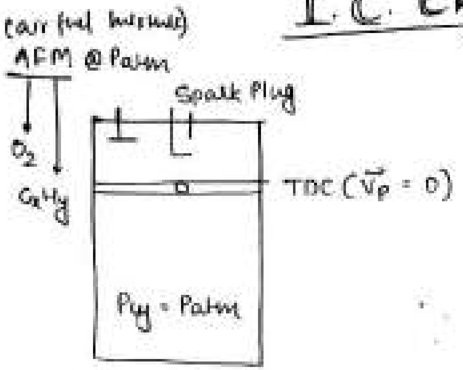
I.C. Engine

2nd june 2017



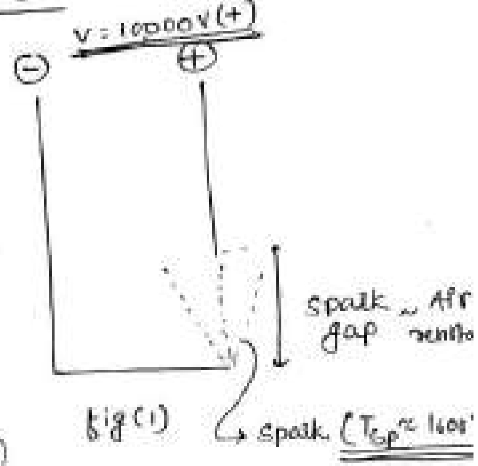
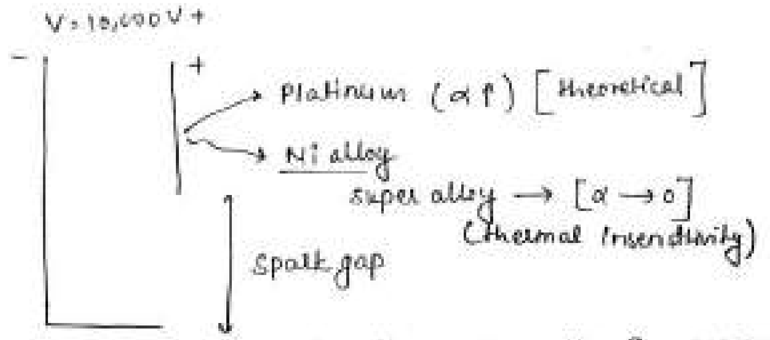


fig (1)

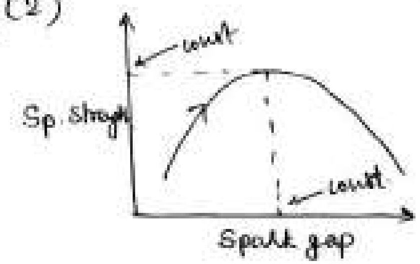
1) For spark plug, platinum is the best suited material (e⁻ density is more) theoretically but nickle alloy should be used for Practical application (fig 2)

2) For sparking high potential difference is required in order to overcome the air resistance present in spark gap (fig 1)



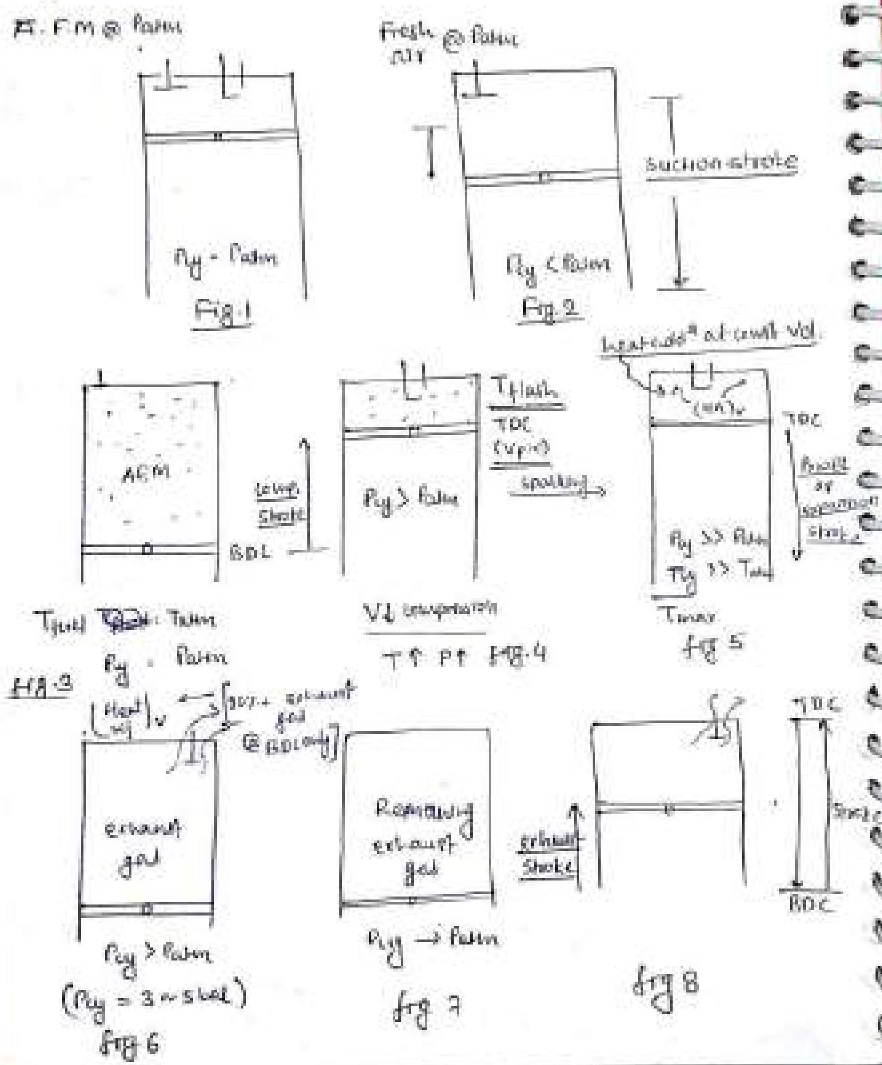
α → thermal coeff. of expansion

fig (2)



Basic of an I.C. Engine

Stroke: when piston move from one dead center to another (TDC → BDC) or (BDC → TDC) is termed as stroke of an engine



Note

1) After Expansion stroke the pressure inside the cylinder is much above the atm. pressure hence as the exhaust valve will open major exhaust gases [90% +] will escape out of the cylinder at BDC only.

⇓

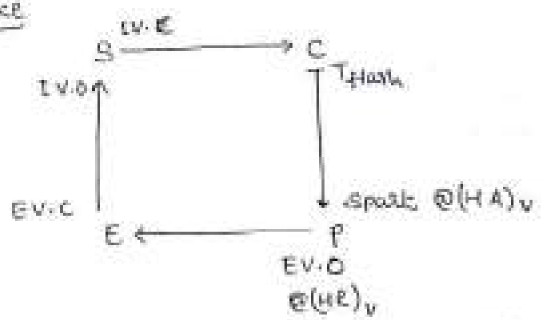
heat rejection at const. volume [fig 6]

2) [After $(HR)_v$ the pressure inside the cylinder reaches to Patm hence to through out remaining exhaust gases [minor exhaust gas] piston displaces from BDC to TDC.]

⇓

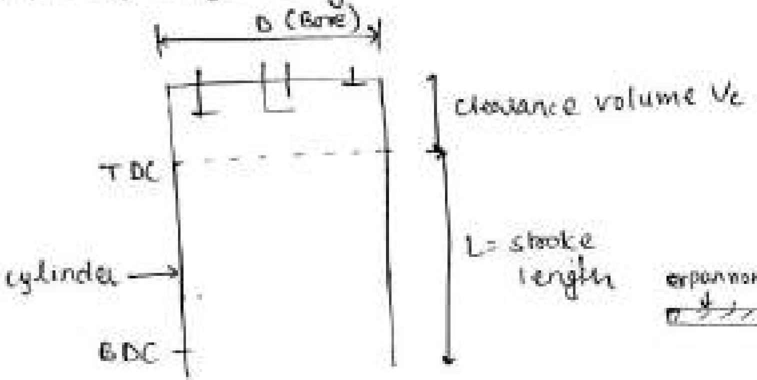
Exhaust stroke [fig 7 & 8]

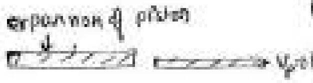
⇒ 4-stroke



Engine Material and Geometry

(I) engine geometry





Reason for clearance volume:

(I) Opening & closing of valve

(II) Thermal expansion

(III) Spark gap

$$\boxed{V_s \text{ (swept volume)} = \frac{\pi}{4} \times D^2 \times L}$$

$$\boxed{\text{Clearance Ratio} = C = \frac{V_c}{V_s}}$$

(II) engine material

Piston & Cylinder

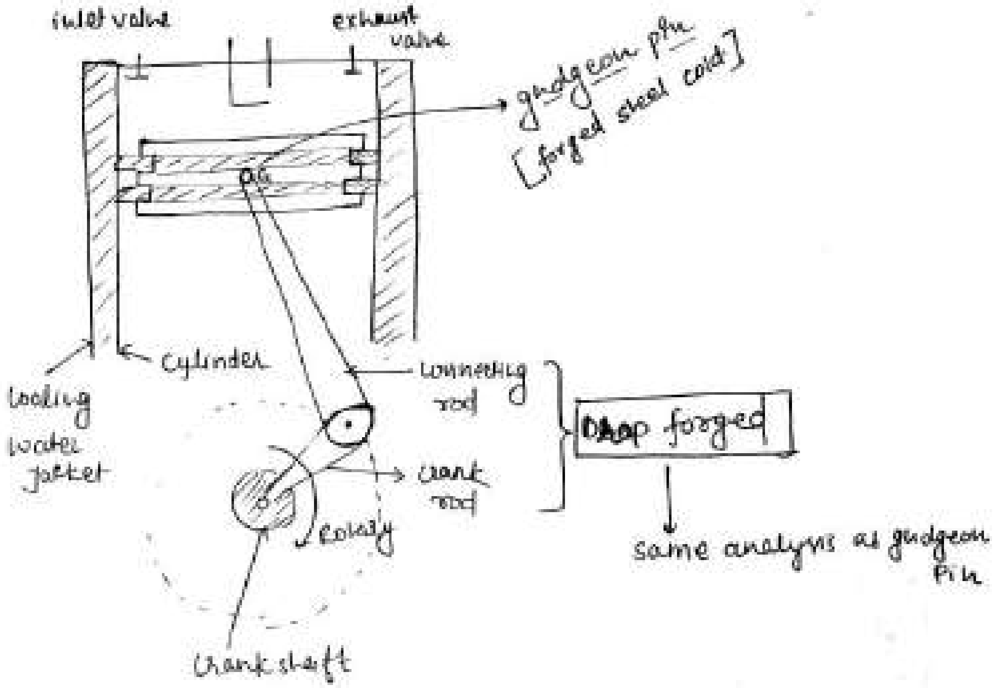
↓

C.I

heavy ~ inertia ⟶ steel alloy ⟶ Aluminium alloy

C.I:
- ↳ $(\text{milage})_{min}$
- ↳ $(\text{strength})_{max}$

Aluminium alloy:
- ↳ inertia ↓
- ↳ $(\text{milage})_{max}$
- ↳ $(\text{strength})_{min}$

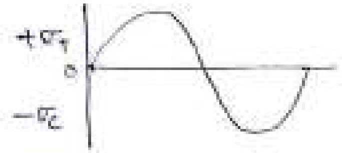


Gudgeon Pin (G)

1) It is subjected to fatigue loading condition hence it should be made up of forged steel to prevent to high failure rate

2) Connecting rod and crank rod is also subjected to same stress condition and it is especially made by drop forging method

fatigue loading condition
(Rev. cyclic loading cond^n) [failure Rate] ↑



stress cycle

cyclic loading

↓

[crack propagation Rate $\rightarrow \infty$]

Gudgeon pin is made by ductile material.

forged steel (cold)

↓

machining (surface-defect $\rightarrow 0$)

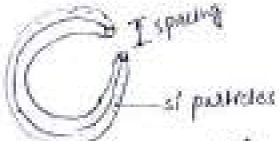
Piston Ring

1) [Piston ring should be brittle because if it is ductile then after some large time it will create a gap from where A.F.M may leak out during compression.]

⇓

Blow-By loss/defect

2) It should be smooth to reduce the frictional losses during reciprocation motion of piston.



N = 2400 rpm (petrol engine generally)

N = 40 rps

1 stroke = $\frac{1}{80}$ sec

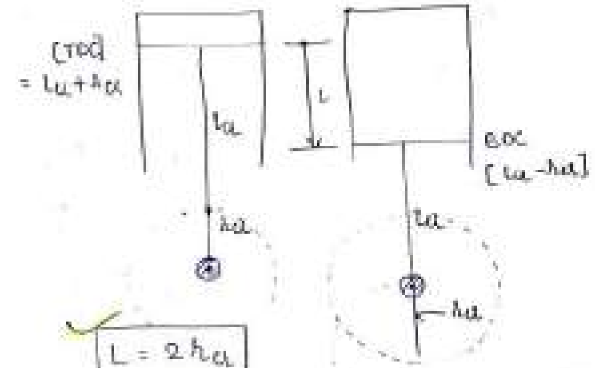
Brittle → Prevents → Blow by loss

Smooth (solid lubrication) (continuous) → Prevents → Frictional losses

smoothness ← (Si) − C-I
or Si − Steel
} material used for piston ring

minimum no. of ring required = 2

Note



$\boxed{L = 2h_{cr}}$

Square ratio engine

$\boxed{L = D}$

Cylinder ratio/
over-square/under-square

$\frac{L}{D}$: cylinder ratio

Air standard Assumption

1) Air as only working fluid $[V_f <<< V_a]$
2) Air should behave like perfect gas R — P – F
3) It follow ideal gas equation => $PV = mRT$
4) $\underbrace{C_p, C_v, \gamma}_{\text{const.}} \neq f(T)$

PROCESS-Ratios (should be more than 1)

Compression

$r_c = \dfrac{V_1}{V_2}$

$r_p = \dfrac{P_2}{P_1}$

$r_T = \dfrac{T_2}{T_1}$

Expansion

$r_e = \dfrac{V_4}{V_3}$

$r_{p,e} = P_3/P_4$

$r_{T,e} = T_3/T_4$

Heat addition/rej

$r_{HA} = \dfrac{V_3}{V_2} = 1 - \underline{IC}$

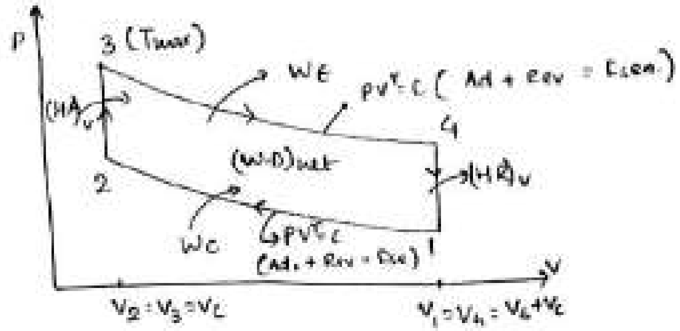
$r_{pHA} = P_3/P_2$

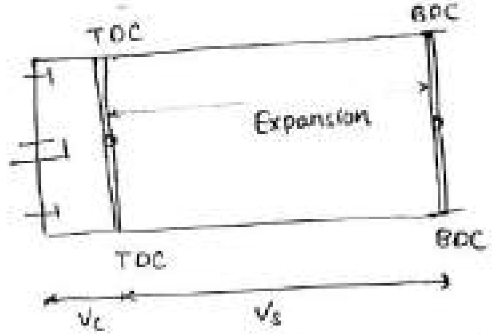
Exc PPT

$r_T HA = \dfrac{T_3}{T_2}$

$$\boxed{r = \frac{V_1}{V_2} = \frac{V_s + V_c}{V_c} = 1 + \frac{1}{C}}$$

Otto cycle or const. volume heat add^n cycle:





$$\eta = \frac{(WD)_{net}}{(HA)_v} = \frac{(HA)_v - (HR)_v}{(HA)_v} = 1 - \frac{(HR)_v}{(HA)_v}$$

$$= 1 - \frac{\int_1^4 C_v dT}{\int_2^3 C_v dT}$$

Air standard ass.

$$\eta_0 = 1 - \frac{T_4 - T_1}{T_3 - T_2} = \frac{1 - T_1 (T_4/T_1 - 1)}{T_2 (T_3/T_2 - 1)}$$

$\dfrac{T_2}{T_1} = (r)^{\gamma-1}$

$\dfrac{T_3}{T_4} = (r_e)^{\gamma-1}$ $\cancel{\dfrac{T_3}{T_4} = (r_e)^{\gamma-1}}$

$r = r_e \quad \dfrac{T_2}{T_1} = \dfrac{T_3}{T_4} \Rightarrow \dfrac{T_4}{T_1} = \dfrac{T_3}{T_2}$

$$\boxed{\eta_0 = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{1}{r}\right)^{\gamma-1}}$$

IFS In an S.I engine working on ideal otto cycle the compression ratio is 5.5. pressure at beginning of compression is 1 bar and temp is 27°C. The pressure at the end of isentropic expansion is 4.5 bar then determine the air std. efficiency and max temp in the cycle. also determine the work done in kJ/kg

$(C_v) = 0.718$ kJ/kg K

Sol^n

$\eta_o = 1 - \left(\frac{1}{5.5}\right)^{0.4} = \underline{0.494}$

$\left(\frac{V_1}{V_2}\right)^{\gamma} = \frac{P_2}{P_1}$

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (5.5)^{0.4}$

$(5.5)^{1.4} \times 1 = P_2 \Rightarrow P_2 = 10.88$ bar

$T_2 = 593.29$

$\frac{P_4}{P_3} = \frac{T_4}{T_3}$

$\frac{10.8}{4.5} = \frac{593.29}{T_3} \Rightarrow T_3 = \cancel{2669.03\,k}\ 2472.04$ K

$(HA)_v = C_v(T_3 - T_2)$

$= 0.718(2472.04 - 593.29)$

$= 1348$ kJ/kg

$W_{net} = 666.37$ kJ/kg

ESE-09 / ESE-06 / GATE-03,04,07,11,14 An engine working on otto cycle has bore = 10 cm and crank radius = 7.5 cm. the clearance volume of cylinder 196.3 cm³ assuming sp heat at const pressure 1 kJ/kg K and gas const (R) = .29 kJ/kgK determine the (i) work done net if Tmax = 2250 K ambient condition are 1 bar pressure 27°C temp

$D = 10$ cm, $L = 2r$, $r = 7.5$, $V_c = 196.3$

$C_p = 1$ kJ/kgK $R = 0.29$

Sol

$V_s = \frac{\pi}{4} \times D^2 \times L = 1178.1$ cm³

$= \frac{\pi}{4} \times 10^2 \times 2 \times 7.5 = 1178.1$ cm³

$C = \frac{V_c}{V_s} = \frac{196.3}{1178.1} = 0.166$

$r = 1 + \frac{1}{C} = 1 + \frac{1}{0.166} = 7.02$

$\left[\begin{array}{l} C_p - C_v = R \\ \Rightarrow C_v = 0.71 \\ \gamma = 1.408 \end{array}\right]$

$\eta_o = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$

$= 1 - \left(\frac{1}{7.02}\right)^{0.408}$

$= 0.548$

$\left(\frac{V_1}{V_2}\right)^{\gamma-1} = \frac{T_2}{T_1}$

$T_2 = 663.62$ K

$W_{net} = \eta_o \times (HA)_v$

$= 0.548 \times C_v(T_3 - T_2)$

$= 0.548 \times 0.71 (2250 - 663.62)$

$= \cancel{1126.3}\ 617.05$ kJ/kg

(ii) Also determine the max pressure in the cycle.

$\left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow P_2 = 15.48$ bar

$\frac{P_3}{P_2} = \frac{T_3}{T_2} \Rightarrow P_3 = 52.49$ bar

$\Rightarrow$

$\tau_{max} = \frac{P_{max}\, D}{2t}$

thickness of cylinder t

Note $\downarrow C_p - \downarrow C_v = R = \text{const.}$ (C_p C_v decrease by same amount)

$$\frac{C_p \downarrow}{C_v \downarrow} = r \uparrow \downarrow$$

$$\eta_0 \downarrow = 1 - \left\{ \frac{1}{(r)^{\gamma - 1}} \right\} \uparrow$$

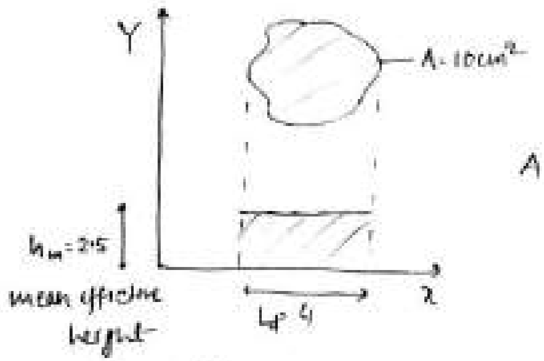
✓ if r decreases then air quality will decrease which decreases the air standard efficiency

Mean Effective Pressure

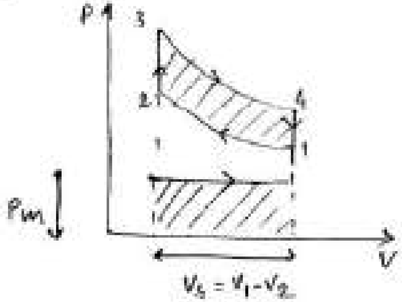
It is an imaginary pressure which will give us the same work done as by the actual cycle for the same change in volume $[V_s = V_1 - V_2]$

$$\boxed{W_{net} = P_m \times V_s}$$

$\hookrightarrow$ Imaginary



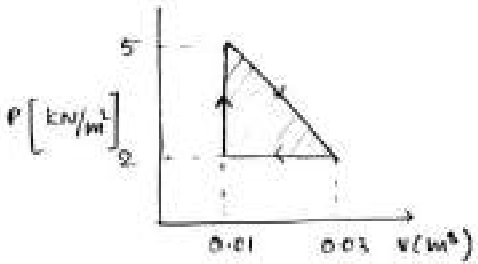
$A = 10 = h_m \times L_f$



$$(WD)_{net} = \int_1^2 P\,dv = P_m (V_1 - V_2)$$

$$\boxed{(WD)_{net} = P_m V_s}$$

Crate The fig below shows a thermodynamic cycle undergone by a certain system determine the mean effective pressure in N/m^2

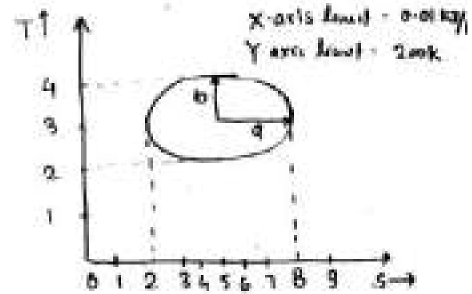


$$W_{net} = \frac{1}{2} \times 3 \times 1000 \times 0.02 = P_m (0.02)$$

$1500\,Pa = P_m$

$1500\,N/m^2 = P_m$

Q. Determine the mean effective pressure for below TD system if V_s is $0.02\,m^3$ (P_m in bar)



Sol^n

$$\oint dQ = \oint dW$$

$$(WD)_{net} = \pi \times a \times b$$

$$= \pi \times 3 \times 0.01 \times 200 \text{ kJ}$$

$$P_m \times V_s = P_m \times 0.02 = \pi \times 3 \times 0.01 \times 200$$

$$\Rightarrow P_m = 942.47 \text{ kPa} = 9.42 \text{ bar}$$

✓ Area of ellipse $= \pi \times (a \times b)$

Basic idea of diesel engine (cycle)

1. In diesel cycle during suction only air is entered into the cylinder and the compression ratio is very high so that the temp after compression should be above the selfignition temp of the fuel
2. In diesel cycle fuel is entered into the cylinder by the use of fuel injector in spray form

Otto cycle

$r_o \approx 6$ to $12 \Rightarrow \frac{T_2}{T_1} = (r)^{\gamma-1} \Rightarrow [T_2 = T_1 r^{\gamma-1}] \leq T_{flash}$

sparking

diesel cycle

$r_d \approx 16$ to $22 \Rightarrow \{T_2 = T_{AC} = T_1 (r_d)^{\gamma-1} \approx 1000\,K\} > T_{self\text{-}ignition}$

$(T_{AC})_{cyl} > (T_{self\text{-}ignition})_{diesel}$

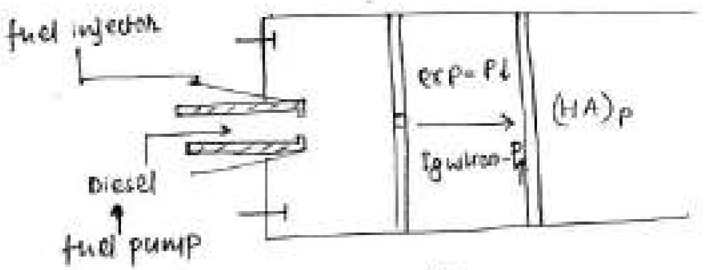
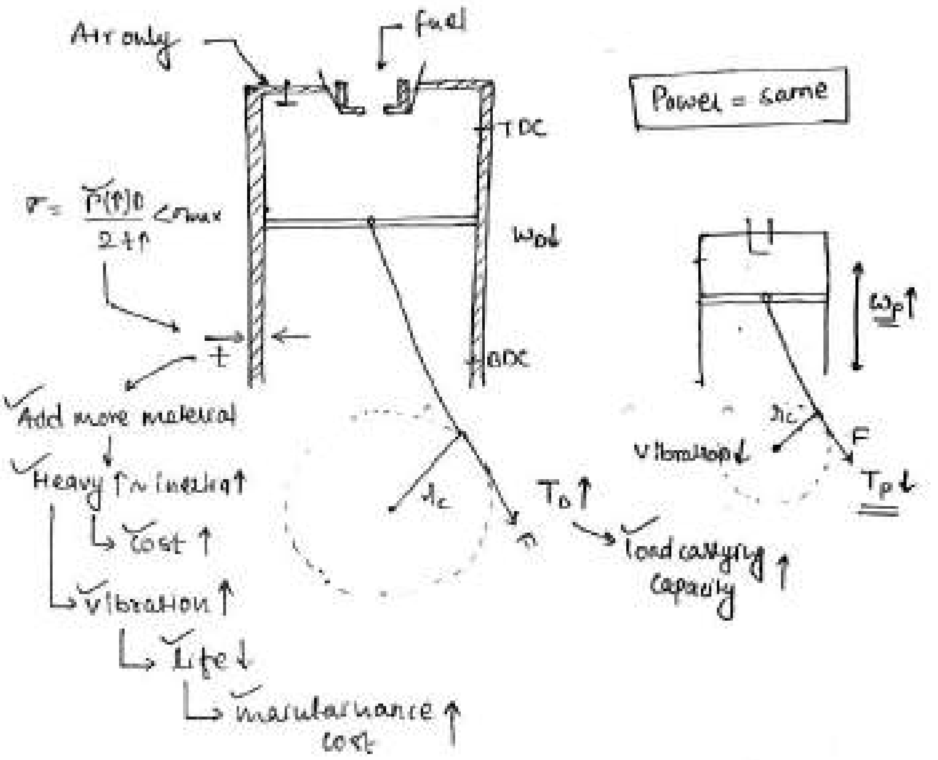


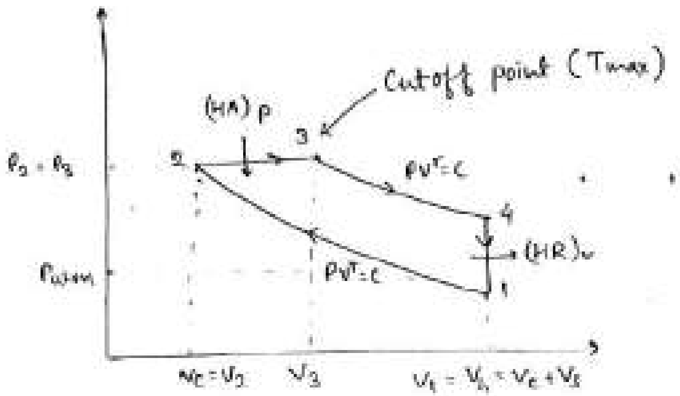
fig. 1

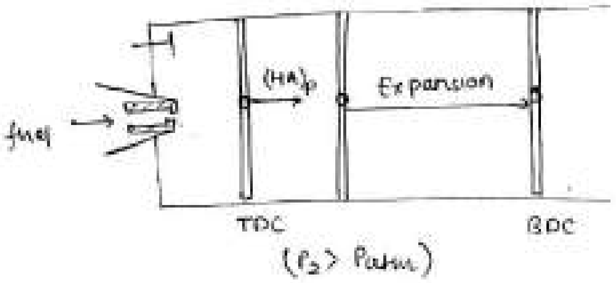
In diesel cycle after compression as the fuel is injected
(i) Due to fuel ignition pressure should increase.
(ii) Due to expansion pressure should decrease

hence in diesel cycle heat addition is assumed to be at const. pressure shown in fig [1]



Diesel cycle or const. pressure heat addition cycle





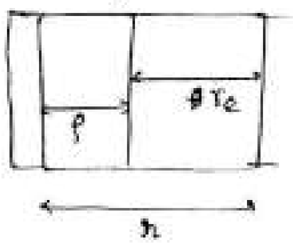
$$r = \frac{V_1}{V_2}$$

$$r_e = \frac{V_4}{V_3} = \frac{V_4/V_2}{V_3/V_2} = \frac{r}{\rho}$$

Cut-off ratio (where fuel injection terminated)

$$\boxed{\rho = \frac{V_3}{V_2}}$$

Ratio theory



$$r = \rho \times r_e = \frac{V_4}{V_2} = \frac{V_3}{V_2} \times \frac{V_4}{V_3} = \frac{V_4}{V_3} \cdot \frac{V_3}{V_2}$$

$$\rho = r/r_e$$

$$r_e = r/\rho$$

$$\eta = \frac{(WD)_{net}}{(HA)_p} = \frac{(HA)_p - (HR)_v}{(HA)_p} = 1 - \frac{(HR)_v}{(HA)_p}$$

Air standard assumption

$$\eta_d = 1 - \frac{T_4 - T_1}{\gamma(T_3 - T_2)}$$

*

$$\eta_d = 1 - \frac{(\rho^\gamma - 1)}{\gamma(r^{\gamma-1}\rho - r^{\gamma-1})}$$

$$\boxed{\eta_d = 1 - \left(\frac{1}{r}\right)^{\gamma-1}\left[\frac{(\rho^\gamma - 1)}{\gamma(\rho - 1)}\right]}$$

*

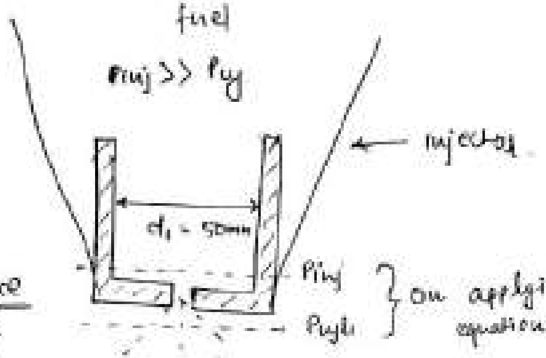
$$\frac{T_2}{T_1} = (r)^{\gamma-1} \Rightarrow T_2 = T_1 (r)^{\gamma-1}$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = \rho \Rightarrow T_3 = T_1 r^{\gamma-1} \times \rho$$

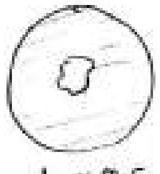
$$\frac{T_4}{T_3} = \left(\frac{\rho}{r}\right)^{\gamma-1}$$

$$\Rightarrow T_4 = T_1 \rho^\gamma$$

fuel-injection principle



} on applying bernoulli's equation we get velocity



$d_0 \propto 0.5$

Rough opening of orifice

physical mixing

– physical delay / lag

Atomisation

↓ (surface area)↑

Vapourisation (liquid → vapour)

↓

Uniform distribution (Air-fuel mixture)

$T_{air} > T_{self\text{-}ignition}$

$T_{atm} = T_{fuel}$

↓

Self ignition

i) Due to high pressure difference and rough opening of orifice atomisation will occur

ii) Process by which liquid split into smallest possible size → Atomisation

iii) Due to atomisation surface area increases which leads to vapourisation

iv) (imp) The time required for the physical mixing of air and fuel particles to create Air-fuel mixture is known as "physical delay or lag"

G-14 In a C.I. engine the inlet pressure is 1 bar and the pressure at the begining of isentropic expansion is 32.42 bar. Assuming ratio of sp. heat γ as 1.4 determine the air-standard efficiency in % if expansion ratio is 8

Soln

$P_3 = P_2 = P_{max} = 32.42$ bar

$P_1 V_1^{\gamma} = P_2 V_2^{\gamma}$

$r = \left(\frac{P_2}{P_1}\right)^{1/\gamma} = 12$

ratio theory

$\rho = \frac{r}{r_e} = \frac{12}{8} = 1.5$

$$\eta_d = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{\rho^{\gamma}-1}{\gamma(\rho-1)}$$

$$= 1 - \left(\frac{1}{12}\right)^{1.4-1} \times \frac{1.5^{1.4}-1}{1.4(1.5-1)}$$

$$\eta_d = 59.59\ \%$$

Note

Statement: cut off takes place at %P of stroke
or heat addn ends at %P of stroke

$V_3 - V_c = P\%(V_s)$

$V_3 - V_c = P\%(V_1 - V_2)$

$\Rightarrow (\rho - 1) = P\%(r-1)$

$\leftarrow V_f = P \times V_s$

$(\rho - 1) = P(r-1)$

final

$$\boxed{\rho - 1 = \%P\,(r-1)}$$

7.14 A diesel engine has compression ratio of 17 and cut off takes place at 10% of the stroke assuming $\gamma = 1.4$ determine the air st. efficiency of diesel engine

Sol

$$\rho - 1 = 0.1(17 - 1)$$

$$\rho = 1.6 + 1 = 2.6$$

$$\eta_d = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \frac{(\rho^{\gamma} - 1)}{\gamma(\rho - 1)}$$

$$= 0.596$$

$$\eta_d = 59.60\%$$

The ambient condition is given as 1 bar pressure and 15°C temp determine the heat added in kJ/kg and also determine the pressure difference required for heat rejection process?

Sol

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \Rightarrow T_2 = T_1 (r)^{\gamma-1}$$

$$T_2 = 288 (17)^{.4}$$

$$= \underline{894.48\,K}$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = \rho = 2.6$$

$$T_3 = \underline{2325.65\,K}$$

$$(HA)_P = C_P (T_3 - T_2)$$

$$= \underline{1438.23\ kJ/kg}$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = 17^{1.4}$$

$$P_2 = P_3 = \underline{52.79\ bar}$$

$$\frac{P_3}{P_4} = \left(\frac{V_4}{V_3}\right)^{\gamma}$$

$$r_e = \frac{r}{\rho}$$

$$= 17/2.6$$

$$= 6.53$$

$$\frac{P_3}{P_4} = \cancel{52.79}\,(6.53)^{1.4}$$

$$\Rightarrow P_4 = 3.81\ bar$$

$(HR)_V = ?$

$\eta_d = 1 - \frac{(HR)_V}{(HA)_P}$

$.596 = 1 - \frac{(HR)_V}{(HA)_P}$

$1 - 0.596 \times 1438.23 = (HR)_V$

$851.02\ kJ/kg = (HR)_V$

Q determine the mean effective pressure also

$$\eta_d = \frac{W_{net}}{HA}$$

$$0.596 \times 1438.23 = W_{net}$$

$$857.24\ kJ/kg = W_{net}$$

$$P v_1 = R T_1$$

$$v_1 = \frac{0.287 \times 288}{100}$$

$$v_1 = 0.826\ m^3/kg$$

$$v_2 = \frac{v_1}{r} = 0.0486\ m^3/kg$$

$$W_{net} = P_m (V_1 - V_2)$$

$$857.24 = P_m (0.826 - 0.0486)$$

$$P_m = 1102.7\ kPa$$

$$P_m = 11.023\ bar$$

Dual Cycle

$(HA)_V$ otto cycle $\rightarrow \eta_o \approx \underline{6\ to\ 12} \Rightarrow$ S.I (Petrol or gas) engine $\Rightarrow P \cdot T_{[...]}$ Spark plug

$(HA)_V + (HA)_P$ dual cycle $\rightarrow \eta_{du} \approx \underline{12\ to\ 16} \Rightarrow$ light $\Rightarrow$ C.I (diesel engine) $\Rightarrow P \cdot T_P \uparrow \times W_d \uparrow$

$\hookrightarrow (T_3)_{cy} >$ Heat generation). fuel injector

$(HA)_P$ diesel cycle $\rightarrow \eta_d \approx \underline{16\ to\ 22} \Rightarrow$ heavy $\Rightarrow$ C.I (diesel engine) $\Rightarrow P \cdot T_d \uparrow \times W_d$

Scanned by CamScanner

Dual cycle / semi. diesel engine

1) The fuel is same for the diesel cycle and dual cycle that is why dual cycle is also called as semi-diesel cycle

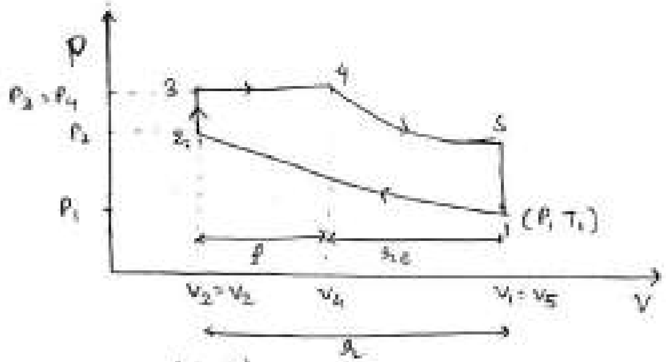
2) In dual cycle, as the fuel is injected some heat is added at const. volume condition and due to which pressure rises at TDC only

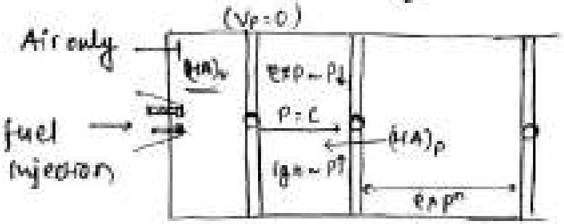
1) $r_{dual} \approx$ 12 to 16

$\frac{T_2}{T_1} = (r)^{\gamma-1} \Rightarrow \boxed{T_2 = T_1(r)^{\gamma-1} \geq (T_{self\,ignition})_d}$

2) $(HA)_v < (HA)_p$

3) $(T_{self\text{-}ign})_{diesel} < (T_{self\text{-}ign})_{Petrol}$





$$\eta = \frac{(WD)_{net}}{(HA)_v + (HA)_p} = 1 - \frac{(HR)_v}{(HA)_v + (HA)_p}$$

Air standard ass.

$$\eta_{dual} = 1 - \frac{C_v(T_5 - T_1)}{C_v(T_3 - T_2) + C_p(T_4 - T_3)}$$

$$= 1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)}$$

$$= 1 - \frac{(\alpha\rho^{\gamma} - 1)}{(r^{\gamma-1}\alpha - r^{\gamma-1}) + \gamma(r^{\gamma-1}\alpha\rho - r^{\gamma-1}\alpha)}$$

imp

$$\boxed{\eta_{dual} = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \frac{(\alpha\rho^{\gamma} - 1)}{(\alpha - 1) + \alpha\gamma(\rho - 1)}}$$

* $\frac{T_2}{T_1} = (r)^{\gamma-1} \Rightarrow T_2 = T_1(r)^{\gamma-1}$

$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2} = \alpha\right) \Rightarrow T_3 = T_1(r)^{\gamma-1} \cdot \alpha$

$\frac{T_4}{T_3} = \left(\frac{V_4}{V_3} = \rho\right) \Rightarrow T_4 = T_1(r)^{\gamma-1} \cdot \alpha \cdot \rho$

$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{\gamma-1} = \left(\frac{1}{r_e}\right)^{\gamma-1} = \left(\frac{\rho}{r}\right)^{\gamma-1} \Rightarrow T_5 = T_1(r)^{\gamma-1} \cdot \alpha \cdot \rho \cdot \left(\frac{\rho}{r}\right)^{\gamma-1}$

$= T_1 \cdot \alpha \cdot \rho^{\gamma}$

$r = \frac{V_1}{V_2}$

$r_e = \frac{V_5}{V_4} = \frac{r}{\rho}$

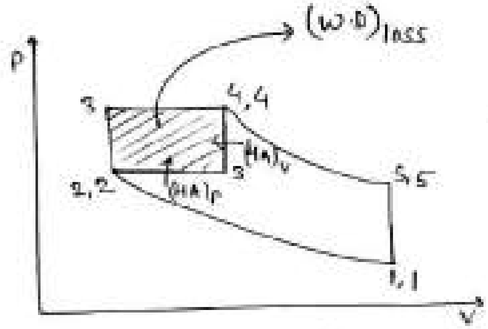
$\rho = \frac{V_4}{V_2 = V_3}$

Explosion ratio

$\alpha = \frac{(P_3)_{max}}{P_2}$

heat addition order in dual cycle

1) HA. take place first at const volume then at const pressure because theoritically if $(HA)_p$ occurs first then $(HA)_v$ the area under the PV diagram will be less and also the efficiency

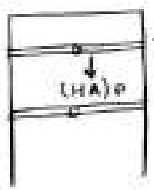


$$(WD)_{net}\{(HA)_v \to (HA)_p\} > (WD)_{net}\{(HA)_p \to (HA)_v\}$$

$$\eta\uparrow = \frac{(WD)_{net}}{HA} = \frac{(WD)_{net}}{(WD)_{net} + (HR)_v}\uparrow = \left\{\frac{1}{\left\{1 + \frac{(HR)_v}{(WD)_{net}\uparrow}\right\}\downarrow}\right\}$$

$$\eta\{(HA)_v \to (HA)_p\} > \eta\{(HA)_p \to (HA)_v\}$$

2) Practically if $(HA)_p$ occurs first then for $(HA)_v$ the piston will have to stop in the middle of it's motion which is impossible in nature.



TDC($V_P = 0$)

Piston motion (TDC → BDC)

impossible

$(HA)_v$

Analysis of air standard efficiency

$$\eta_{a,d,dual} = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(\alpha\rho^{\gamma} - 1)}{[(\alpha-1) + \alpha\gamma(\rho-1)]}$$

1) $\eta\uparrow \to r\uparrow$

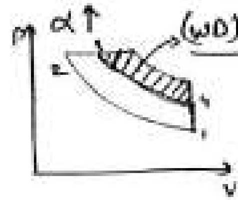
2) $\eta\downarrow \to \rho\uparrow$

3) $\eta\uparrow \to \gamma\uparrow$

⇒ α should not be increase because $\alpha\uparrow$ explosion occurs.

4) $\eta\uparrow \to \alpha\uparrow$

1) $\eta\uparrow \propto 1 - \left(\frac{1}{r\uparrow}\right)\downarrow$



2) Cut-off ratio

$$\eta\downarrow \propto 1 - \left(\frac{\rho^{\gamma}\uparrow}{\rho\uparrow}\right)\uparrow$$

$\downarrow \gamma = \frac{C_p\uparrow}{C_v\uparrow}$

$$\eta = \frac{(WD)_{net}\uparrow}{(HA)\uparrow}$$ (Low grade $\xrightarrow{\text{completely}}$ high grade) (not possible)

3) $\eta\uparrow \propto 1 - \frac{1}{\gamma\uparrow}$ and also $\eta \propto \frac{1}{r^{\gamma-1}}\uparrow$

$$\propto (\rho^{\gamma\uparrow})\uparrow$$

$\boxed{\text{E-11}}$ A diesel engine having a compression ratio of 16 and cut off take place at 10% of stroke assuming sp heat at const volume as 0.718 KJ/kgK an adiabatic index of 1.4 determine the % decrease in efficiency if sp heat at const volume is increase by 10%.

Sol

$(\rho-1) = \rho\%(r-1) \Rightarrow (\rho-1) = 0.1(17-1) = 1.6$

$\rho = 2.6$

$$\eta_d = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(\rho^{\gamma}-1)}{\gamma(\rho-1)}$$

$\eta_d = \underline{59.05\%}$

$C_{v_1} = 1.1 C_v$

$C_{p_1} - C_{v_1} = C_p - C_v$

$C_{p_1} - 1.1C_v = \gamma C_v - C_v$

$C_{p_1} = \gamma C_v + 0.1 C_v$

$$\frac{C_{p_1}}{C_{v_1}} = \frac{\gamma + 0.1}{1.1} = 1.364$$

$$\eta_{d_1} = 1 - \left(\frac{1}{r}\right)^{\gamma_1-1} \frac{(\rho^{\gamma_1}-1)}{\gamma_1(\rho-1)}$$

$\eta_{d_1} = \underline{55.65\%}$

% decrease in efficiency $= \left|\frac{\eta_{d_1} - \eta_d}{\eta_d}\right| \times 100$

$= |-5.8|\% = 5.8\%$

Thermochemistry

It is a branch of science in which thermal energy is produced from the fuel by oxidation reaction then only the chemical energy will released.

Q Propane butane (C_4H_{10}) is burnt in oxygen atmosphere then determine the % of carbon mono oxide in exhaust if 1.5 mole of oxygen is less available then stochimetric combustion.

Sol

$C_4H_{10} + 6.5\,O_2 \longrightarrow 5H_2O + 4CO_2 + 0$

$C_4H_{10} + 5\,O_2 \longrightarrow 5H_2O + 3CO + CO_2$

$$\%\,CO = \frac{3}{5+3+1} \times 100 = \frac{3}{9} \times 100 = 0.33 \times 100 = \underline{33.3\%}$$

Q Cetane is burnt ($C_{10}H_{22}$) in air atmosphere then determine the % of carbon di-oxide in exhaust if 3.5 mole are less

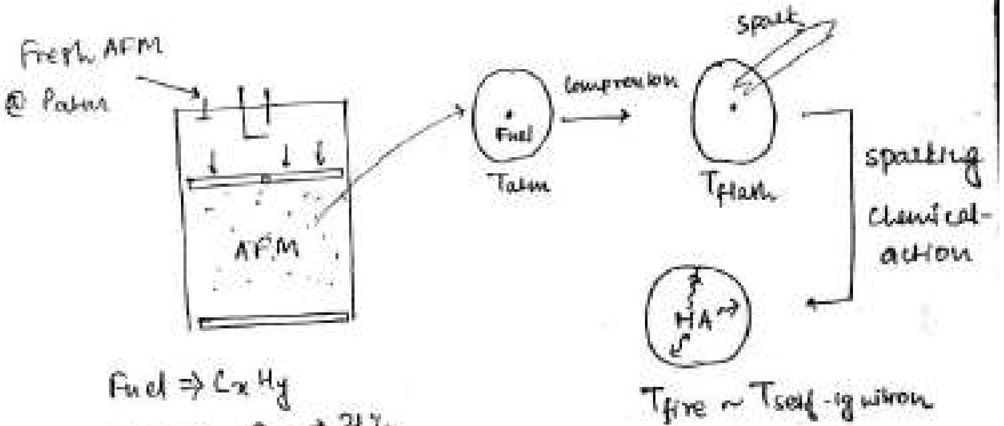
Sol

$$C_{10}H_{22} + 15.5\left\{O_2 + \frac{79}{21}N_2\right\} \longrightarrow 11H_2O + 10CO_2 + 15.5 \times \frac{79}{21}N_2 + \Delta H + \Delta C$$

$$C_{10}H_{22} + 12\left\{O_2 + \frac{79}{21}N_2\right\} \longrightarrow 11H_2O + 3CO_2 + 7CO + 12 \times \frac{79}{21}N_2$$

$$\%\,CO_2 = \frac{3}{3 + 11 + 7 + 12 \times \frac{79}{21}} \times 100$$

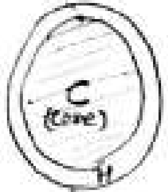
$\%\,CO_2 = 4.53\%$



Fuel $\Rightarrow C_xH_y$

Air $\Rightarrow$ $O_2 \rightarrow 21\%$, $N_2 \rightarrow 79\%$

(Imp)



- H burn first
- H burns rapidly
- $\Delta H >> \Delta C$

$H_2 \longrightarrow$ explosive element

Stoichiometry or perfect reaction

$$CH_4 + 2\left\{O_2 + \frac{79}{21}N_2\right\} \xrightarrow[\text{combu}^n]{\text{oxd}^n} 2H_2O + CO_2 + 2\times\frac{79}{21}\times N_2 + \underbrace{\Delta H + \Delta C}_{\boxed{\text{Calorific value}}}$$

Rich fuel

25% less air

$$CH_4 + 1.5\left\{O_2 + \frac{79}{21}N_2\right\} \xrightarrow[\text{comb}^n]{\text{oxd}^n} \underbrace{2H_2O + CO + 1.5\frac{79}{21}N_2}_{\boxed{\text{Exhaust gas}}}$$

50% less air

$$CH_4 + 1\left\{O_2 + \frac{79}{21}N_2\right\} \xrightarrow[\text{comb}^n]{\text{oxd}^n} 2H_2O + \underbrace{C}_{\text{unburnt hydro carbon}} + \frac{79}{21}N_2$$

lean fuel

$$CH_4 + 3\left\{O_2 + \frac{79}{21}N_2\right\} \longrightarrow 2H_2O + CO_2 + 3\times\frac{79}{21}N_2 + O_2$$

Q Propane (C_3H_8) is burnt in O_2 atmosphere with 10% deficit O_2 w.r.t to the perfect combustion assuming no hydrocarbon in the product determine the percentage of CO in the exhaust.

$$C_3H_8 + 5O_2 \longrightarrow 4H_2O + 3CO_2$$

$$C_3H_8 + 4.5O_2 \longrightarrow 4H_2O + CO + 2CO_2$$

$$\% \text{ of } CO = \frac{1}{4+1+2}\times 100 = \frac{1}{7}\times 100 = 0.1428\times 100 = 14.28\%$$

Stages of combustion (Carbon)

$$C \xrightarrow[O_2]{\text{oxd}^n} CO \xrightarrow[O_2]{\text{oxd}^n} CO_2$$

Equivalance Ratio (ϕ)

$$\phi = \frac{m_{f,\text{actual}}/m_a}{m_{f,\text{stoichiometry}}/m_a} = \frac{(FAR)_{act}}{(FAR)_{sto}} \quad \begin{matrix} >1 \leftarrow \text{Rich} \\ 1 \leftarrow \text{stoich.} \\ <1 \leftarrow \text{lean} \end{matrix}$$

$$(AFR)_{stoch} = \frac{m_a}{m_f} = \frac{2\left\{2\times16 + \frac{79}{21}\times2\times14\right\}}{1\times12+4\times1} = 17.16$$

25% less $(AFR)_{act} = \frac{1.5\times\left\{2\times16 + \frac{79}{21}\times2\times14\right\}}{1\times12+4\times1} = 12.8$

$$\phi = \frac{17.16}{12.8} = \underline{1.33}$$

$\longrightarrow$ 33.3% fuel is more

Swept volume

$N = 2400$ rpm

$\frac{N}{60} = 40$ rps

- 2 stroke: 40 suction/sec
- 4 stroke: 20 suction/sec

2 stroke: E, C

4 stroke: E, C, P

$$\dot{V}_s = \frac{\pi}{4} \times D^2 \times L \times K \times \frac{N}{60 \times n} \ m^3/sec$$

K → no. of cylinder

$$n = \frac{\text{no. of stroke}}{2}$$

Energy Transformation:

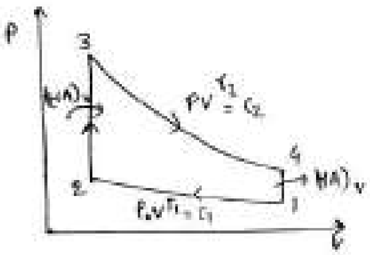
1) Heat addition

$$\text{1 kg fuel} \xrightarrow{\text{Comb}^n} H.A/kg = (C.V)_{fuel}$$

$$(H.A)/sec = \dot{m}_f \times (C.V)_{fuel}$$

Near to end of Exhaust

Actual PV diagram



1) At the end of $(HR)_v$ piston displaces from BDC to TDC to throw out the remaining exhaust gases [fig.(1)]

2) At the very end of exhaust stroke exhaust valve will close down. and for the remaining exhaust stroke their will be the compression of exhaust gases [fig (2)]

3) Due to the compression of exhaust gases

$$P_{cy} \approx (1.1 \text{ to } 1.2)\ P_{atm}$$

at the start of suction stroke their is no entry of fresh A.F.M. [fig (3)].

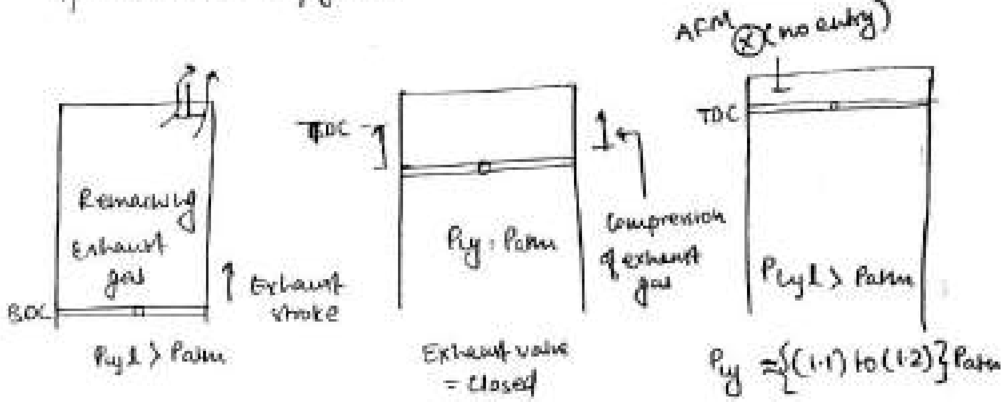


fig (1) fig (2) fig (3)

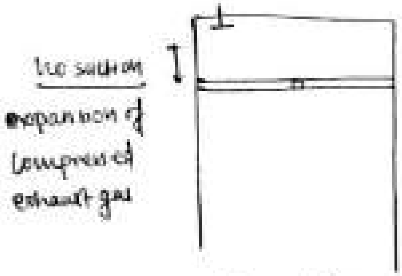
4) Hence some piston movement during suction stroke is wasted for the expansion of the previous cycle compressed gas and the actual suction volume is less than the theoritical swept volume (fig 4)

Volumetric Efficiency: It is a ratio of actual volume of the air that enter into the ~~cycle~~ cylinder to the theoritical swept volume.

$$\eta_{vol} = \frac{\dot{V}_a}{\dot{V}_s}$$

$$P_1 \dot{V}_a = \dot{m}_a RT \qquad \dot{V}_s = \frac{\pi}{4} \times D^2 \times L \times K \times \frac{N}{60 \times 2}$$

2 → 4-stroke



$P_{cy} \rightarrow P_{atm}$

(fig 4)

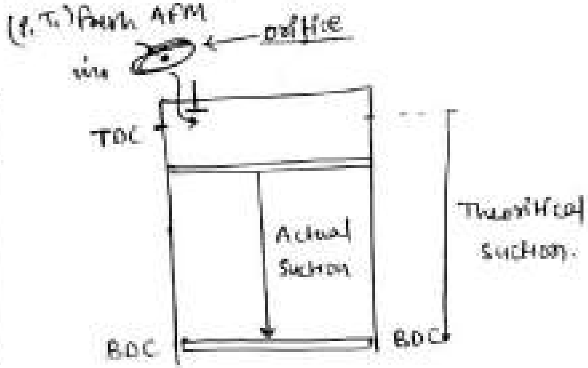
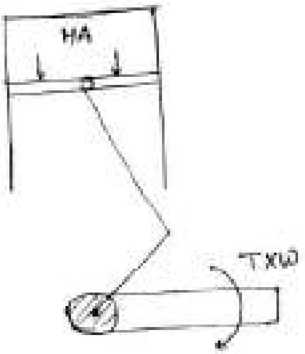


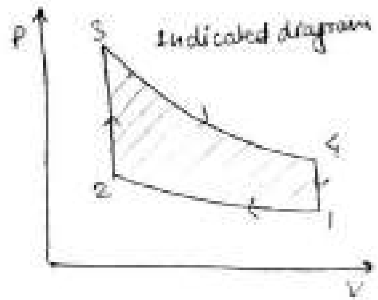
fig (5)

Energy Transformation

2) Indicated Power (I.P)

It is energy available to the piston in the reciprocating nature due to the expansion of gas and it is determined by using PV diagram ~~tool~~ (Indicated diagram)

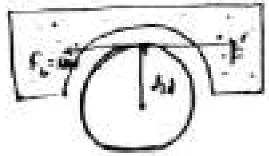


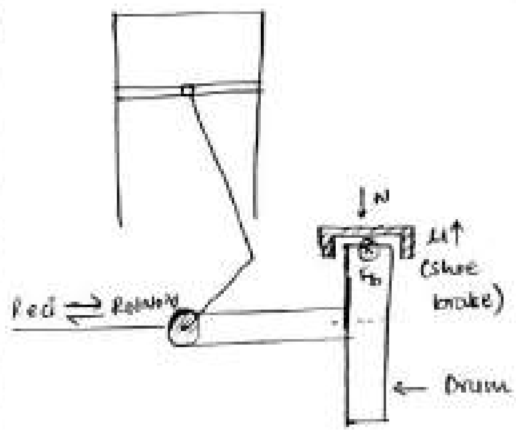


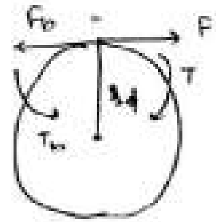
$$I\,WD/_{sec} = \frac{I.P.}{\text{Piston (reciprocating)}}$$

$$\left[IP = I\,WD/_{sec} = P_m \times \dot{V}_s \right]$$

3) Break power







Brake Power (B.P) = $T_b \times \omega$

Rotary

Shaft

Brake power : It is the energy available engine shaft and it is determined by using brake mechanism.

Brake Power (Power developed) = $T_b \times \omega$ ⟶ $\omega = \frac{2\pi N}{60}$

$T_b = F_b \times r_d$

F_b ⟶ μN Brake force

Brake load ⟵ $m_b \times g$

I.P = F.P + B.P

Indicated Power = $\underbrace{\text{Friction Power}}$ + Brake power

It can be controlled only

$$\boxed{\eta_{mechanical} = \frac{B.P}{I.P}}$$ ← Performance

$$\eta_{mech} = \frac{B.P\uparrow}{I.P}$$ → friction power ↓

Thermal Efficiency:

η_{th} → $\eta_{ith} = \frac{I.P}{(HA)/sec}$ — Indicated thermal eff.

η_{th} → $\eta_{bth} = \frac{B.P}{(HA)/sec} = \eta_{engine} = \eta_{overall}$ — brake thermal eff

Mean effective pressure:

$B.P = BWD/sec = P_{b,m} \times \dot{V}_s$ ← [if only mean eff. pressure given]

$I.P = IWD/sec = P_{i,m} \times \dot{V}_s$

Specific

mathematically $\quad X_{S_Y} = \frac{Y}{X}$

sfc → $bsfc = \frac{\dot{m}_f}{B.P} = \frac{kg}{kW\text{-}h} \propto$ Economy

sfc → $isfc = \frac{\dot{m}_f}{I.P}$

$$bsfc\downarrow = \frac{\dot{m}_f\downarrow}{BP\uparrow} \propto \text{Economy}\uparrow$$

brake sp. air consumption $= \frac{\dot{m}_a}{B.P}$

$\boxed{E\text{-}15}$ $\quad Power_{S_{size}} = size/Power$

✓ for the same power $(size)_{CI} > (size)_{CI}$

Note 1) bsfc (brake sp. fuel consumption) is defined as the amount of fuel consumption for the 1 kW of power

2) It is used to compare any two engines on the basis of economy.

Q. The following observation were made on the petrol engine are - (i) Amount of fuel consumption is 3kg/h

(ii) Calorific value of fuel is 42.2 MJ/kg

engine develops the power of 11.25 kW. the engine efficiency is 50% of it's air standard efficiency of the engine the clearance volume of the engine is 200 cm³ and cylinder ratio is 1.1 then determine the, compression ratio, cylinder dimensions.

Sol.

$$\eta_{engine} = 0.5\,\eta_{air\ standard}$$

$$\frac{B.P}{(HA)_s} = 0.5\left[1 - \left(\frac{1}{r}\right)^{\gamma-1}\right]$$

$$\frac{11.25}{\frac{3}{3600}\times 42200} = 0.5\left[1 - \left(\frac{1}{r}\right)^{.4}\right]$$

$$0.61 = 1 - \left(\frac{1}{r}\right)^{.4}$$

$$\left(\frac{1}{r}\right)^{.4} = .389$$

$$\Rightarrow r = 10.58$$

$r = 1 + \frac{1}{c}$

$10.58 = 1 + \frac{1}{c}$

$c = \frac{1}{9.58} = 0.104$

$c = \frac{V_c}{V_s} = \frac{200}{V_s} = 0.140 \Rightarrow V_s = 1916\ cm^3$

$V_s = \frac{\pi}{4} \times D^2 \times L \qquad \frac{L}{D} = 1.1$

$= \frac{\pi}{4} \times D^3 \times 1.1$

$\Rightarrow D = 13.04\ cm$

$L = 14.34\ cm$

Q
E-08 Following data are for the four stroke, 4-cylinder, petrol engine cylinder dimensions are bore = 11cm and stroke is of 13 cm, engine speed 2250 rpm, brake power of engine is 50 kW and the frictional power of engine 15 kW fuel consumption rate is 10.5 kg/hr $(C.V)_f = 50000\ kJ/kg$ the air fubilation rate of the engine is 300 kg/hr at the ambient condition of 1.03 bar, 15°C temp.

determine - (1) mechanical efficiency
(2) Brake thermal efficiency
(3) Indicated ~~ma~~ mean effective pressure (bar)
(4) volumetric efficiency of engine.

sol

$\dot{V}_s = \frac{\pi}{4} \times D^2 \times L \times K \times \frac{N}{60 \times 2}$

$= \frac{\pi}{4} \times 0.11^2 \times 0.13 \times 4 \times \frac{2250}{60 \times 2} = \cancel{0.023\ m^3/s}$

$= 0.0926\ m^3/s$

$IP = BP + FP$

$= 50 + 15 = 65\ kW$

$\eta_{mech} = \frac{B.P}{I.P} = \frac{50}{65} = 0.769 \Rightarrow \underline{76.9\%}$ (70% + good)

$(HA)/_{sec} = \dot{m}_f \times (CV)_f$

$= \frac{10.5}{3600} \times 50,000 = 145.83\ kW$

$\eta_{bth} = \frac{BP}{(HA)/_{sec}} = \frac{50}{145.83} = 0.3428 \Rightarrow 34.28\%$ [30% ± 5%]

$\eta_{ith} = \frac{I.P}{(HA)/_{sec}} = \frac{65}{145.83} = 44.57\%$

$P_1 \dot{V}_a = \dot{m} RT$

$\cancel{103} \times \dot{V}_a = \frac{300}{3600} \times 0.287 \times 288 = 0.0668\ m^3/s$

$\eta_{vol} = \frac{\dot{V}_a}{\dot{V}_s} = \frac{0.0668}{0.092} = 0.7217 \Rightarrow 72.17$

$AFR = \frac{\dot{m}_a}{\dot{m}_f} = \frac{300}{10.5} = 28.57$

Q G-10 A four stroke diesel engine has a displacement volume of 0.0259 m³ (25.9 L) the engine has a output of 950 kW at 2200 rpm determine the mean effective pressure in mega pascal.

sol

$B.P = P_m \dot{V}_s$

$950 = P_m \times 0.0259 \times \frac{2200}{60 \times 2}$

$P_m = 2000\ kPa \longrightarrow P_m = 2\ MPa.$

Testing of an I.C. Engine:

$$\eta_{bth}\uparrow = \frac{B.P\uparrow}{(HA)/sec} \sim \text{Efficiency}\uparrow$$

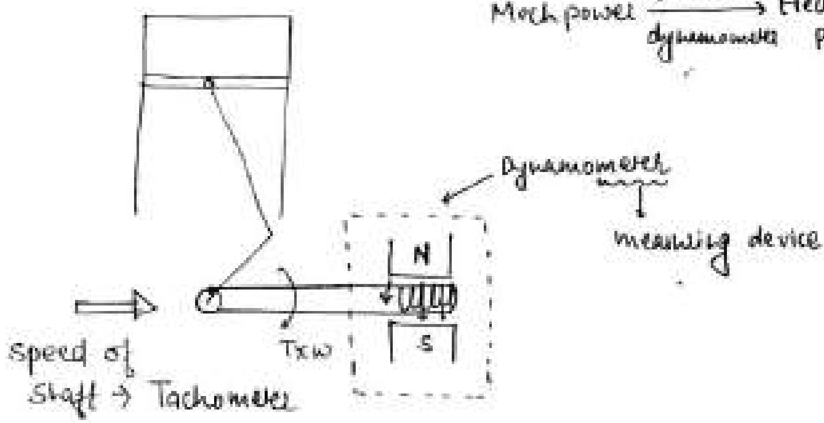
$$bsfc\uparrow = \frac{\dot{m}_f}{(B.P)\downarrow} \sim \text{Economy}\uparrow$$

$$\eta_{mech}\uparrow = \frac{(B.P)\uparrow}{I.P} \sim \text{Performance}\uparrow$$

Test-1 Dynamometer

dynamometer is basically the torque measuring device and it is used to absorb the power (brake power) during the period at which engine is tested.

$$\text{Mech power} \xrightarrow[\text{dynamometer}]{\text{generator}} \text{Electrical Power}$$



$$B.P = \boxed{T} \times \omega$$ ← Tachometer

Test 2 MOTORING TEST

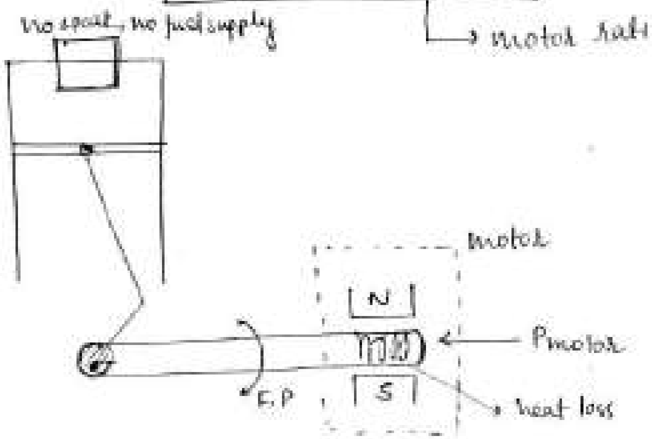
> The primary objective of motoring test is to determine frictional power of the engine but it's secondary objective is to determine performance of the engine.

2) In motoring test, first the running of engine must be stoped by making cylinder free [shorting of spark plug → S.C., cut-off the fuel supply → C.F.]

3) Then the engine shaft is coupled with the shaft of the motor and the power input to the motor is observed. if eff. of motor is given then friction power is equal to

$$\boxed{F.P. = P_{motor} \times \eta_{motor}}$$

→ motor rating

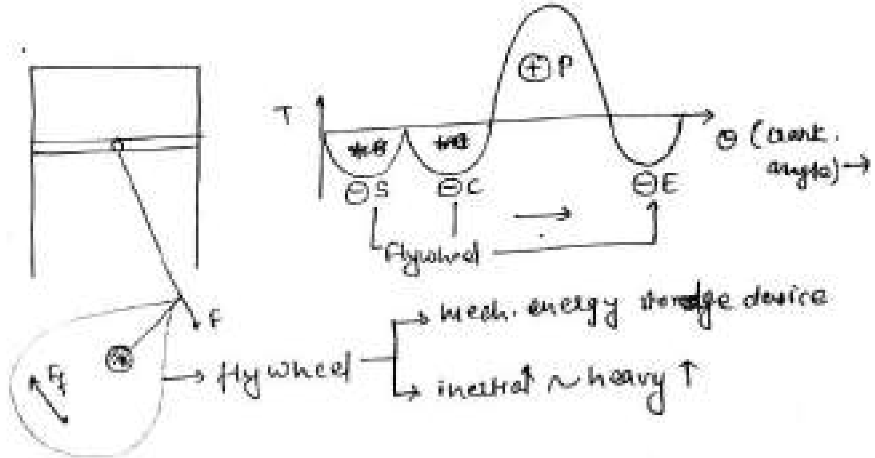


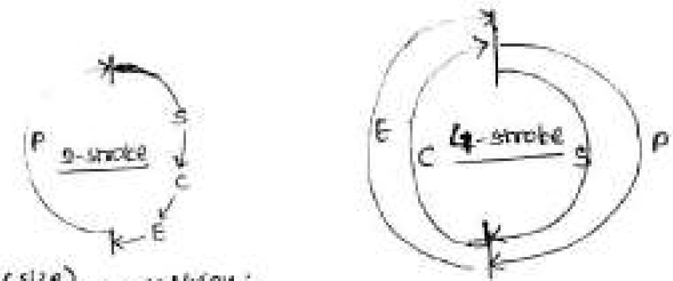
4) Overall conclusion from test ① & test ② is to determine performance of the engine

$$I.P. = B.P. + F.P.$$

$$\eta_{mech} = \frac{B.P.}{I.P.}$$

Fly wheel: It is the heavy mass or metal body which is use to store mechanical energy during power stroke in I.C. engine and excess energy is provided for the remaining suction, compression and exhaust stroke.





Note: (size) weight comparison:

$$(\text{Flywheel})_{2\text{-stroke}} < (\text{Flywheel})_{4\text{-stroke}}$$

multi-cylinder engine

1) In multicylinder engine (generally four cylinder engine) when one cylinder is giving power the other three cylinder demands power hence requirement of fly wheel decreases.

no flywheel

↳ Heavy↓ ~ [light + compact]
size↓

Note: flywheel size

4st. CI > 4st. SI > 2st. CI > 2st. SI

William's Line Method:

→ It is the graph between fuel consumption (Y-axis) and brake power (X-axis) and it is extra plotted on the ⊖ve axis of brake power to determine the friction power of engine.

→ It also tells about the amount of fuel consumption when doesn't develop any power $\dot{m}_{f,0}$

Q During the testing of an I.C engine following observation were made

<1> When engine consume 10kg/hr of the fuel then it develop the power of 100 kW

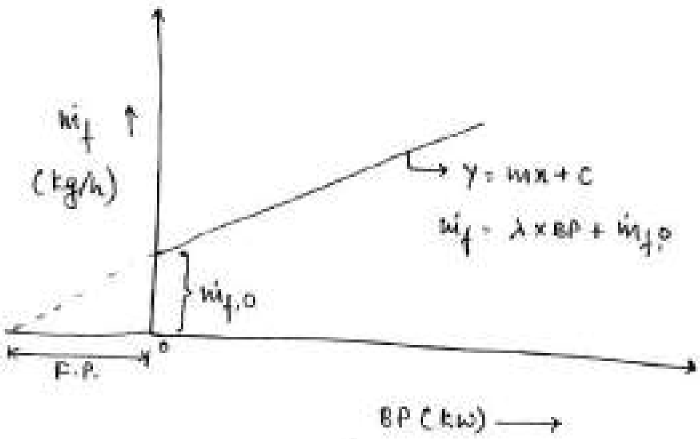
<2> When engine develops 220 kW of power then it consume only 20kg/hr of the fuel.

Determine:

<i> Amount of fuel consumption in order to overcome frictional power of the engine $\dot{m}_{f,0}$

<ii> Frictional power of the engine.

<iii> Power developed when it consume 15 kg/hr of fuel.



$10 = \lambda(100) + \dot{m}_{f,0}$

$20 = \lambda(220) + \dot{m}_{f,0}$

$\lambda = \frac{1}{12}$ $\quad \dot{m}_{f,0} = 1.67$ $\quad \Rightarrow \dot{m}_f = \frac{BP}{12} + \cancel{\dot{m}_{f,0}}\ 1.67$

<i> Amount of fuel to overcome frictional power.

$\Rightarrow \dot{m}_{f,0} = 1.67$

<ii> Frictional power of the engine.

$0 = \frac{BP}{12} + \cancel{\dot{m}_f}\ 1.67$

$B.P = -20$ kW

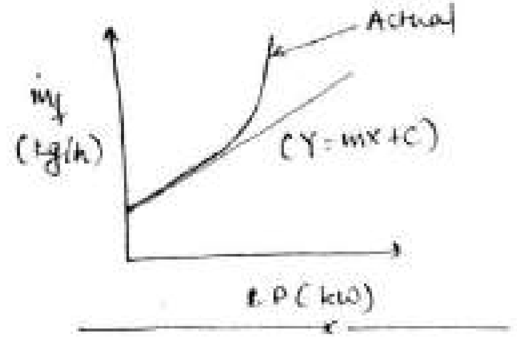
frictional power = 20 kW [Power never be negative]

<iii> $15 = \frac{BP}{12} + 1.67$

$BP = 13.33 \times 12$

$BP = 159.96$ kW

Actual graph



Q G 04 During a morse key test on a 4-cylinder engine the following measurement of brake power were taken determine the mechanical efficiency of the engine

All cyl fired = 3037
cyl 1 not fired = 2102 → 935 kW = IP_1
cyl 2 not fired = 2102 → 935 kW = IP_2
cyl 3 not fired = 2100 → 937 kW = IP_3
cyl 4 not fired = 2098 → 939 kW = IP_4

Sol

$$IP = IP_1 + IP_2 + IP_3 + IP_4 = 3746 \text{ kW}$$

$$\eta_{mech} = \frac{BP}{IP} = \frac{3037}{3746} = 81.04\%$$

Morse-key Test

1) It is use to obtain indicated power of the multicylinder engine without any pressure measuring device.

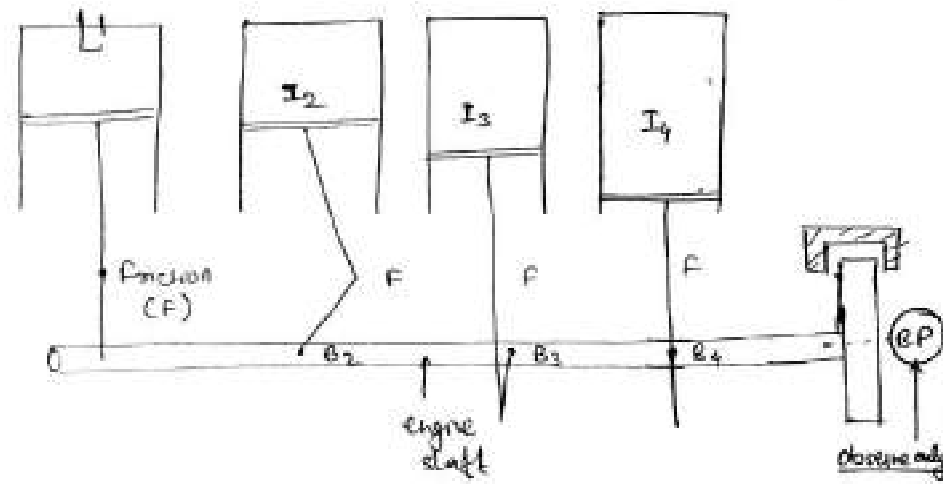
2) During the testing the only observation was taken was B.P. for different working condition.

3) During this experiment one by one cylinder were made free and the following observation were made

4) Test is applicable for S.I and C.I both

5) **Note**

M.K.T is only applicable to multi-cylinder engine. and minimum no. of cylinders are $\underline{2}$



All cyl fired ⟶ 4B = B.P

cyl 1 not fired ⟶ $3B_1$
cyl 2 not fired ⟶ $3B_2$
cyl 3 not fired ⟶ $3B_3$
cyl 4 not fired ⟶ $3B_4$

$$3B_m = \frac{3B_1 + 3B_2 + 3B_3 + 3B_4}{4}$$

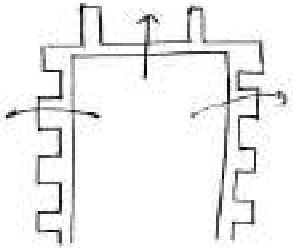
= mean 3-cyl brake power.

$$4B + 4F = 4I$$
$$3B_m + 4F = 3I_m$$
$$\overline{4(4B - 3B_m) = 4I_m = I.P}$$

Note

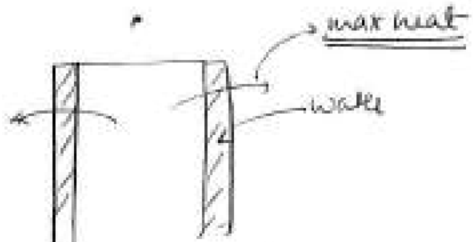
$$4B + 4F = 4I$$
$$3B_1 + 4F = 3I_1$$
$$\overline{4B - 3B_1 = I_1}$$ = difference in reading of B.P

Surface Area of cooling



Air cooled

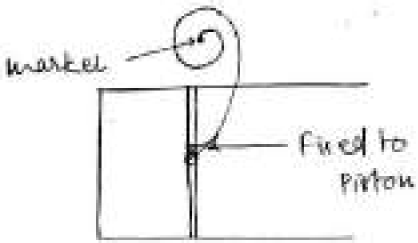
$$(A)_{HT} = \pi DL + \frac{\pi}{4} \times D^2$$

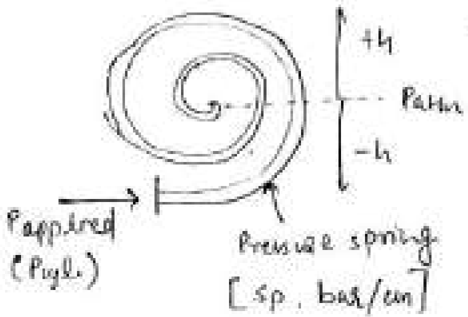


water cooled

$$(A)_{HT} = \pi DL$$

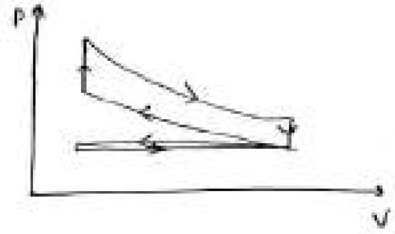
Experimental setup :





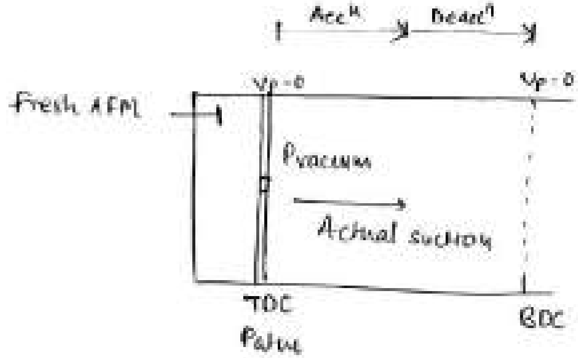
Actual PV diagram :

As per otto PV diagram



1) Actual suction.

In actual, suction will take place only if the pressure inside the cylinder is less than atmospheric so that fresh AFM will take into the cylinder.



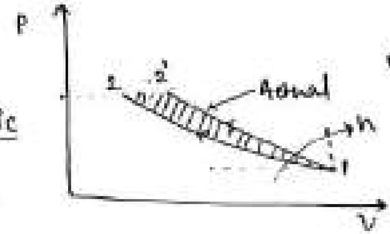
2) Actual compression

In actual, during compression temperature will rise and due to the temperature difference there will a heat loss to the water jacket hence more compression work should be provided.

not adiabatic

i.e. polytropic

$(n > \gamma)$



W_{extra} ~ compensate heat loss to water jacket

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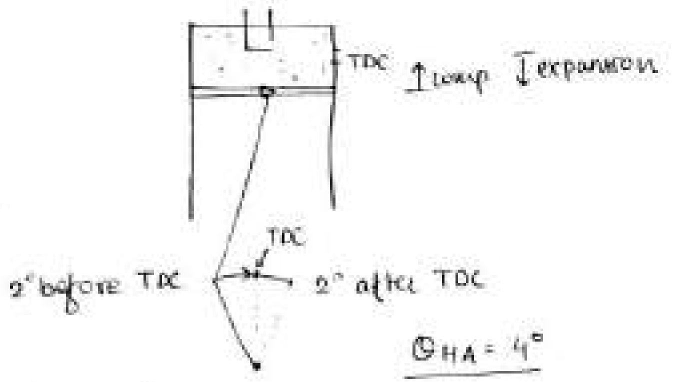
Actual Heat addition

In actual, HA will begin before TDC and ends at after TDC but for a very small crank movement



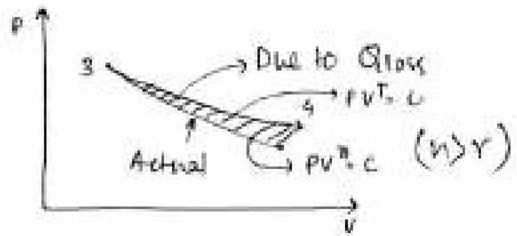
$\theta_{HA} = 4°$

$\theta = \omega t$

$$t = \frac{\theta}{\omega} = \frac{4° \times \frac{\pi}{180}}{\frac{2 \times \pi \times 2400}{60}} = \frac{1}{3600} \text{ sec} \longrightarrow 0 \text{ (very less negligible)}$$

As the crank movement is very small it appears like that there is no change in volume [V = const]

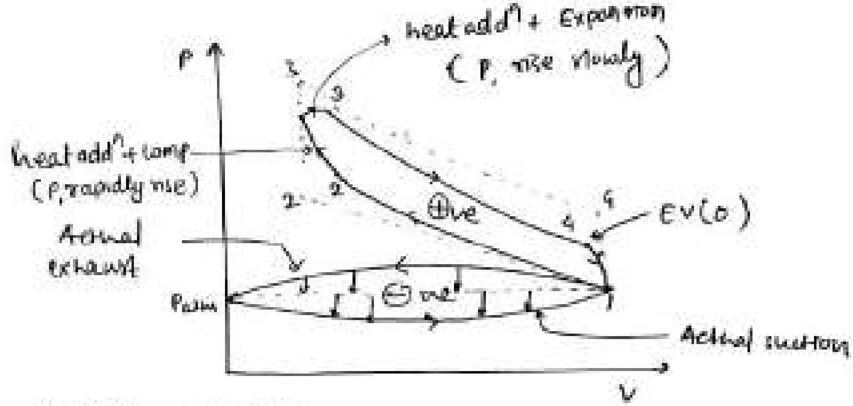
Actual Expansion:



Actual Exhaust

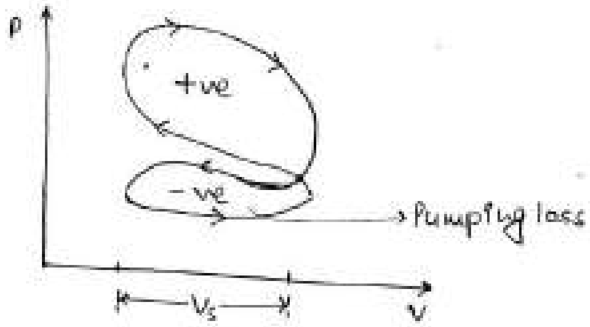
In actual exhaust will gone out of the cylinder only if the Pressure inside the cylinder is more than Patm

Actual PV Diagram



Overall conclusion:

1) In actual PV diagram there are two areas one is ⊕ve and one is ⊖ve which is also termed as Pumping loss [the amount of work done required for the actual exhaust and suction stroke]

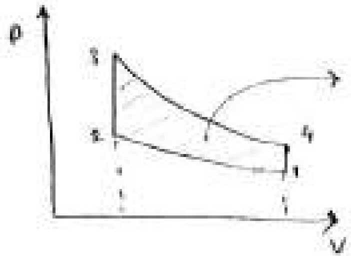


Net area = +ve − [Pumping loss] = $P_m \times V_s$ = (WD)/sec = I.P

2) Diagram factor:

It is the ratio of actual indicated power to the theoritical indicated power

$$D.F. = \frac{I.P}{(I.P)_t}$$



$(WD)_{theoretical} = (I.P)_{theoretical}$

$= (I.P)_t$

Energy Transformation Chart

$(HA)/_{sec} = \dot{m}_f \times (CV)_f \xrightarrow{(HR)_v} (I.P)_t \xrightarrow{Q_{loss} + \text{Pump loss}} I.P = P_m \times \dot{V}_s$ (Piston, reciprocating)

$\eta_{otto} = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \frac{(\alpha \rho^{\gamma} - 1)}{(\alpha - 1) + \alpha\gamma(\rho - 1)} \sim (50\% \pm)$

$\eta_{mech} = \frac{I.P}{B.P}$ (70% +)

Control ← F.P

$\eta_{ith} = \frac{I.P}{(HA)/_{sec}} \sim (40\% \pm)$

B.P = T × ω

(Rotary, shaft)

$\underbrace{T}_{} \times \omega$

$\underbrace{F_b}_{} \times r$ ← drum radius

→ Arm length (dynamometer)

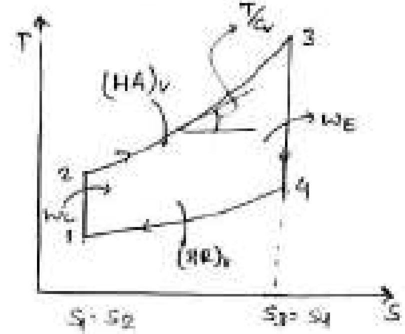
kN

$m_b \times g$

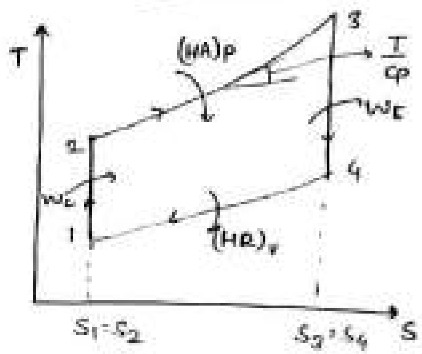
$\eta_{engine} = \eta_{overall} = \eta_{bth} = \frac{B.P}{(HA)/_{sec}} \sim (30\% \pm)$

T-S Diagram

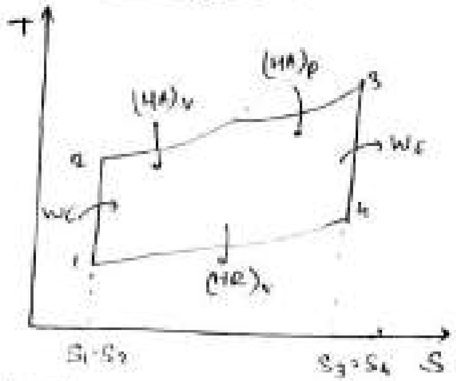
otto cycle



diesel cycle



Dual cycle



$$\left\{ \left(\frac{dT}{ds}\right)_P = \frac{T}{C_P} \right\}_{T\cdot C} < \left\{ \left(\frac{dT}{ds}\right)_V = \frac{T}{C_V} \right\}_{T\cdot C}$$

Q [C-9] In an air standard otto cycle the C.R. is 10 the condition at the beginning at the compression process is 100 kPa and 27°C the heat added at const volume is 1500 kJ per kg while the 700 kJ/kg of heat is rejected during the other const volume process. the sp. const of air 0.287 kJ/kg K. determine the mean effective pressure of cycle in bar.

Sol

$(HA)_v = 1500\ kJ/kg$

$(HR)_v = 700\ kJ/kg$ $\Big\}$ $(WD)_{net} = 800\ kJ/kg$

$$800\ kJ/kg = P_m (v_1 - v_2)$$

$$\left[v_1 = \cancel{RT}\ \frac{RT_1}{P_1} = 0.861\ m^3/kg\right]$$

$$800 = P_m \left(0.861 - \frac{0.861}{10}\right)$$

$$P_m = 1032.39\ kPa$$

$$P_m = 10.324\ bar$$

Also determine the adiabatic index

$$\eta_0 = 1 - \left(\frac{1}{r}\right)^{\gamma-1} = \frac{(WD)_{net}}{HA}$$

$$= \frac{800}{1500} = 0.533$$

$$\left(\frac{1}{r}\right)^{\gamma-1} = 0.466$$

$$\gamma-1\ \ln\frac{1}{10} = \ln 0.466$$

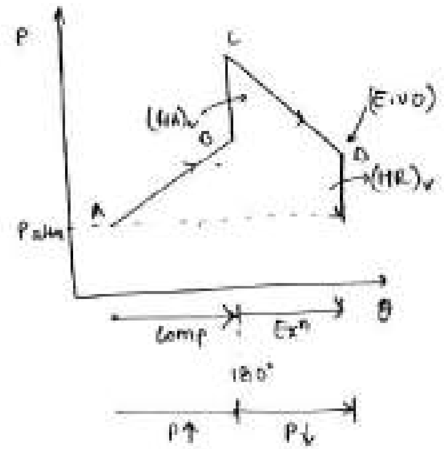
$$\gamma = 1.33$$

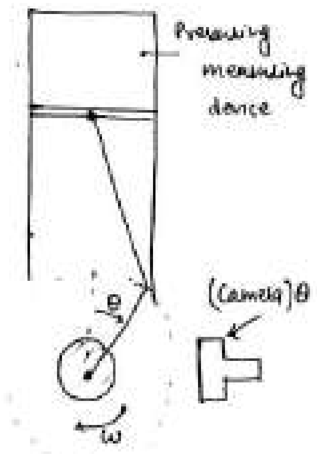
Note: whenever energies are give then η is only based on energy principle.

Combustion (only for ESE)

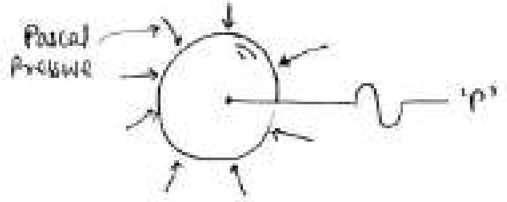
S.I. Engine

Theoritical P-θ diagram





experimental setup



Pressure measuring device

1st stage of combustion (a ⟶ b)

1) When sparking begins(a) spark from the spark plug provide energy nearest fuel particle and due to which it increases it's temperature due to chemical action fig (1.1)

2) Due to increment of nearest temp of nearest fuel particle it transfer heat to the next fuel particle and there motion of heat transfer before ignition is termed as Propagation of Nucleus fig (1.2, 1.3)

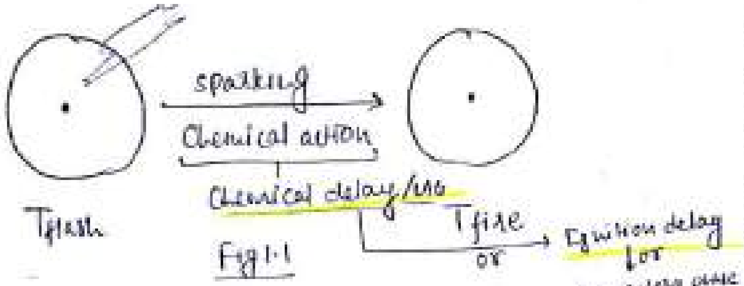


Fig 1.1

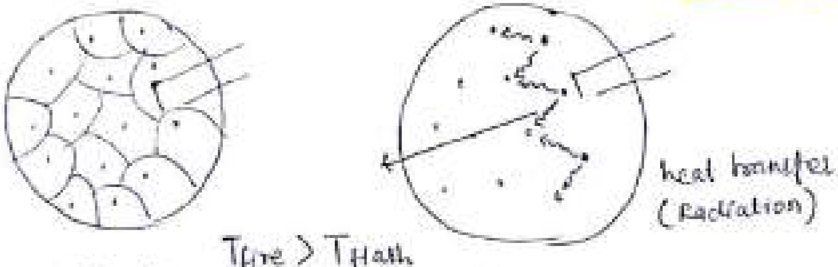


Fig 1.3

Fig 1.2

Propagation of Nucleus

3) When the temp. is equal to or more than self-ignition temp. then at the end of chemical action it release thermal energy. In a flame front will be formed (b) fig (2.1)

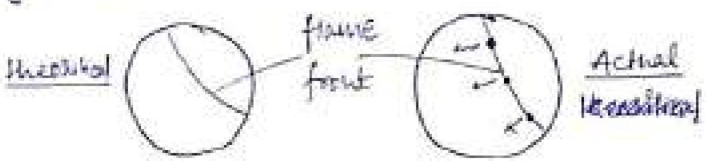


fig 2.1

When the first flame front will be formed it is termed end of first stage of combustion

Note

Time required for the completion of chemical action is termed as Chemical Delay and this is the only time required for ignition begins it means ignition delay

[time between spark begin & ignition begin]

For the S.I engine

Chemical Delay = Ignition Delay = Preparatory Phase

(a) →

Fig.1 Top view

Fig.2

Flame front : The line joining the ignited fuel particle at same instant

Propagation of flame front:

1) After the formation of first flame front, the heat from this flame front ignites the next flame front and this motion of flame front from the spark plug to the other end is termed as propogation of flame front

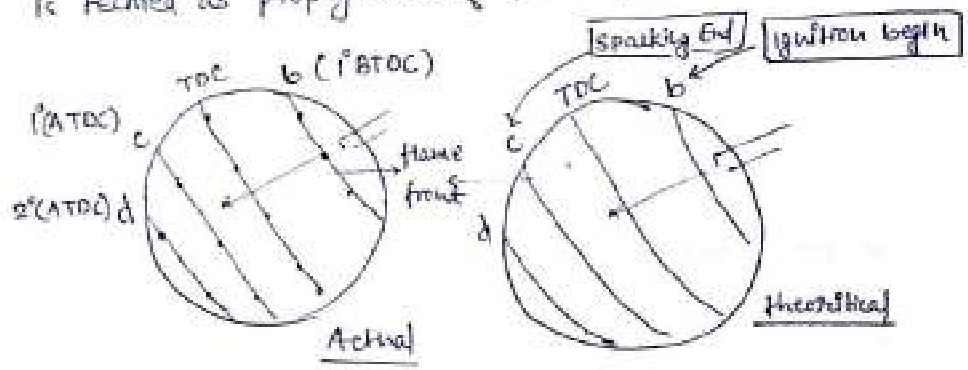
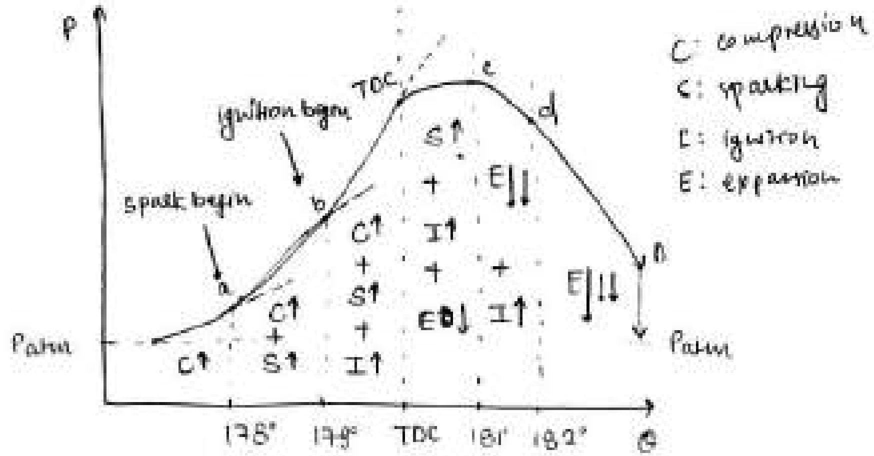


Fig.2



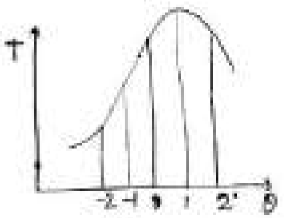
Sol:

N = 2400 rpm

$$t_{HA} = \frac{\theta_{HA}}{\omega} = \frac{1}{3600}\ \text{sec} = \frac{4\times\frac{\pi}{180}}{2\pi\times 2400/60}$$

$$\frac{1\times\pi/180}{2\pi\times\frac{2400}{60}},\quad t_{ab} = \frac{1}{14400}\ \text{sec} = C.D = I.D = \text{Preparatory Phase}$$

① An engine is running at 2400 rpm and P-θ diagram is given determine the time required for heat addition and also the chemical delay



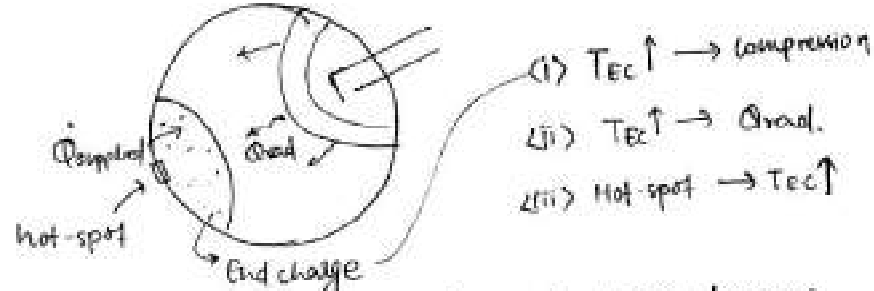
Operation during flame-front motion

1) Due to expansion near spark plug flame front air particles there is compression of end charge which increases $T_{E.C}$

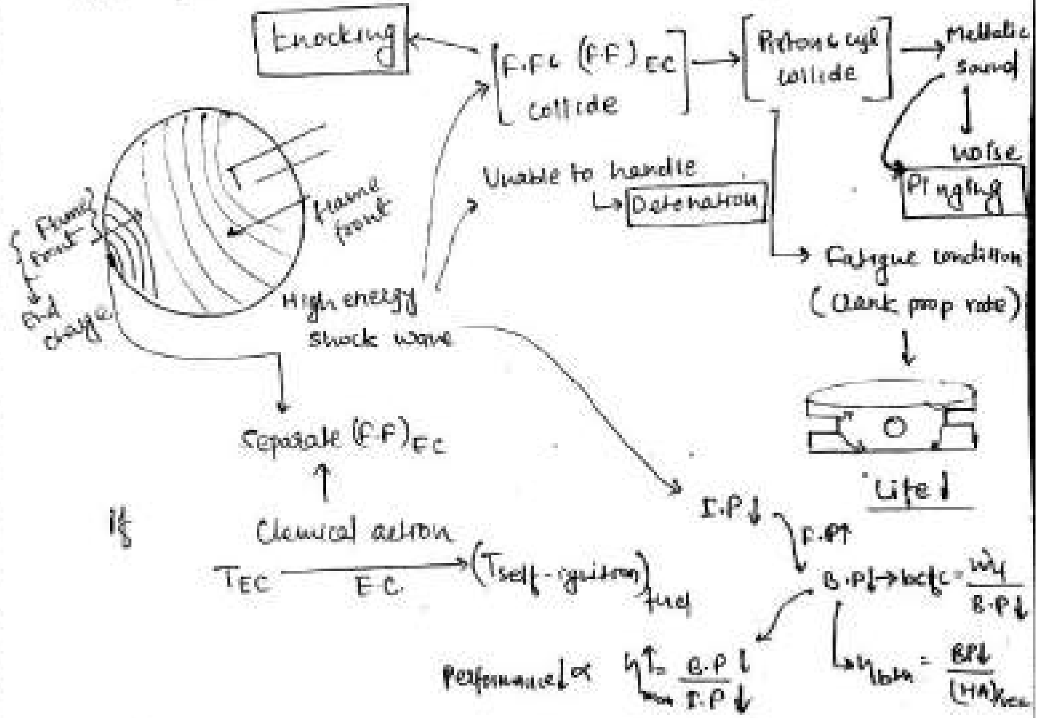
2) End Charge: The A.F.M at the other end of spark plug

3) Due to radiation heat transfer $(T)_{E.C}$ increases further

4) If there is any HOT-SPOT then $T_{E.C}$ increases



Due to above reasons if temp of end charge becomes equal to or above itself ignition temperature then a separate flame-front will form. and collision of these two flame-front is termed as knocking or Detonation



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Factors affecting knocking

if $\boxed{T_{EC}\uparrow \xrightarrow[(CD)_{EC}\downarrow]{\text{Chemical - Action}} (T_{self\text{-}ignition})}$

charge at heat mil ...

1) Compression ratio

By increasing C.R in S.I engine the temp of End charge will increase and that increases the tendency of detonation but theoretically cycle efficiency will increase.

$r_c \approx 6 \text{ to } 12 \Rightarrow \frac{T_2}{T_1} = (r)^{\gamma-1} \Rightarrow T_{AFM}\uparrow \sim T_{EC}\uparrow \Rightarrow \text{knocking} \Rightarrow B.P.\downarrow$

$\eta_o \; 1 - \left[\frac{1}{r}\right]^{\gamma-1}$ (theoretical)

$\eta_{bth}\downarrow - \eta_{engine}\downarrow$

[Due to knocking engine efficiency reduced by 50%]

2) $\text{Load}\uparrow \sim \text{Torque}\uparrow \Rightarrow (HA)\uparrow \Rightarrow T_{cy}\uparrow \sim T_{EC}\uparrow \Rightarrow \text{knocking}\uparrow$

$\dot{m}_f\uparrow$

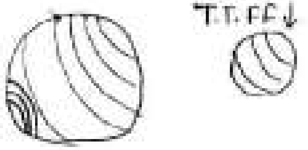
3) $\omega\uparrow \Rightarrow \text{Turbulence}\uparrow \rightarrow q_{convection}\uparrow \Rightarrow T_{H.S.}\downarrow \Rightarrow \text{knocking}\downarrow$

$\hookrightarrow h\uparrow$

Fresh AFM

4) Distance of flame travel$\downarrow \Rightarrow$ Time taken by flame front $\downarrow$

T.T.FF $\downarrow$



Diameter of cylinder = 12 cm

$D.OFT\uparrow \Rightarrow T.T.FF\uparrow > (CD)_{EC} \Rightarrow \text{knocking}\uparrow$

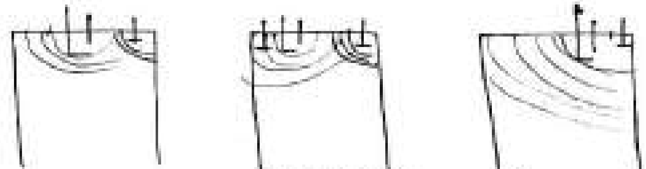
Note

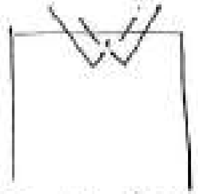
larger the distance of flame travel, then it will allow more time for the chemical delay of End charge and that may increase the knocking due to this $(D)_{cyl}$ is restricted to 12 cm $\rightarrow$ concept of multi-cylinder

$V_s = \frac{\pi}{4} \times D^2\downarrow \times L \times k\uparrow \times \frac{N}{60\times 2}$

(11) Using a fuel of long straight chain $\rightarrow$ knocking$\uparrow$

c) location of spark plug





$\Rightarrow$ Twin or dual spark plug

6) $T_{inlet\ Air}\uparrow \Rightarrow T_{AFM}\uparrow \sim T_{EC}\uparrow \Rightarrow \text{knocking}\uparrow \rightarrow \eta_{engine}\downarrow$

7) $T_{cy}\uparrow \Rightarrow T_{hot\ spot}\uparrow \Rightarrow T_{EC}\uparrow \Rightarrow (CD)_{EC}\downarrow \Rightarrow \text{knocking}\uparrow$

8) $T_{inlet\ value}\uparrow \Rightarrow T_{AFM}\uparrow \Rightarrow T_{EC}\uparrow \Rightarrow \text{knocking}\uparrow \Rightarrow \eta_{bth}\downarrow$

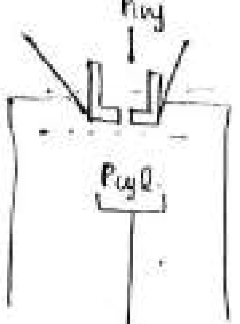
9) spark retard / ignition delay $\uparrow \Rightarrow$ Knocking$\downarrow$

10) 10% rich mixture $\Rightarrow$ knocking

Ignition Delay ~~Knocking~~ in Diesel engine

Note For better atomisation the opening should be rough and pressure difference ($P_{inj} - P_{cyl}$) is high which will decrease the physical delay. It means ignition delay will decrease.

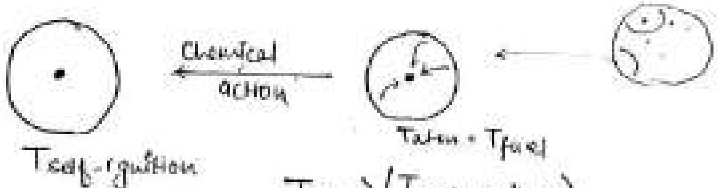
$C.I = (S.I.)^{-1}$



⑥ Rough opening of orifice
0.5 - 0.25 mm

$\Delta P \uparrow = P_{inj} - P_{cyl}$

→ Atomisation $\xrightarrow{(SA)\uparrow}$ Vaporisation $(l \to v)$ → A.F.M (Uniform - dist.)



$T_{air} \geq (T_{self\text{-}ignition})_{diesel}$

$\downarrow$ Ignition Delay = $\downarrow$ Physical Delay + Chemical Delay $\downarrow$

→ T_{air} ← $r_d \uparrow \propto$ 16 to 22

T_{air} ← $\dot{m}_f \uparrow \Leftarrow HA \uparrow \Rightarrow T_{cy} \uparrow$

$T \uparrow$

AFR at different situation

1. cold starting - 2 to 3
2. idling condition - 5 to 9
   (low load or no load)
3. High speed or acc. condⁿ - 12 to 13
4. Stoichimetric - 13.4 to 15
5. Cruising speed - 16
6. ... - 17 to 21

Q. A rev. thermodynamic cycle containing only three

P_{TDC}

⇒ Knocking in C.I. engine
→ occurs just after combustion
→ Occurs due to autoignition of first charge
→ Can be minimise to inject the fuel just before TDC

⇒ Chemically correct fuel-air ratio for SI engine mixture should be 10% rich to the stoichiometric mixture