot (\frac{1}{3600} \text{ h/s})} = \frac{442 \text{ kW}}{1866.67 \text{ kW}} = 0.2368 \rightarrow 23.7\%$

Ex 2: Twin cylinder two stroke engine has a swept volume of 150 cm^3 . Max. power output is 16 kW at 11000 rpm. At this condition, $b_{std} = 0.11 \text{ kg/m}^3$, gravimetric air-fuel ratio is 12:1. If the ambient (intake) conditions were 10°C and 1.03 bar, and the fuel has a calorific value of 44 MJ/kg, calculate b_{mix} , $\eta_{th, b}$, volumetric efficiency (η_v).

1) 2 stroke $\Delta V = 150 \text{ cm}^3 = 15 \cdot 10^{-5} \text{ m}^3$
 $P_b = 16 \text{ kW}$ & $N = 11000 \text{ rpm}$
 $\Delta b_{std} = 0.11 \text{ kg/m}^3$
 $\Delta AF = 12:1$ & $T_{std} = 10^\circ\text{C}$ & $P_{amb} = 1.03 \text{ bar}$ & $HV_{fuel} = 44 \text{ MJ/kg}$

2) $b_{mix} = \frac{P_b \cdot \eta_r}{V_d \cdot N} \rightarrow \frac{16 \text{ kW} \cdot (\frac{1000 \text{ W}}{1 \text{ kW}})}{15 \cdot 10^{-5} \text{ m}^3 \cdot 11000 \text{ rpm} \cdot (\frac{1 \text{ min}}{60 \text{ s}})} = 6309 \text{ kg/h}$
 $\times 1 \text{ Pa} = 1 \text{ N/m}^2$ & $1 \text{ bar} = 10^5 \text{ Pa}$
 $= 6309 \text{ kg/h}$

3) $\eta_{th, b} = \frac{P_b}{\dot{m}_f \cdot Q_{HV}} = \frac{P_b}{b_{std} \cdot \Delta V \cdot N \cdot Q_{HV}} = \frac{16 \text{ kW}}{0.11 \text{ kg/m}^3 \cdot 15 \cdot 10^{-5} \text{ m}^3 \cdot 11000 \text{ rpm} \cdot (\frac{1 \text{ min}}{60 \text{ s}}) \cdot 44 \text{ MJ/kg}} = 0.2066 \rightarrow 20.7\%$

4) $\eta_v = \frac{\dot{m}_a \cdot \eta_r}{\rho_a \cdot V_d \cdot N} = \frac{b_{std} \cdot \Delta V \cdot N}{\rho_a \cdot V_d \cdot N} \rightarrow 0.11 \text{ kg/m}^3 = \frac{\dot{m}_a}{196 \text{ W}} \rightarrow \dot{m}_a = 0.00209 \text{ kg/s}$

$AF = \frac{\dot{m}_a}{\dot{m}_f} \rightarrow 12 = \frac{\dot{m}_a}{0.00209} \rightarrow \dot{m}_a = 0.02508 \text{ kg/s}$
 $\rho_a = \frac{P_{amb}}{R T_{amb}} \rightarrow \rho_a = \frac{1.03 \text{ bar} \cdot (\frac{10^5 \text{ N/m}^2}{1 \text{ bar}})}{(287 \text{ J/kg} \cdot \text{K}) \cdot (10 + 273.15 \text{ K})} = 1.267 \text{ kg/m}^3$

$\eta_v = \frac{0.02508 \text{ kg/s}}{(1.267 \text{ kg/m}^3) \cdot (15 \cdot 10^{-5} \text{ m}^3) \cdot (11000 \text{ rpm} \cdot (\frac{1 \text{ min}}{60 \text{ s}}))} = 0.7105 = 71\%$

MECHANICAL EFFICIENCY

$$P_{ig} = P_b + P_f$$

↓ ↓ ↓
indicated brake friction

$$\eta_m = \frac{P_b}{P_{ig}} = \frac{P_{ig} - P_f}{P_{ig}}$$

Typical values:
0.85 at 1800 to 2100 rpm
Piston max. rod speed,

FUEL CONVERSION EFFICIENCY

Relates power output to rate of energy input

$$\eta_f = \frac{W_c}{\dot{m}_f Q_{HV}} = \frac{P}{\dot{m}_f \cdot Q_{HV}} \rightarrow \text{fuel heating value}$$

$$\eta_f = \frac{3600}{540 (\text{g/kWh}) \cdot Q_{HV} (\text{MJ/kg})}$$

THERMODYNAMIC EFFICIENCY

$$\eta_t = \frac{W_c}{Q_{in}} = \frac{W_c}{\dot{m}_f Q_{HV} \eta_c} = \frac{P}{\dot{m}_f Q_{HV} \eta_c}$$

$$\eta_t = \frac{\eta_f}{\eta_c}$$

AIR-TO-FUEL RATIOS

$$AF = \frac{\dot{m}_a}{\dot{m}_f}$$

$$FA = \frac{\dot{m}_f}{\dot{m}_a}$$

flow rate flow rate
air mass flow rate

mixture that contains
optimum proportion

Equivalence Ratio (ϕ)

$$= \frac{\text{Actual F/A ratio}}{\text{Stoichiometric F/A ratio}}$$

$\phi = 1 \rightarrow$ Chemically correct
 $\phi < 1 \rightarrow$ lean mixture
 $\phi > 1 \rightarrow$ rich mixture

VOLUMETRIC EFFICIENCY

Measure of losses in the intake system

Can be greater than 100% for turbocharged engines

Normal between 80% to 90%

only for 4-stroke cycles

$$\eta_v = \frac{\text{mass inducted}}{\text{mass at inlet Temp}} = \frac{\rho_a \cdot \dot{V}_a}{\rho_{a,i} \cdot \dot{V}_d N}$$

$$\eta_v = \frac{\dot{m}_a}{\rho_{a,i} \cdot \dot{V}_d} \rightarrow \text{gr mass in cylinder} (\dot{m}_a = AF \cdot \dot{m}_f)$$

SPECIFIC WEIGHT, VOLUME, POWER

S. Weight

$$\frac{\text{Engine weight}}{\text{Engine Power}} = \frac{W}{bP}$$

Specific Volume

$$\frac{\text{Engine Volume}}{\text{Engine Power}} = \frac{V_d}{bP}$$

Specific Power

$$\frac{\text{Engine Power}}{\text{Piston face area (sq ft)}} = \frac{bP}{A_p}$$

Ex 3

Polls-Lager CV12 turbocharged four stroke direct injection diesel engine has a displacement of 26.1 L. Engine has a maximum output at 900 kW at 2200 rpm and is derated to 392.5 kW at 1800 rpm for industrial use. What is the bmp for each of these types?

1) 4-stroke $P_{b,max} = 900 \text{ kW}$ at $N = 2200 \text{ rpm}$

$V_d = 26.1 \text{ L} = 26.1 \cdot 10^{-3} \text{ m}^3$ $P_{b,ind} = 392.5 \text{ kW}$ at $N = 1800 \text{ rpm}$ 4-stroke

$$2) \text{ bmp}_{max} = \frac{P_{b,max} \cdot \eta_R}{V_d \cdot N} \rightarrow \frac{900 \text{ kW} \left(\frac{1000 \text{ W}}{1 \text{ kW}} \right) \cdot 2}{(26.1 \cdot 10^{-3} \text{ m}^3) \cdot (2200 \text{ rpm} \cdot \frac{1 \text{ min}}{60 \text{ s}})} = 172910 \text{ Pa} = 1.73 \text{ bar}$$

$1 \text{ Pa} = 1 \text{ N/m}^2$ $1 \text{ bar} = 10^5 \text{ Pa}$

$$3) \text{ bmp}_{ind} = \frac{P_{b,ind} \cdot \eta_R}{V_d \cdot N} \rightarrow \frac{392.5 \text{ kW} \left(\frac{1000 \text{ W}}{1 \text{ kW}} \right) \cdot 2}{(26.1 \cdot 10^{-3} \text{ m}^3) \cdot (1800 \text{ rpm} \cdot \frac{1 \text{ min}}{60 \text{ s}})} = 1015325 = 10.15 \text{ bar}$$

Ex 4

Ad 100 has a five-litre, V6, SI engine operating at 2400 rpm. Engine has a compression ratio of 10.2:1 and $\eta_v = 0.91$. Bore and stroke are related as $L = 0.92B$. Calculate stroke length, average piston speed, clearance volume of one cylinder, and flow rate into the engine.

1) 6 cylinder $V_d = 5 \text{ L} = 5 \cdot 10^{-3} \text{ m}^3$ $N = 2400 \text{ rpm}$ $r_c = 10.2:1$ $\eta_v = 0.91$ $L = 0.92B$

$$2) V_d = N_c \cdot \frac{\pi B^2}{4} \cdot L \rightarrow 5 \cdot 10^{-3} \text{ m}^3 = 6 \cdot \frac{\pi B^2}{4} \cdot 0.92B \rightarrow B = 0.105 \text{ m} = 10.5 \text{ cm}$$

6 cylinder 5 litres

$$3) \bar{S}_p = 2 \cdot L \cdot N \rightarrow L = 0.92B \rightarrow 0.92 \cdot 0.105 \text{ m} = 0.0966 \text{ m}$$

average piston speed $\bar{S}_p = 2 \cdot (0.0966 \text{ m}) \cdot (2400 \text{ rpm} \cdot \frac{1 \text{ min}}{60 \text{ s}}) = 7.728 \text{ m/s}$

$$4) r_c = \frac{\text{max volume}}{\text{min volume}} = \frac{V_{d,c} + V_c}{V_c} \rightarrow 10.2 = \frac{(5 \cdot 10^{-3} \text{ m}^3) + V_c}{V_c} \rightarrow V_c = 5.43 \cdot 10^{-4} \text{ m}^3 \rightarrow 6 \text{ cylinder if}$$

compression ratio $\frac{5.43 \cdot 10^{-4}}{6} = 9.05 \cdot 10^{-5} \text{ m}^3$

$$5) \eta_v = \frac{\dot{m}_a \cdot \eta_c}{\rho_a \cdot V_d \cdot N} \rightarrow 0.91 = \frac{\dot{m}_a \cdot 2}{(1.2 \text{ kg/m}^3) \cdot (0.005 \text{ m}^3) \cdot (2400 \cdot \frac{1 \text{ min}}{60 \text{ s}})} \rightarrow \dot{m}_a = 0.1092 \text{ kg/s}$$

Ex 5

6 cylinder two-stroke engine produces a torque of 1100 Nm at a speed of 2100 rpm. It has a bore of 123 mm and a stroke of 127 mm. What is the bmp and mean piston speed?

1) Two-stroke, $N = 2100 \text{ rpm}$ $B = 123 \text{ mm} = 0.123 \text{ m}$ $L = 127 \text{ mm} = 0.127 \text{ m}$ $T = 1100 \text{ Nm}$

$$2) V_d = \frac{\pi B^2}{4} \cdot L \rightarrow V_d = \frac{\pi \cdot (0.123)^2}{4} \cdot (0.127) = 0.00305 \text{ m}^3 \rightarrow$$

$$\text{bmp} = \frac{2 \pi T \cdot \eta_R}{V_d} \rightarrow \frac{2 \cdot \pi \cdot 1100 \text{ Nm} \cdot 1}{0.00305 \text{ m}^3} = 763264 \text{ N/m}^2 = 7.63 \text{ bar}$$

2-stroke

$$3) \bar{S}_p = 2 \cdot L \cdot N \rightarrow 2 \cdot (0.127 \text{ m}) \cdot (2100 \text{ rpm} \cdot \frac{1 \text{ min}}{60 \text{ s}}) = 8.87 \text{ m/s}$$

~ THERMODYNAMIC CYCLES ~

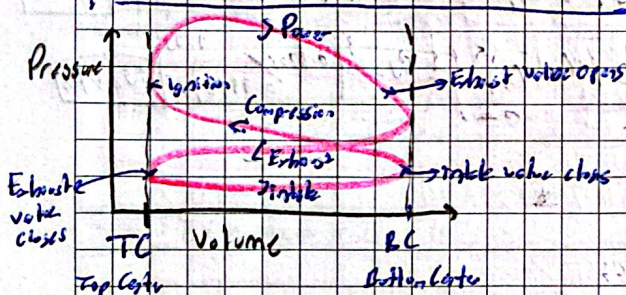
✓ Air-standard Analysis is used to perform elementary analysis of Internal Combustion (IC) engine cycles.

- ✓ Simplifications to the real cycle:
- 1) Fixed amount of air (ideal gas) for working fluid.
 - 2) Combustion process is not considered.
 - 3) Intake and exhaust process not considered.
 - 4) Engine friction and heat losses not considered.
 - 5) Specific heat is independent of temperature.

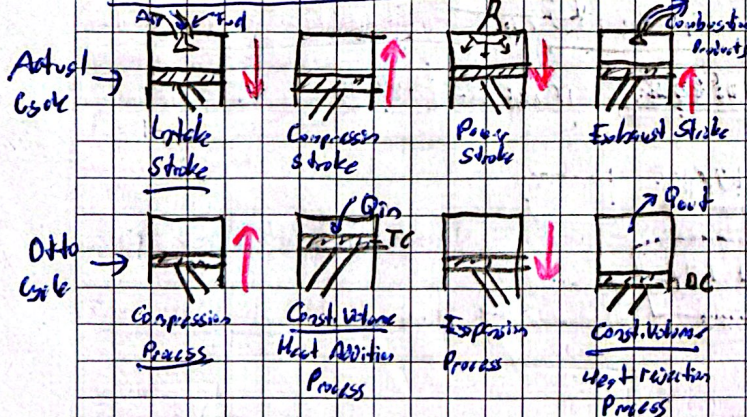
✓ Two types of reciprocating (piston/crank type) engine cycles analyzed are:

- 1) Spark Ignition - Otto Cycle
- 2) Compression Ignition - Diesel Cycle

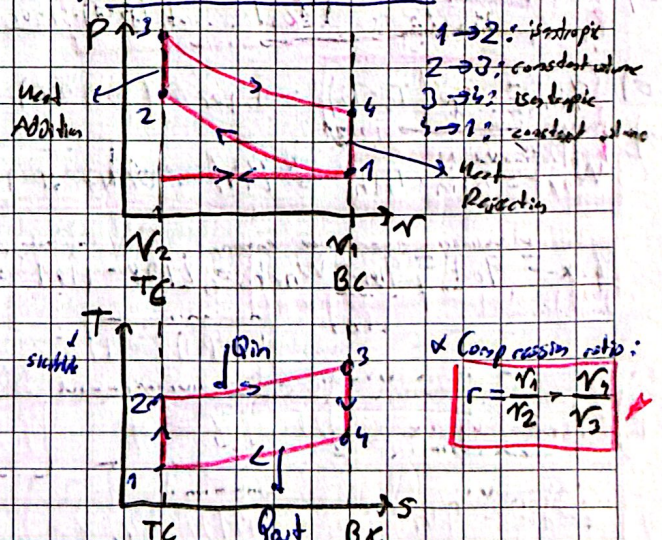
Actual Spark-Ignition (SI) ENGINE CYCLE



Actual SI vs Thermodynamic Otto Cycle



Air-Standard Otto Cycle



Ex 1 A CI engine operating on the air-standard Diesel cycle has cylinder conditions at the start of compression of 65°C and 130 kPa . Light diesel fuel is used at equivalence ratio $= 0.8$ with a combustion efficiency $\eta_c = 0.98$. Compression ratio is $r_c = 19$. Calculate a) T at end of cycle ($^\circ\text{C}$) b) P at end of cycle (kPa) c) Cut-off ratio d) $\eta_{th, ind} = ?$ (%) e) Heat loss in exhaust (kJ/kg)

a-b) $T_1 = 65^\circ\text{C} = 338\text{ K}$

1) $P_1 = 130\text{ kPa}$

2) $T_2 = T_1 (r_c)^{k-1} = 338 \cdot 19^{0.35} = 947\text{ K} = 674^\circ\text{C}$

$P_2 = P_1 (r_c)^k = 130 \cdot 19^{1.35} = 6922\text{ kPa}$

3) $Q_{Hv} \cdot \eta_c = (AF+1) \cdot c_p \cdot (T_3 - T_2)$, $AF = (AF)_{stoich} / \phi = \frac{14.5}{0.8} = 18.125$

$42500 \cdot (0.98) = (18.125 + 1) \cdot (1.108) (T_3 - 947) \rightarrow T_3 = 2913^\circ\text{C}$

$P_3 = P_2 = 6922\text{ kPa}$

4) $V_4 = V_1 = RT_1/P_1 = \frac{(0.287) \cdot 338}{130} = 0.7462$, $T_4 = T_3 (V_3/V_4)^{k-1} = (2913) \cdot (0.1208/0.7462)^{0.35} = 1247^\circ\text{C}$

$V_3 = RT_3/P_3 = \frac{0.287 \cdot 2913}{6922} = 0.1208$, $P_4 = P_3 (V_3/V_4)^k = 6922 \cdot (0.1208/0.7462)^{1.35} = 552\text{ kPa}$

c) $\beta = \frac{T_3}{T_2} = \frac{2913\text{ K}}{947\text{ K}} = 3.08$

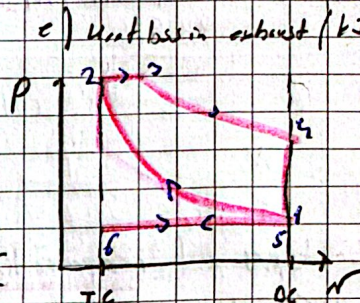
cut-off ratio

d) $\eta_{th, diesel} = 1 - \left(\frac{1}{r_c}\right)^{k-1} \cdot \frac{\beta^k - 1}{k(\beta - 1)} \rightarrow \eta_{th} = 0.547 = 54.7\%$

e) $q_{in} = c_p (T_3 - T_2) = 1.108 \cdot (2913 - 947) = 2138\text{ kJ/kg}$

$W_{net} = q_{in} \cdot \eta_{th} = 2138 \cdot 0.547 = 1191\text{ kJ/kg}$

$q_{ex} = q_{out} = q_{in} - W_{net} = 2138 - 1191 = 947\text{ kJ/kg}$



First Law Analysis of Otto Cycle

$$k = \frac{C_p}{C_v}$$

1→2: Isentropic Compression:

$$(u_2 - u_1) = \frac{Q}{m} - \left(-\frac{W_{in}}{m}\right)$$



ideal gas

$$\frac{W_{in}}{m} = (u_2 - u_1) = C_v (T_2 - T_1)$$

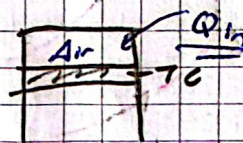
$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = r^{k-1}$$

$$\frac{P_2}{P_1} = \frac{T_2 \cdot r_1}{T_1 \cdot r_2}$$

2→3: Constant Volume Heat Addition:

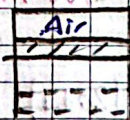
$$(u_3 - u_2) = \left(\frac{Q_{in}}{m}\right) - \frac{W}{m}$$

$$\frac{Q_{in}}{m} = (u_3 - u_2) = C_v (T_3 - T_2)$$



$$\frac{P_3}{P_2} = \frac{T_3}{T_2} \quad \text{ideal gas}$$

3→4: Isentropic Expansion:



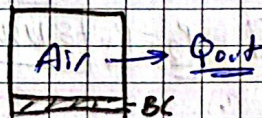
ideal gas

$$\frac{W_{out}}{m} = (u_3 - u_4) = C_v (T_3 - T_4)$$

$$\frac{P_4}{P_3} = \frac{T_4}{T_3} \cdot \frac{r_3}{r_4}$$

4→1: Constant Volume Heat Rejection:

$$\frac{Q_{out}}{m} = (u_4 - u_1) = C_v (T_4 - T_1)$$



$$\frac{P_4}{P_1} = \frac{T_4}{T_1} \quad \text{ideal gas}$$

$$\text{Net Cycle Work: } W_{cycle} = W_{out} - W_{in} = m(u_3 - u_4) - m(u_2 - u_1)$$

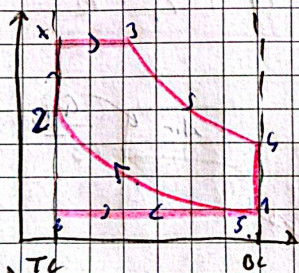
$$\text{Cycle Thermal Efficiency: } \eta_{th} = \frac{W_{cycle}}{Q_{in}} = \frac{(u_3 - u_4) - (u_2 - u_1)}{(u_3 - u_2)} = 1 - \frac{u_4 - u_1}{u_3 - u_2} = 1 - \frac{C_v (T_4 - T_1)}{C_v (T_3 - T_2)}$$

Indicated Mean Effective Pressure:

$$i_{mp} = \frac{W_{cycle}}{V_1 - V_2} \quad , \quad i_{mp} = \frac{1}{k-1} \left(\frac{Q_{in}/m}{v_1} \right) \cdot \left(\frac{r}{r-1} \right) \eta_{th}$$

Ex 12 An inline six cylinder 3.0 litre CI engine using light diesel fuel of $AF = 20$. Operates on air-standard Dual cycle. Half of the fuel can be considered burned at const. volume, and half at const. pressure with combustion efficiencies $\eta_c = 100\%$. Cylinder conditions at the start of compression are 60°C and 101kPa . $r_c = 14:1$. Calculate:

- T at each state (K)
- P at each state (kPa)
- cut-off ratio
- pressure ratio
- η_{th} (%)
- Heat added during combustion (kJ/kg)
- Net indicated work (kJ/kg)



(g/b) $T_1 = 60^\circ\text{C} = 333\text{K}$
 $P_1 = 101\text{kPa}$
 $T_2 = T_1 (r_c)^{\gamma-1} = 333\text{K} \cdot 14^{0.35} = 820\text{K}$
 $P_2 = P_1 (r_c)^\gamma = 101\text{kPa} \cdot 14^{1.4} = 3561\text{kPa}$

Half of heat in const. volume:
 $Q_{HV} \cdot \eta_c = (AF+1) \cdot c_v (T_x - T_2) = \frac{1}{2} \cdot 42500 \cdot 1 = (20+1) \cdot (0.821) (T_x - 820)$
 $T_x = 2072\text{K}$

Half of heat in const. pressure:
 $Q_{HV} \cdot \eta_c = (AF+1) \cdot c_p (T_3 - T_x) = \frac{1}{2} \cdot 42500 \cdot 1 = (20+1) \cdot (1.107) \cdot (T_3 - 2072)$
 $T_3 = 2985\text{K}$

$P_x = P_2 \cdot \frac{T_x}{T_2} = 3561 \cdot \frac{2072}{820} = 8794\text{kPa}$
 $P_3 = P_x \cdot \frac{T_3}{T_x} = 8794\text{kPa}$

$v_4 = v_1 = R T_1 / P_1 = 0.203 \cdot 333 / 101 = 0.666\text{m}^3/\text{kg}$
 $v_3 = R T_3 / P_3 = 0.287 \cdot 2985 / 8794 = 0.957\text{m}^3/\text{kg}$
 $T_4 = T_3 (v_3 / v_4)^{\gamma-1} = 1542\text{K}$
 $P_4 = P_3 (v_3 / v_4)^\gamma = 408\text{kPa}$

c) $\beta = \frac{T_3}{T_x} = \frac{2985}{2072} = 1.441$
d) $\alpha = \frac{P_3}{P_2} = \frac{8794}{3561} = 2.47$

cut-off ratio

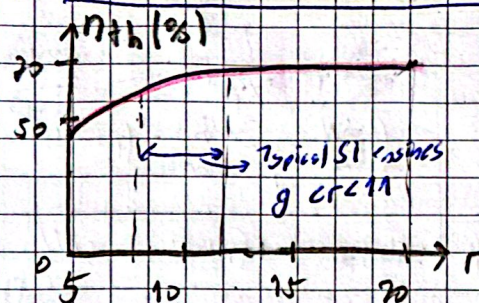
pressure ratio

e) $\eta_{th, dual} = 1 - \left(\frac{1}{r_c} \right)^{\gamma-1} \cdot \frac{\alpha \beta^\gamma - 1}{(k \cdot \alpha (\beta - 1) + \alpha - 1)} = 0.589 = 58.9\%$

f) $q_{in} = c_v (T_x - T_2) + c_p (T_3 - T_x) \rightarrow q_{in} = 2064\text{kJ/kg}$

g) $w_{net} = \eta_{th} \cdot q_{in} = 1192\text{kJ/kg}$

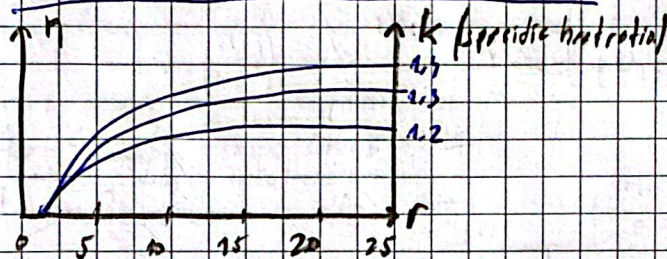
COMPRESSION RATIO - THERMAL EFFICIENCY



$$\eta_{th} = 1 - \frac{1}{r^{k-1}}$$

Constant C_v

SPECIFIC HEAT RATIO - THERMAL EFFICIENCY

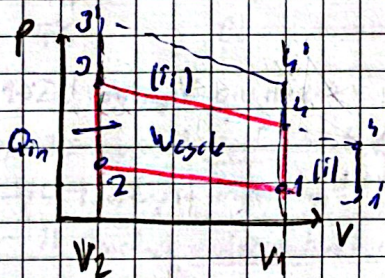


$$1.2 < k < 1.4$$

$$k \approx 1.3$$

FACTORS EFFECTING WORK PER CYCLE

- Net cycle work can be increased by
- increasing r (1-2) (i)
 - increasing Q_{in} (2-3) (ii)



$$imep = \frac{W_{cycle}}{V_1 - V_2} = \frac{Q_{in}}{V_1} \left(\frac{r}{r-1} \right) \eta_{th}$$

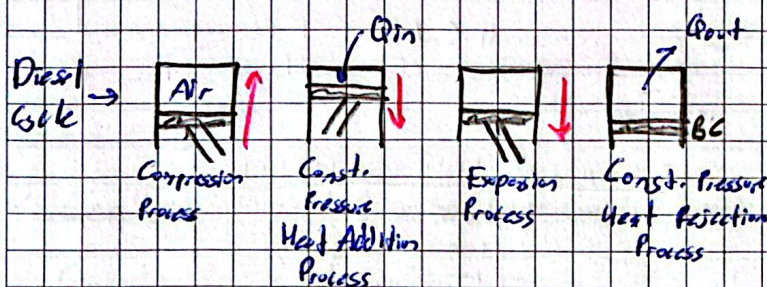
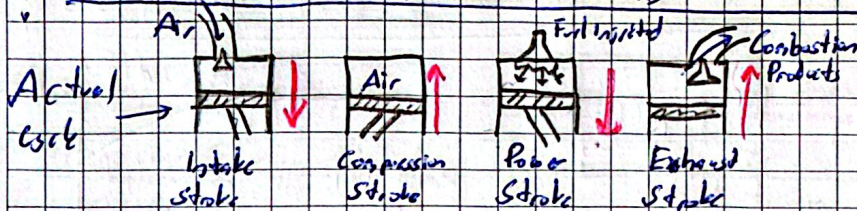
COMPRESSION RATIO - THERMAL EFFICIENCY - IMEP



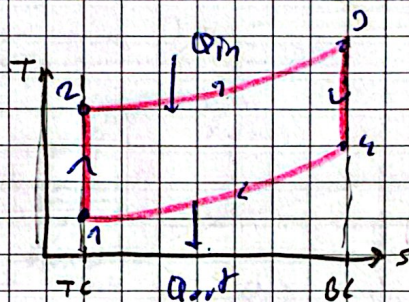
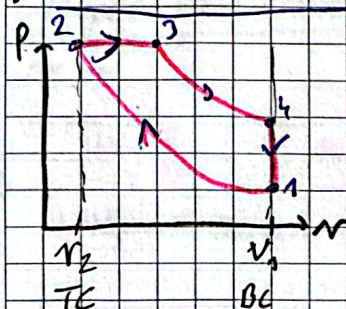
$$imep = \frac{Q_{in}}{V_1} \left(\frac{r}{r-1} \right) \left(1 - \frac{1}{r^k} \right)$$

(EARLY)

COMPRESSION IGNITION (CI) ENGINE CYCLE VS THERMODYNAMIC DIESEL ENGINE



Air-Standard DIESEL CYCLE



Cut-off ratio:

$$r_c = \frac{v_3}{v_2}$$

1-2: Isentropic compression

2-3: Constant pressure heat addition

3-4: Isentropic expansion

4-1: Constant volume heat rejection

FIRST LAW ANALYSIS OF DIESEL CYCLE

1-2: Isentropic compression (same with Otto cycle)

2-3: Constant pressure heat addition

3-4: Isentropic expansion

$$\frac{W_{out}}{m} = (u_3 - u_4) = C_v (T_3 - T_4)$$

$$\frac{P_4}{P_3} = \frac{T_4}{T_3}, \frac{P_4}{P_1} = \frac{T_4}{T_1}$$

4-1: Constant volume heat removal (same with Otto cycle)

$$(u_3 - u_2) = \left(\frac{Q_{in}}{m} \right) - \frac{P_2 (v_3 - v_2)}{m}$$

$$\frac{Q_{in}}{m} = (h_3 - h_2) = C_p (T_3 - T_2)$$

$$r_c = \frac{T_3}{T_2} = \frac{v_3}{v_2} \leftarrow \text{ideal gas}$$

Thermal Efficiency

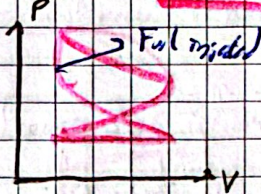
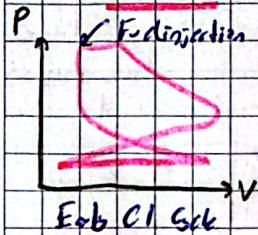
$$\eta_{th}^{diesel} = 1 - \frac{Q_{out}/m}{Q_{in}/m} = 1 - \frac{(u_4 - u_1)}{(h_3 - h_2)}$$

For the same compression ratio (r_c), $\eta_{diesel} < \eta_{otto}$
 CI needs higher r_c than SI to ignite

Early vs Modern CI Engines

α In early CI engines, fuel was injected when piston reached TC

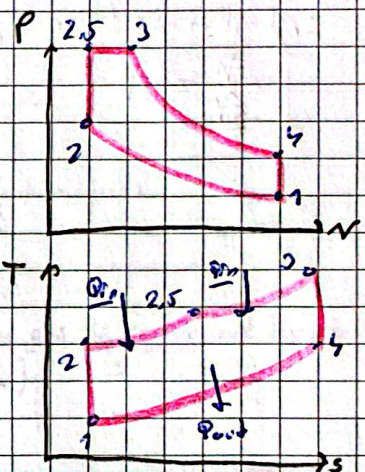
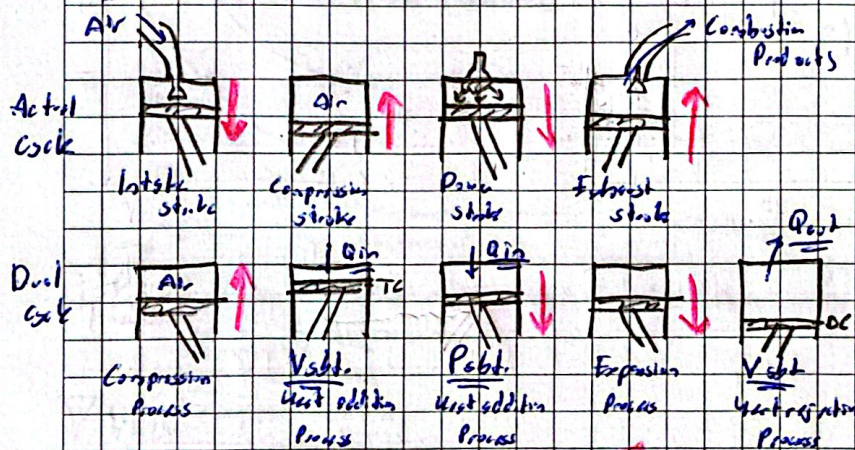
$\rightarrow$ In modern CI engines fuel is injected before TC



α [Combustion Process of early CI $\rightarrow$ Constant pressure heat addition $\rightarrow$ Diesel Cycle]

Combustion Process of modern CI $\rightarrow$ Combination of const V and P $\rightarrow$ Dual Cycle

MODERN CI ENGINE AND THERMODYNAMIC DUAL CYCLE



$$\eta_{\text{dual, const}} = 1 - \frac{1}{r^{\gamma-1}} \left[\frac{r_c^{\gamma} - 1}{(\gamma-1) + \alpha \cdot \gamma (r_c - 1)} \right]$$

$$r_c = \frac{V_3}{V_2}$$

$$\alpha = \frac{P_3}{P_2}$$

$$\frac{Q_{in}}{m} = (u_{2.5} - u_2) + (h_3 - h_{2.5})$$

$$= c_v(T_{2.5} - T_2) + c_p(T_3 - T_{2.5})$$

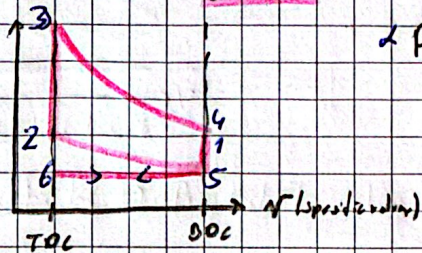
Comparison of Efficiencies

α For same inlet conditions P_1, V_1 and same compression ratio $\rightarrow \eta_{\text{otto}} > \eta_{\text{dual}} > \eta_{\text{diesel}}$

α For same inlet conditions P_1, V_1 and same peak pressure $P_3 \rightarrow \eta_{\text{diesel}} > \eta_{\text{dual}} > \eta_{\text{otto}}$
 (actual design limitation in engines)

INTAKE EXHAUST AND IN-CYLINDER FLOW

In the Otto and Diesel cycles, exhaust process is modeled as const. volume heat rejection, it does not consider the actual gas flow.



Process 5-6 indicates a decrease in N which is incorrect.

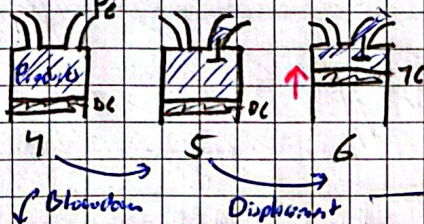
$$N = \frac{V}{m} \rightarrow \begin{matrix} \text{decrease} \\ \text{decrease} \end{matrix} \quad N \text{ actually remains the same.}$$

This inconsistency results from treating an open system problem with a closed system model.

ACTUAL EXHAUST STROKE (4-6)

Actual exhaust process consists of two phases → 1) Blowdown

2) Displacement

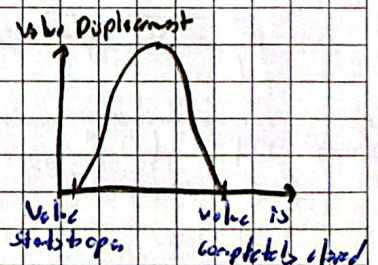


1) $P_4 > P_c$ (atm) → Gas flows out and P drops to P_c
without manifold pressure

2) Remaining gas is pushed out by the piston (Supernat)

VALVE OPENING AND CLOSING

In these cycles it is assumed the valves open and close instantaneously.
 In reality a cam is used to open and close valves.



VALVE SIZING

$$A_v \geq 1.3 b^2 \cdot \frac{U_p}{C_d} \rightarrow \begin{matrix} \text{average piston velocity} \\ \text{spread of speed of gas in intake port} \end{matrix}$$

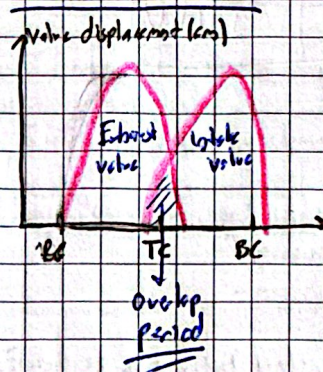
average valve area Cylinder bore

Since there is only so much room available for valves, it is common to have multiple valves per cylinder.

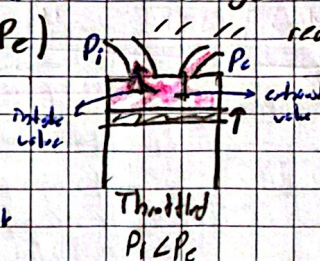
This increases valve area to piston ratio → permits higher engine speeds

Heads are often valve-shaped or domed → this permits $\frac{A_v}{A_p}$ up to 0.5.

VALUE OVERLAP



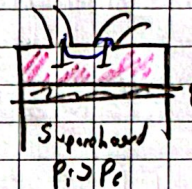
Part Throttle: Residual gas flows into the intake port.
($P_i < P_e$) reduces performance



wide open throttle

WOT: Some fresh gas can flow out the exhaust valve, reducing fuel efficiency and increasing emissions.
($P_i = P_e$)

Supercharge: Fresh gas can flow out the exhaust valve.
($P_i > P_e$)



PRESSURE LOSSES IN INTAKE SYSTEM

ΔP_{air} = Pressure losses in air cleaner

ΔP_u = Intake losses upstream of throttle

ΔP_{thr} = Loss across throttle

ΔP_{valv} = Loss across intake valve

FACTORS AFFECTING $\eta_v \rightarrow$

$$\eta_v = \frac{P_a \cdot P_r}{P_i \cdot V_i \cdot N}$$

1) Closing the intake valve after BC (backflow)

Best time to close the intake valve is when $P_{inlet} = P_{atmos}$.

High engine speed $\rightarrow$ higher flow velocities $\rightarrow$ large pressure drop $\rightarrow$ close valve later after BC.

Low engine speed $\rightarrow$ small pressure drop $\rightarrow$ close valve earlier after BC.

2) RAM effect

Inertia of the air in intake system increases the pressure in the intake port, called ram effect.

Progressively more important at higher engine speeds.

To take advantage of ram effect, close intake valve after BC.

3) Intake and exhaust tuning

IN-CYLINDER FLUID FLOW

Three parameters to characterize in-cylinder fluid motion i) swirl ii) squish iii) tumble

Swirl is the rotational flow about cylinder axis

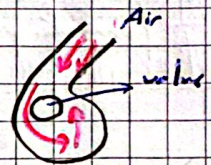


Swirl is used to:

- promote rapid combustion in SI engines
- rapidly mix fuel and air in CI engines and gasoline direct injection engines

Swirl is generated by:

- tangentially directing the flow into cylinder
- using helical ports



Swirl theory

Simply modeled as gas rotating at angular velocity ω

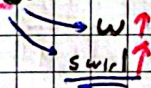
Tangential flow velocity = $u = \omega \cdot r$

Swirl ratio $R_s = \frac{\omega}{2\pi N}$ engine speed (rps)

Engine swirl

Many engines have a bowl in the piston

During compression, as piston reaches TDC more of the air enters cavity



Or a cylinder head cavity



Squish and Tumble

Squish is radial flow at the end of compression stroke, compressed gases flow into cavity.



Squish

As piston reaches TDC, squish motion creates a secondary flow called tumble where rotation occurs.



Tumble

INTAKE FLOW



shear layers

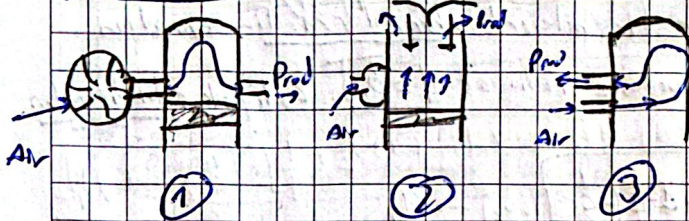
vortices

become unstable and eventually break into turbulent motion

Energy is required to generate turbulence, if no energy is supplied, turbulence decays.

A common source of energy is shear layers in the mean flow.

2-Stroke Engine In-Cylinder Flow



(Squeeze)

SCAVENGING PERFORMANCE

× If cylinder volume is completely filled with air:

$$D_r = \frac{\text{mass of delivered air per cycle}}{\text{displaced volume} \times \text{ambient density}}$$

Drifts ratio

$$D_r = \frac{r}{r-1} > 1$$

$$\Gamma = \frac{\text{mass of delivered air retained}}{\text{mass of delivered air}}$$

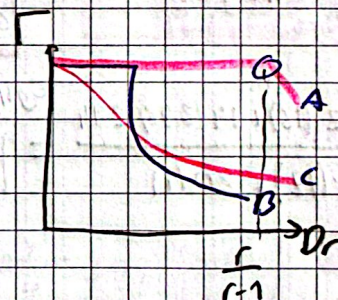
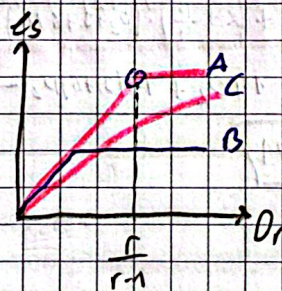
Trapping efficiency

$$e_s = \frac{\text{mass of delivered air retained}}{\text{mass of trapped cylinder charge}}$$

Scavenging efficiency

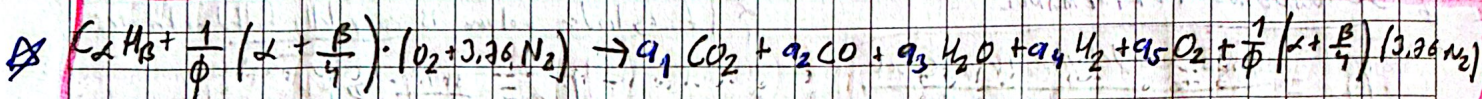
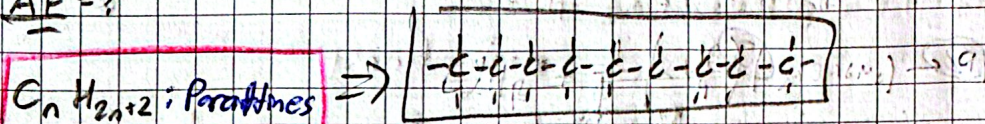
SCAVENGING MODELS

- Perfect Scavenging (Nominal)
- Short Circuiting
- Perfect Mixing

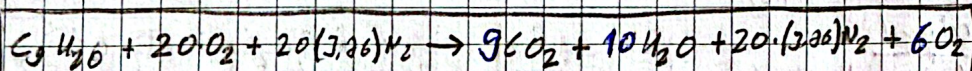
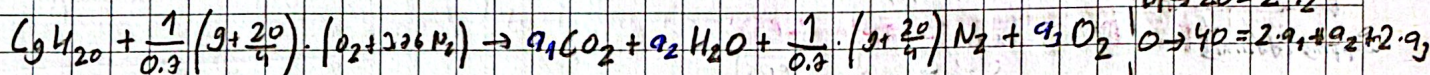
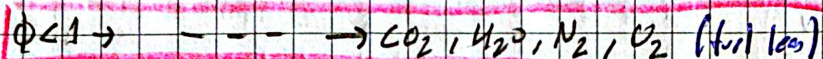


Ex 1 Draw the chemical structural formula of 2-methyl-2,3-dimethylbutane (C_9H_{20}).

This is an isomer of which chemical family. Write the balanced chemical reaction equation for one mole of this fuel burning with an equivalence ratio of $\phi = 0.8$. Calculate stoichiometric $AF = ?$



General form for $\phi < 1, \phi = 1, \phi > 1$

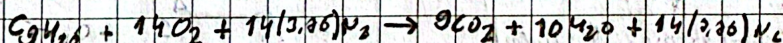


$$\begin{aligned} C &\rightarrow 9 = a_1 \\ H &\rightarrow 20 = 2 \cdot a_2 \\ O &\rightarrow 40 = 2 \cdot a_1 + a_2 + 2 \cdot a_3 \\ &40 \end{aligned}$$

$AF = \frac{m_a}{m_f} = \frac{n_a \cdot M_a}{n_f \cdot M_f}$

number of moles $\rightarrow$ mole weight

Stoichiometric Equation:



$(AF)_s = \frac{14 \cdot 2 \cdot (16) + 14 \cdot 3.76 \cdot 2 \cdot 14}{9 \cdot (12) + 20 \cdot 14} = 15.01$

COMBUSTION THEORY

Composition of Standard Dry Air

Air is a mixture of gases O_2, N_2, Ar, CO_2, H_2O

For combustion, dry air is taken $\rightarrow 21\% O_2, 79\% N_2$.

$$\frac{n_{N_2}}{n_{O_2}} = \frac{0.79}{0.21} = 3.76 \rightarrow \text{For every mole of } O_2, \text{ there are 3.76 moles of } N_2.$$

Composition of Standard Moist Air

Atmospheric air always contains some water vapor.

$$w = \frac{m_{H_2O}}{m_{Air}}$$

specific humidity

$$\phi = \frac{P_{H_2O}}{P_{sat}(T)}$$

relative humidity

$$0 < \phi < 1$$

partial water pressure

equilibrium vapor pressure of water at specific T

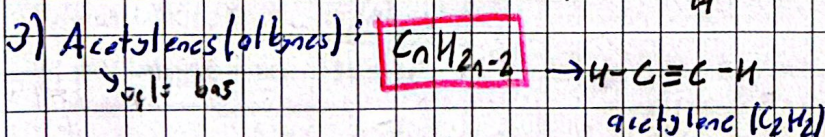
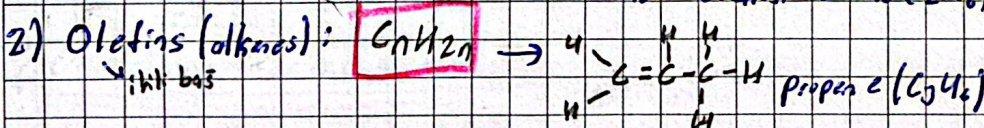
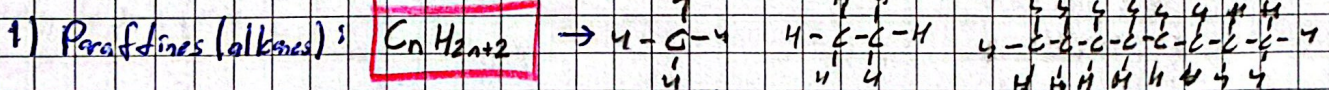
$$w = \frac{m_{H_2O}}{m_{Air}} = \frac{\bar{M}_{H_2O} \cdot n_{H_2O}}{\bar{M}_{Air} \cdot n_{Air}} = \frac{18 \cdot P_{H_2O}}{29 \cdot P_{Air}} = 0.62 \cdot \frac{P_{H_2O}}{P - P_{H_2O}} = 0.62 \cdot \left(\frac{\phi \cdot P_{sat}(T)}{P - \phi \cdot P_{sat}(T)} \right)$$

$$\frac{n_{H_2O}}{n_{Air}} = 1.61 \cdot w$$

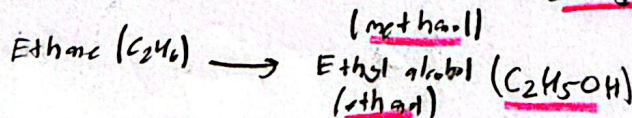
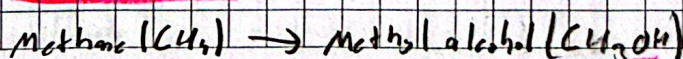
number of moles

Hydrocarbon Fuels

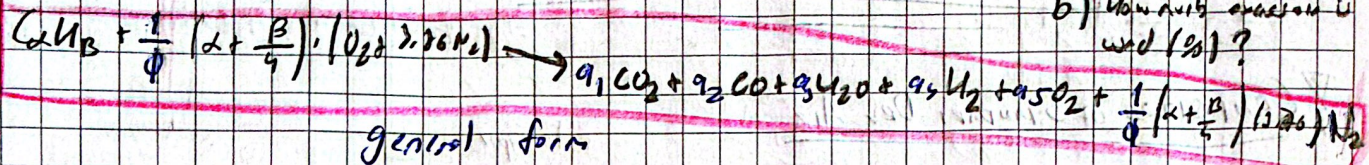
Most common hydrocarbon fuels are Alkyl compounds and are grouped as:



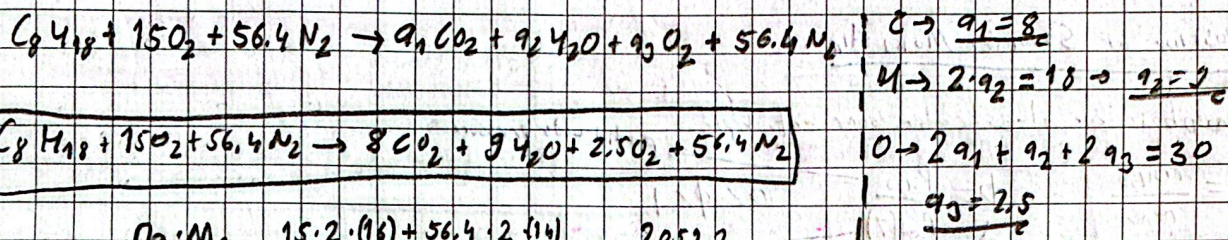
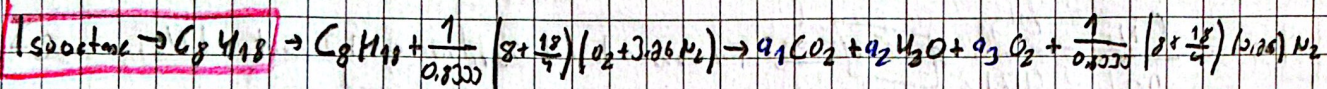
Note: For alcohols one hydroxyl (OH) is replaced with one H



Ex: Isooctane is burned with air in an engine at an equivalence ratio $\phi = 0.8033$. Assuming complete combustion, write balanced chemical reaction equation. Calculate: a) A/F?



$\phi = 0.8033 < 1 \therefore$ Rich $\rightarrow$ Products $\rightarrow$ CO_2, H_2O, N_2, O_2



$$\begin{aligned} C &\rightarrow a_1 = 8 \\ H &\rightarrow 2a_2 = 18 \rightarrow a_2 = 9 \\ O &\rightarrow 2a_1 + a_2 + 2a_3 = 30 \\ &\quad a_3 = 2.5 \end{aligned}$$

a) $A/F = \frac{n_a \cdot M_a}{n_f \cdot M_f} = \frac{15 \cdot 2 \cdot (16) + 56.4 \cdot 2 \cdot (14)}{8 \cdot (12) + 18 \cdot (1)} = \frac{2052.2}{114} = \boxed{A/F = 18.063}$

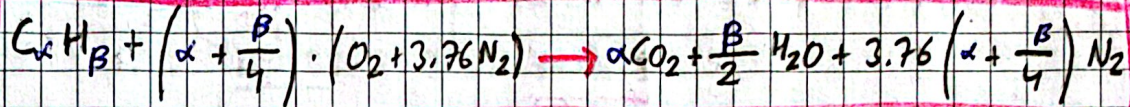
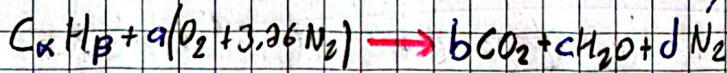
b) $\gamma = \frac{1}{\phi} = 1.2 \rightarrow \boxed{\text{Excess air is } \%20}$

H
(1.555).1

213.54662
14,22

Combustion Stoichiometry

N_2 is not oxidized $\rightarrow$ inert



stoichiometric proportions of fuel and air

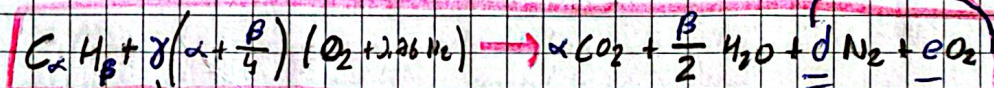
α Stoichiometric mass based air/fuel ratio for $C_x H_y$ fuel:

$$(A/F)_s = \frac{1}{(F/A)_s} = \frac{\left(1 + \frac{(y/x)}{4}\right) \cdot (32 + 3.76 \cdot 28)}{12 + (y/x) \cdot 1}$$

Expression only depends on y/x

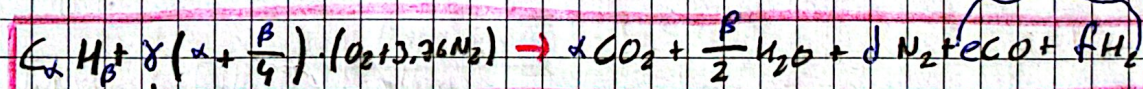
Expression only applies to stoichiometric mixture

Fuel Lean Mixture $\rightarrow$ Excess air



$\gamma > 1 \rightarrow$ excess air

Fuel Rich Mixture $\rightarrow$ Less than stoichiometric air



$\gamma < 1 \rightarrow$ insufficient air

Three unknowns (d, e, f), two balance equations (O, N)

Off-Stoichiometric Mixtures

$$\phi = \frac{(A/F)_s}{(A/F)_{mixture}} = \frac{(F/A)_{mixture}}{(F/A)_s}$$

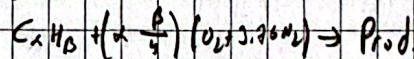
equivalence ratio

$\phi = 1 \rightarrow$ stoichiometric

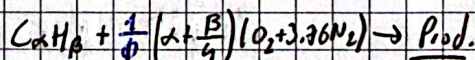
$\phi < 1 \rightarrow$ fuel lean

$\phi > 1 \rightarrow$ fuel rich

stoichiometric mixture:



off-stoichiometric mixture:

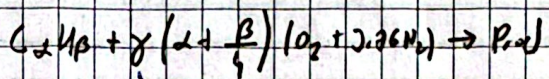


Can not solve without additional info

7 OFF-STOICHIOMETRIC CONDITIONS

• Another terminology to describe air in combustion:

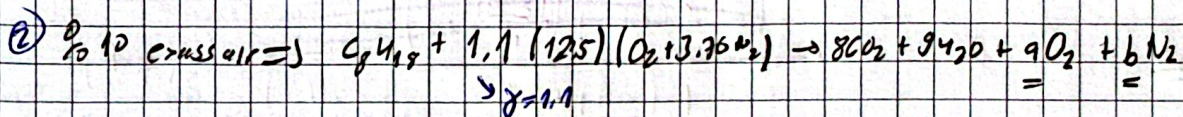
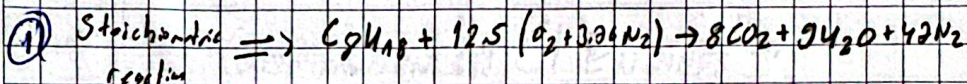
150% stoichiometric air = 150% theoretical air = 50% excess air



→ $x = 1.5 \rightarrow 9250$ cross air

F-100

Consider a reaction of octene with 10% excess air, what is ϕ ?



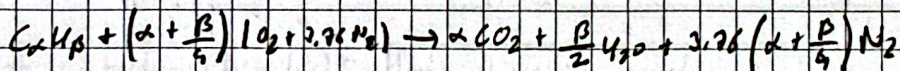
$$0 \rightarrow 16 + 9 + 2a = 1.1 (12.5) \cdot (2) \rightarrow \underline{a = 1.25}$$

$$N \rightarrow b = 1,1 / 12,5) \cdot (0,16) \rightarrow \underline{b = 51,2}$$

$$\textcircled{3} \phi = \frac{(A/F)_s}{(A/F)_{\text{critical}}} = \frac{12.5(4.26)/1}{1.1(12.5)(4.26)/1} = 0.91$$

YEAR OF COMBINATION

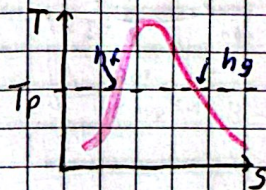
2 Max amount of energy is released from a fuel when reacted with a stoichiometric amount of air and all the C and H is converted to CO_2 and H_2O



Max energy is referred as heat of combustion or heating value and given as per mass of fuel.

<u>Fuel</u>	<u>$\Delta H_R (298K) (MJ/t)$</u>	<u>H_2</u>	<u>ΔH_R</u>
$C_2H_2(g)$ C_{2H_2}	21.0	Hydrocarbons	↓
$H_2(g)$ $H_{2, gas}$	141.6	Alcohols	

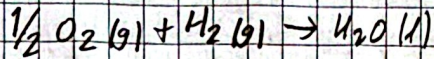
α There are two possible values for heat of combustion depends on whether the water in products is taken as saturated liquid or vapor.



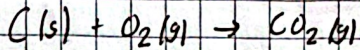
Higher heat of combustion: Product water is taken as liquid ($h_{H_2O} = h_f$)

Lower heat of combustion: " " " vapor ($h_{H_2O} = h_g$)

Heat of Formation



$$\bar{Q} = -286,000 \text{ kJ/kmol H}_2\text{O}$$



$$\bar{Q} = -394,000 \text{ kJ/kmol CO}_2$$

Formation Reactions

At standard temperature and pressure (STP)

101 kPa, 273 K

Standard Heat of Formation:

$$\rightarrow \bar{h}_{f, \text{H}_2\text{O}} = -286,000 \text{ kJ/kmol}$$

$$\rightarrow \bar{h}_{f, \text{CO}_2} = -394,000 \text{ kJ/kmol}$$

COMBUSTION IN CI ENGINES

COMBUSTION PHASES STAGES

Ignition Delay (a): Fuel is injected towards the end of compression stroke.

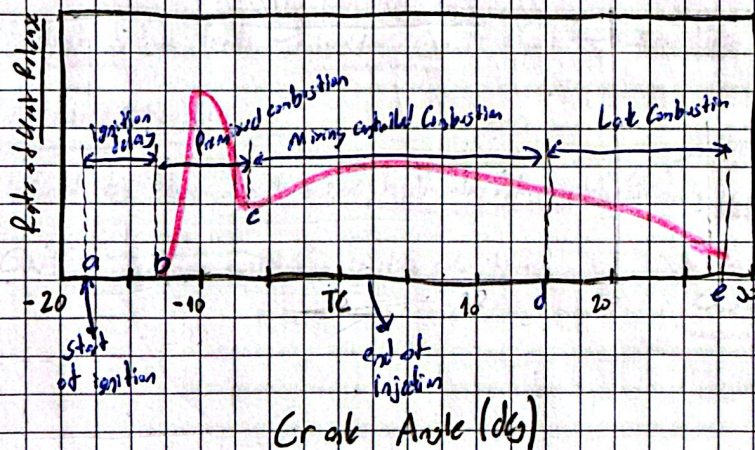
↳ Liquid fuel atomizes into small drops and penetrates into combustion chamber

↳ Fuel vaporizes and mixes with high temp, high pressure air

Premixed Combustion Phase (b): Combustion of the fuel-air mixture (premixed)

Mixing Controlled Combustion Phase (c): Burning rate is controlled primarily by the fuel-air mixing process

Late Combustion Phase (d): Heat release may proceed at lower rate into the expansion stroke. Although no additional fuel is injected, combustion of unburned fuels are responsible for this.



1) CI Engine Types

1) Direct Injection: Single open combustion chamber into which fuel is injected directly.
 * For very large engines with low speeds. → Time available for mixing is long.

2) Indirect Injection: A chamber is divided into two regions.



* Single high speed engines used in automobiles.

* High swirl or turbulence is generated in prechamber.

2) Ignition Delay

* Ignition delay is defined as the time or crank angle interval from the start of fuel injection to the onset (beginning) of combustion.

* Both physical and chemical processes must take place before a significant fraction of the fuel chemical energy is released.

Physical processes: Fuel atomization, evaporation, mixing of fuel vapor with exhauster air.

- Good atomization → High fuel pressure, Optimum fuel viscosity, Small injector diameter, High exhaust pressure.

- Rate of evaporation → Droplet diameter, Fuel volatility, Velocity, Partial P of air.

3) Fuel Ignition Quality

* Ignition quality of a fuel is defined by its cetane number (CN).

* Low cetane fuels → ignition delay is long → under extreme cases produces a knocking sound "direct knock".

* High cetane fuels → ignition delay is short → smoother engine operation.

4) CETANE NUMBER

* Defined by blends of two pure hydrocarbon reference fuels.
 → HMN → CN=15
 → Hexadecane → CN=100

* CN ↑ ignition quality ↑ ignition delay time ↓

$$CN = \left(\frac{\% \text{hexadecane}}{100} \right) + 0.15 \left(\frac{\% \text{HMN}}{100} \right)$$

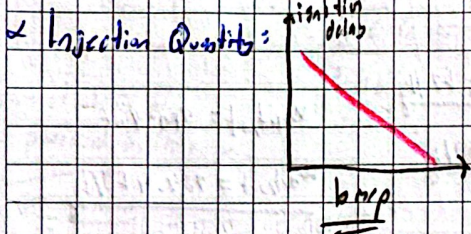
CELANE NUMBER MEASUREMENT

- Method uses a standard single cylinder engine with standard operating conditions.
- With the engine running at these conditions with test fuel, compression ratio is varied until combustion starts at 70° → ignition delay period of 15°.
- Same procedure is repeated using blends of HMMN and hexadecane. The blend that gives 15° ignition delay with the same compression ratio is used to calculate the test fuel CN.

Note: Octane Number and Cetane Number are correlated.

FACTOR AFFECTING IGNITION DELAY TIME

- Injection Timing: At normal engine conditions, min. delay is occurred with the start of injection at about 10-15 TC.
- Earlier or later injection timing results in increase in ignition delay time.



- Intake Air Temperature and Pressure:
Increase in either will result in decrease in ignition delay.
- $T \uparrow$ Ignition Delay $\downarrow$, $P \uparrow$ Ignition Delay $\downarrow$

~ DIESEL SOLUAK ~

Ex:

An ideal diesel cycle has a max. cycle temperature 2000°C . State at the start of the beginning of the compression is $P_1 = 95 \text{ kPa}$, $T_1 = 15^{\circ}\text{C}$. Cycle is executed in a four-stroke, 8-cylinder engine with cylinder bore of 10 cm and a piston stroke of 12 cm . Minimum volume ratio is 5 percent of max. cylinder volume. Determine the power produced by this engine when it is operated at 1600 rpm . Use constant specific heats at room temp. ($c_p = 1.005 \text{ kJ/kg}\cdot^{\circ}\text{C}$, $k = 1.4 = \frac{c_p}{c_v}$)

$$v_1 = \frac{R T_1}{P_1} = \frac{0.287 \cdot 288 \text{ K}}{95 \text{ kPa}} = 0.8701 \text{ m}^3/\text{kg}$$

$$m = N_{\text{cyl}} \frac{\Delta V}{v_1} = 8 \cdot \frac{\pi B^2 S / 4}{v_1} = \frac{\pi (0.10)^2 \cdot 0.12 \cdot 1600}{0.8701} = 0.00865 \text{ kg}$$

total air mass taken by all 8 cylinders

$$\dot{m} = \frac{m \cdot n}{N_{\text{rev}}} = \frac{(0.00865) (1600/60 \text{ rev/s})}{2 \text{ rev/cycle}} = 0.1155 \text{ kg/s}$$

rate at which air is processed by the engine

$$r = \frac{V_{\text{max}}}{V_{\text{min}}} = \frac{1}{0.05} = 20$$

compression ratio

At the end of compression:

$$\frac{T_2}{T_1} = r^{k-1} \Rightarrow T_2 = T_1 \cdot r^{k-1} = 288 \cdot 20^{1.4-1} = 954.6 \text{ K}$$

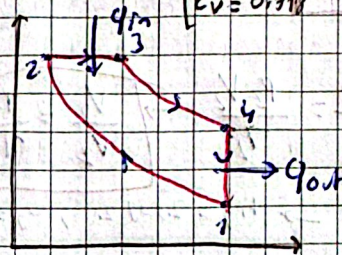
$$q_{\text{in}} = c_p (T_3 - T_2) = 1.005 (2243 - 954.6) = q_{\text{in}} = 1325 \text{ kJ/kg}$$

$$r_c = \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2243}{954.6} = 2.351$$

cut-off ratio

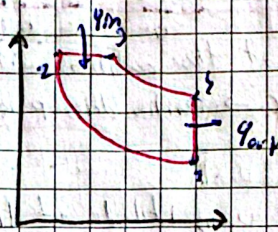
$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} \cdot \frac{r_c^k - 1}{k(r_c - 1)} = 0.6303$$

$$\dot{W}_{\text{net}} = \dot{m} \cdot \eta_{\text{th}} \cdot q_{\text{in}} = (0.1155) \cdot (0.6303) \cdot (1325) = 96.5 \text{ kW}$$



Exp 2

An ideal diesel engine has a compression ratio of 20. The state of air at the beginning of compression process is 95 kPa , 20°C . If the max temperature is not to exceed 2200°C . determine:
 $\eta_{th} = ?$
 $\text{MEP} = ?$



$$\eta_{th} = \frac{W_{net,out}}{q_{in}}$$

$$W_{net,out} = q_{in} - q_{out}$$

$$1 \rightarrow 2: T_2 = T_1 \cdot r^{(k-1)} = 293 \cdot 20^{(1.4-1)} = 931.1 \text{ K}$$

$$2 \rightarrow 3: \frac{P_3 V_3}{T_3} = \frac{P_2 V_2}{T_2} \rightarrow \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{2200}{931.1} = 2.365$$

$$2 \rightarrow 4: T_4 = T_3 \left(\frac{V_3}{V_4} \right)^{k-1} = 2200 \cdot \left(\frac{2.365 \cdot V_2}{V_4} \right)^{1.4-1} = 2200 \cdot \left(\frac{2.365}{r} \right)^{0.4} = 920.6 \text{ K}$$

$\frac{V_3}{V_2} = r \rightarrow \frac{V_3}{V_4} = \frac{r}{r} = 1$

$$q_{in} = h_3 - h_2 = c_p (T_3 - T_2) = 1.005 \cdot (2200 - 931.1) = 1235 \text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = c_v (T_4 - T_1) = 0.818 (920.6 - 293) = 504.6 \text{ kJ/kg}$$

$$\eta_{th} = \frac{W_{net,out}}{q_{in}} = \frac{784.4}{1235} = 0.635$$

$$W_{net,out} = q_{in} - q_{out}$$

$$W_{net,out} = 1235 - 504.6 = 784.4 \text{ kJ/kg}$$

$$\text{MEP} = \frac{W_{net,out}}{V_1 - V_2}$$

$$V_1 = \frac{R T_1}{P_1} = \frac{0.287 \cdot 293}{95} = 0.885 \text{ m}^3/\text{kg}$$

$$V_{min} = V_2 = \frac{V_{max}}{r} \quad \left(r = \frac{V_{max}}{V_{min}} \right)$$

$$\text{MEP} = \frac{784.4}{0.885 (1 - 1/20)}$$

$$\text{MEP} = \frac{784.4}{(0.885) \cdot (1 - 1/20)} = 970.1 \text{ kPa}$$

Exp 3 Same as Exp 2 but replace isentropic expansion process with a polytropic expansion with polytropic exponent $n=1.35$

1-2: $T_1 = 293K \rightarrow \left. \begin{array}{l} u_1 = 203.46 \\ v_1 = 659.2 \end{array} \right\} \text{Table A-12}$

$V_{r2} = \frac{v_2 \cdot v_{r1}}{v_1} = \frac{1}{v_{r1}} = \frac{1}{20} (659.2) = 32.96 \rightarrow \left. \begin{array}{l} T_2 = 512.7 \\ v_2 = 943.12 \end{array} \right\} \text{Table A-12}$

2-3:

$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \rightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200K}{512.7} = 2.410$

$T_3 = 2200K \rightarrow h_3 = 2503.2, u_3 = 1712.4 \text{ (Table A-12)}$

$q_{in} = h_3 - h_2 = 2503.2 - 943.12 = 1556$

3-4: Polytropic Expansion

$T_4 = T_3 \left(\frac{v_3}{v_4} \right)^{n-1} = T_3 \left(\frac{2.410 v_2}{v_1} \right)^{n-1} = T_3 \left(\frac{2.410}{1} \right)^{n-1} = 2200 \left(\frac{2.410}{20} \right)^{0.35} = 1045.6$

$w_{3-4} = \frac{R(T_3 - T_4)}{1-n} = \frac{0.287(2200 - 1045.6)}{1 - 1.35} = 943.8 \text{ kJ/kg}$

$E_{in} - E_{out} = \Delta E_{system} \Rightarrow -q_{3-4, out} - w_{3-4, out} = u_4 - u_3 \rightarrow q_{3-4, out} = -w_{3-4, out} - (u_4 - u_3)$

4-1: $q_{4-1, out} = u_1 - u_4 = 552.1$

$q_{3-4, out} = 1235 \text{ kJ/kg}$

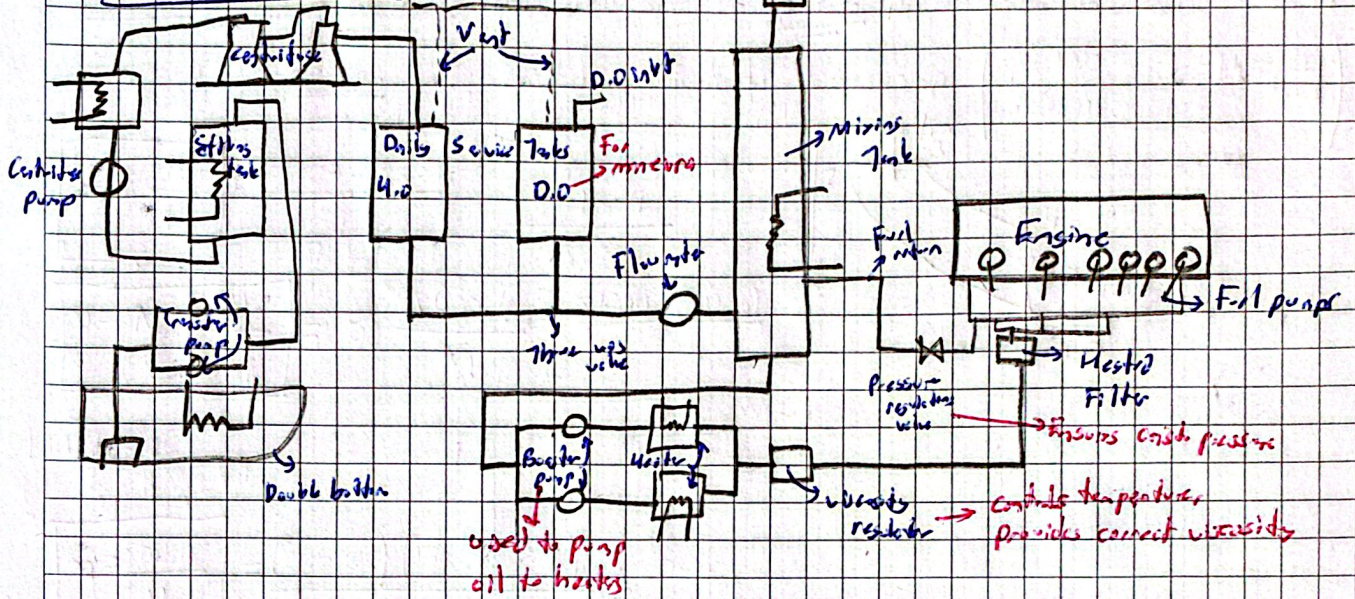
Total $q_{out} = q_{3-4, out} + q_{4-1, out} = 719.6$

$w_{net, out} = q_{in} - q_{out} = 1556 - 719.6 = 836.4$

$\eta = \frac{w_{net, out}}{q_{in}} = \frac{836.4}{1556} = 0.538$

~ MACHINERY SERVICE SYSTEMS AND EQUIPMENTS ~

FUEL OIL SYSTEM



α System also includes atoms and values which can be closed in the context of a disc

LUBRICATING OIL SYSTEM

2. Lubrication oil is stored in bottom of the crankcase, (sump) in a drain tank below the engine

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