

-: HAND WRITTEN NOTES:-
OF 166

MECHANICAL ENGINEERING

1

-: SUBJECT:-

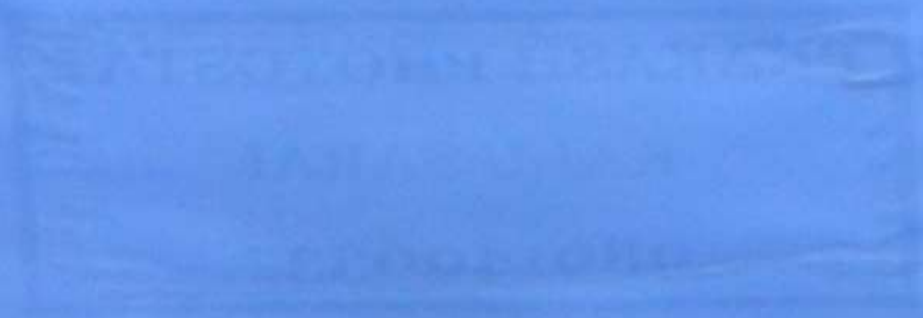
INTERNAL COMBUSTION ENGINE



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WWW.GATENOTES.IN



Engine Nomenclature :-

(3)

Top Dead centre (TDC)

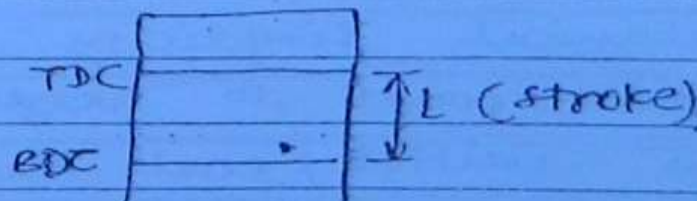
It is the dead centre when piston is nearest to Farthest from the crankshaft (It is equal to I.D.C. in case of horizontal engines)

Bottom dead centre (BDC)

It is the dead centre when piston is nearest from the crankshaft (it is equal to O.D.C. in case of horizontal engines)

Stroke (L)

The distance b/w two dead centres is known as stroke.



Displacement Volume (or) Swept Volume (V_s)

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

D = Bore (or) inner dia. of cylinder

- Introduction -

Engine :-

Engine is a device which converts one form of energy into the other useful form.

④

Heat Engines $\rightarrow$ C.E $\rightarrow$ H.E. $\rightarrow$ mech. work

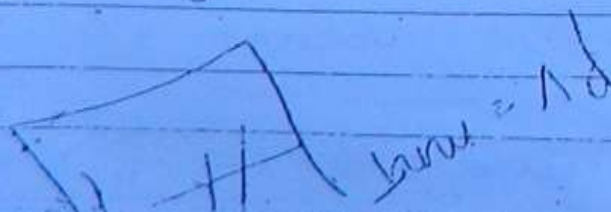
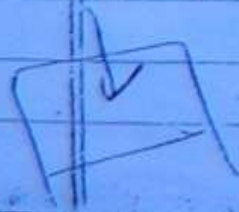
Engines are broadly classified as external Combustion engines & Internal Combustion engines.

1. External Combustion Engine
2. Internal Combustion Engine

In External combustion engines the product of combustion transfer heat to working fluid whereas in I.C engine the product of combustion produce power directly in the same cylinder.

Advantages of I.C. Engines :-

1. Mechanical simplicity
2. Higher power to weight ratio.
3. Low initial cost due to the absence of boiler, condenser etc.
4. Higher efficiency.



hvac = 1d

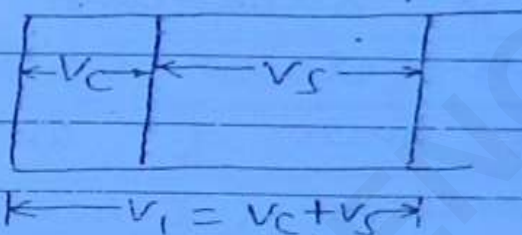
Clearance volume (V_c)

It is the volume of the cylinder when the piston is at TDC or IDC.

⑤

Compression ratio (CR or γ)

It is defined as the volume before compression to the volume after compression.



$$\gamma = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$$

$$\boxed{\gamma = 1 + \frac{V_s}{V_c}}$$

~~$$\gamma = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$$~~

~~$$\gamma = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$$~~

~~$$\gamma = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$$~~

Air Standard Cycles

(or) (Ideal I.C. engine cycles)

(or) (const. volume cycle)

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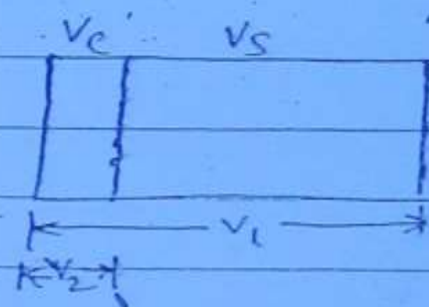
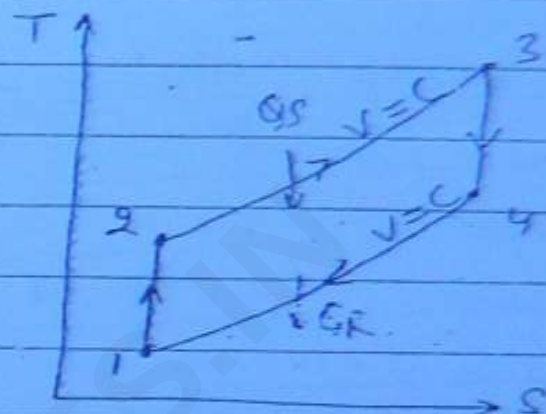
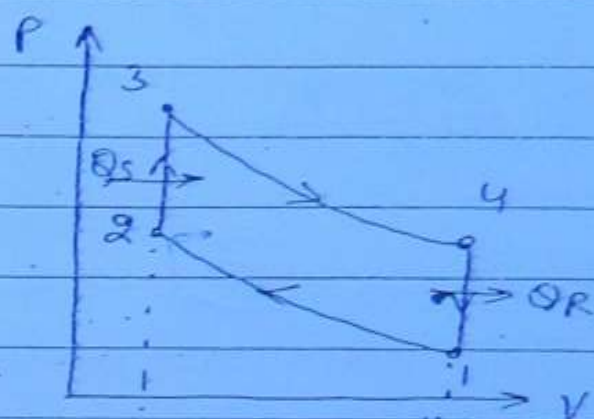
Assumptions made in Air Standard Cycles:-

1. The working substance is air and it behaves as an ideal gas.
2. The working substance is of fixed mass. (closed system analysis)
3. The specific heats of working fluid remains constant. (γ remains constant)
4. The working fluid does not undergo any ~~chem~~ chemical change.
5. All the processes are reversible processes.

Otto Cycle

(or) (Const. volume cycle)

(7)



1-2

Adiabatic Compression

2-3

Const. vol. heat addition (Q_s)

3-4

Adiabatic Expansion

4-1

Const. vol. heat rejection (Q_R)

*

clockwise $\rightarrow$ cycle is power producing cycle

*

Anticlockwise $\rightarrow$ cycle is power consuming cycle.

Note: Swept volume $V_s = V_1 - V_2$

Air standard efficiency (or) Ideal effⁿ of Otto cycle:-

$$\eta = \frac{\text{output}}{\text{input}} = \frac{W}{Q_S} = \frac{Q_S - Q_R}{Q_S}$$

$$\eta = 1 - \frac{Q_R}{Q_S}$$

(8)

$$Q_S = m C_V (T_3 - T_2)$$

$$Q_R = m C_V (T_4 - T_1)$$

$$\eta = 1 - \frac{m C_V (T_4 - T_1)}{m C_V (T_3 - T_2)}$$

$$\eta = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

$$\eta = 1 - \frac{T_1 \left(\frac{T_4}{T_1} - 1 \right)}{T_2 \left(\frac{T_3}{T_2} - 1 \right)}$$

1-2 (Adiabatic)

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$$

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

3-4 (adiabatic)

$$T_3 V_3^{\gamma-1} = T_4 V_4^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3} \right)^{\gamma-1} = \left(\frac{V_1}{V_2} \right)^{\gamma-1} = (r)^{\gamma-1} \quad \text{--- (2)}$$

∴ from ① and ② we have

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$\boxed{T_1 T_3 = T_2 T_4}$$

⑨

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

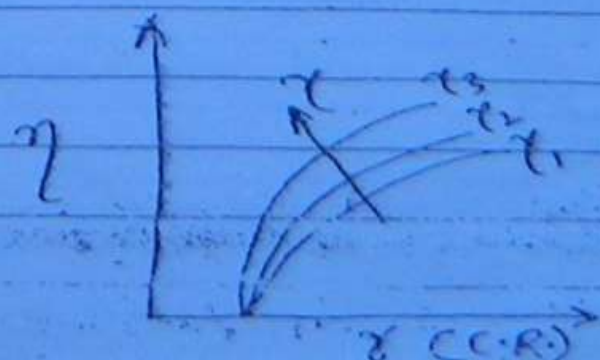
$$\eta = 1 - \frac{T_1}{T_2}$$

$$\eta = 1 - \frac{1}{(T_3/T_1)}$$

$$\boxed{\eta = 1 - \frac{1}{(\gamma)^{\gamma-1}}}$$

∴ The efficiency of Otto cycles is depend upon compression ratio (γ) and adiabatic index (γ).

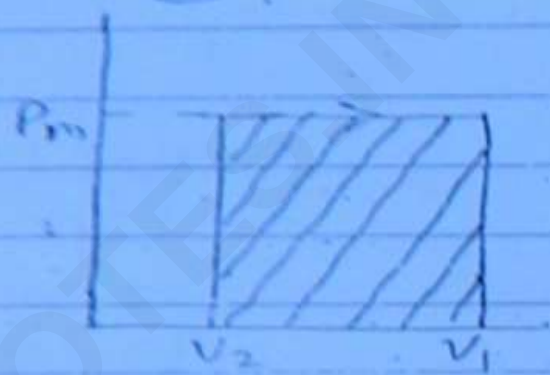
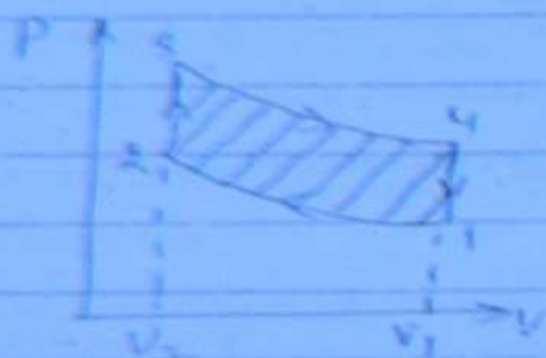
efficiency vs compression ratio (η vs γ)



mean effective pressure (mep)

is the constant pressure (or hypothetical pressure) during expansion producing same work as the actual cycle.

(10)



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

$$P_m = \frac{w_{net}}{V_1 - V_2}$$

$$P_m = \frac{w_{net}}{V_s}$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

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

$$mep = \frac{w_{net}}{V_1 - V_2}$$

$$r = \frac{V_1}{V_2} \quad V_s = V_1 - V_2$$

$$\eta = 1 - \frac{1}{(10)^{1.4-1}} = 0.601$$

$$\eta = \frac{W}{Q_c}$$

(12)

$$0.601 = \frac{100}{Q_c}$$

$$Q_c = 166.14 \text{ kW}$$

In an engine working on ideal Otto cycle the compression ratio of ($r = 5.5$) the work output per cycle is $23.625 \times 10^5 V_c$ Joule where V_c = clearance volume in m^3 then the mean effective pressure is

$$\eta = 1 - \frac{1}{(5.5)^{1.4-1}} =$$

$$W = 23.625 \times 10^5 V_c$$

$$mep = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_s}$$

$$mep = \frac{23.625 \times 10^5 V_2}{V_1 - V_2}$$

$$= \frac{23.625 \times 10^5 V_2}{V_2 \left(\frac{V_1}{V_2} - 1 \right)}$$

$$\frac{23.625 \times 10^5}{\frac{V_1}{V_2} - 1} = \frac{23.625 \times 10^5}{5.5 - 1}$$

$$= 5.25 \times 10^5 \text{ N/m}^2$$

$$= 5.25 \text{ bar}$$

Engine

- (5) An A working on air standard Otto cycle has a cylinder dia. of 10 cm and stroke length of 15 cm. If the clearance volume of 196.3 cm³ and heat supply per kg air is 1800 kJ/kg (90). then the work output is.

Solⁿ:-

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

$$= \frac{\pi}{4} \times (10)^2 \times 15$$

$$=$$

$$V_c = 196.3 \text{ cm}^3$$

$$V_1 = V_s + V_c$$

$$=$$

$$V_2 =$$

$$\eta = \frac{W}{Q_s}$$

$$\eta = 1 - \frac{1}{(\gamma)^{\frac{1}{\gamma-1}}}$$

$$\gamma = 1 + \frac{V_s}{V_c}$$

$$\gamma = 1 + \frac{\frac{\pi}{4} D^2 L}{V_c}$$

$$\gamma = 1 + \frac{\frac{\pi}{4} (10)^2 \times 15 \text{ cm}^3}{196.3 \text{ cm}^3} = \gamma$$

$$\eta = 1 - \frac{1}{(\gamma)^{\frac{1}{\gamma-1}}} = \frac{1}{(\gamma)^{\frac{1}{\gamma-1}}}$$

$$\eta = \frac{W}{Q_s}$$

$$W = \eta \times Q_s$$

$$= 0.541 \times 1800$$

(14)

$$W = 973 \text{ KJ/Kg}$$

The mean effective pressure of an Otto cycle can be expressed as

a. $\frac{\Delta P}{(\gamma-1)(\gamma-1)}$

where η_{th} = Thermal effⁿ.

b. $\frac{\eta_{th} (\Delta P)}{(\gamma-1)(\gamma-1)}$

ΔP = pressure rise during heat addition

c. $\frac{\eta_{th} (\Delta P)}{\gamma(\gamma-1)}$

γ = pressure ratio

d. $\frac{\eta_{th} (\Delta P)}{(\gamma-1) \gamma}$

γ = adiabatic index

$$mep = \frac{W_{net}}{V_1 - V_2}$$

$$\eta_{th} = \frac{W_{net}}{Q_s}$$

$$mep \times (V_1 - V_2) = \eta_{th} \cdot Q_s$$

$$mep = \frac{\eta_{th} \cdot Q_s}{(V_1 - V_2)}$$

$$W_{net} = \eta_{th} \times Q_s$$

(15)

$$= \eta_{th} \times m C_v (T_3 - T_2)$$

$$= \eta_{th} \times \frac{m \cdot R}{\gamma - 1} (T_3 - T_2)$$

$$W_{net} = \frac{\eta_{th} (P_3 V_3 - P_2 V_2)}{\gamma - 1}$$

$$W_{net} = \frac{\eta_{th} \times (P_3 V_2 - P_2 V_2)}{\gamma - 1}$$

$$W_{net} = \frac{\eta_{th} V_2 (P_3 - P_2)}{\gamma - 1}$$

$$W_{net} = \frac{\eta_{th} V_2 \Delta P}{\gamma - 1}$$

$$m_{ep} = \frac{W_{net}}{V_1 - V_2}$$

$$m_{ep} = \frac{\eta_{th} V_2 \Delta P}{(\gamma - 1)(V_1 - V_2)}$$

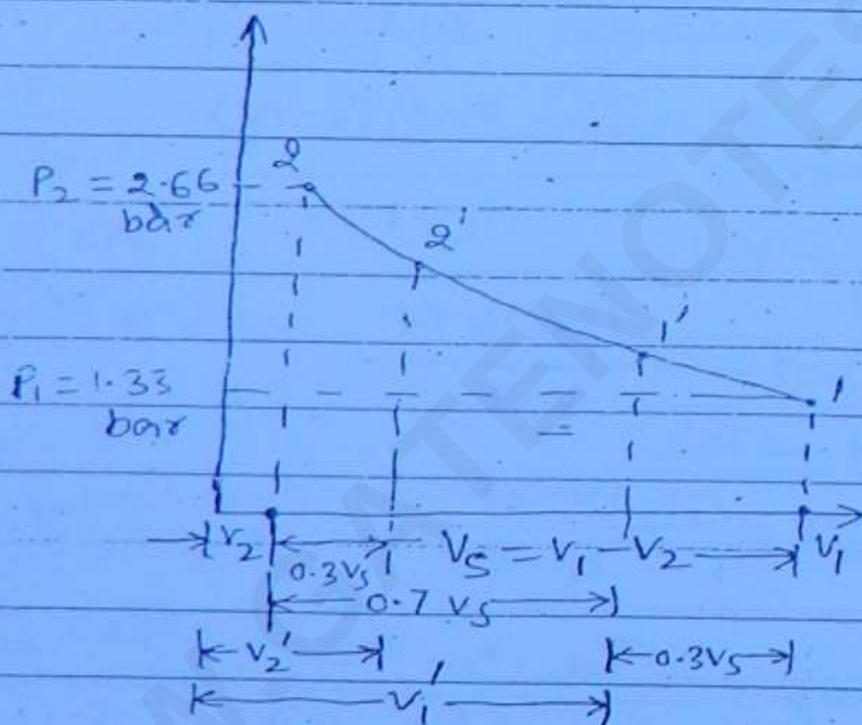
$$m_{ep} = \frac{\eta_{th} V_2 \Delta P}{(\gamma - 1) V_2 \left(\frac{V_1}{V_2} - 1 \right)}$$

$$m_{ep} = \frac{\eta_{th} \Delta P}{(\gamma - 1)(r - 1)}$$

- ⑦ An engine working on Otto cycle the pressure inside the cylinder at 30% and 70% of compression stroke is equal to 1.33 bar and 2.66 bar resp. assuming that compression follows the law $PV^{1.33} = \text{Const.}$ Find the compression ratio.

Soln:-

(16)



$$V_1' = V_2 + 0.7(V_1 - V_2)$$

$$V_2' = V_2 + 0.3(V_1 - V_2)$$

$$V_1' = V_2 \left[1 + 0.7 \left(\frac{V_1}{V_2} - 1 \right) \right]$$

$$V_1' = V_2 [1 + 0.7(r - 1)]$$

Similarly

$$V_2' = V_2 [1 + 0.3(r - 1)]$$

We know

$$P_1' V_1'^{1.33} = P_2' V_2'^{1.33}$$

$$\textcircled{17} \quad \frac{V_1'}{V_2'} = \left(\frac{P_2'}{P_1'} \right)^{\frac{1}{1.33}}$$

$$\frac{V_2' [1 + 0.7(\gamma - 1)]}{V_1' [1 + 0.3(\gamma - 1)]} = \left(\frac{2.66}{1.33} \right)^{\frac{1}{1.33}}$$

$$\gamma = 4.51$$

Q. Show that for max. work per kg of air in Otto cycle for a given upper and lower temp. of T_3 & T_1 .

The temp. at the end of compression T_2 and the end of expansion T_4 are equal and are given by

$$T_2 = T_4 = \sqrt{T_1 T_3}$$

Soln:

$$W = Q_s - Q_R$$

$$W = C_v (T_3 - T_2) - C_v (T_4 - T_1)$$

$$W = C_v [T_3 - T_2 - T_4 + T_1]$$

For 1-2

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$$

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

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

$$T_3 - T_2 = \frac{T_3 - T_1}{\gamma} + T_1$$
$$0 = 1 - \frac{1}{\gamma} \left(\frac{1}{\gamma} \right)^{\gamma-1}$$
$$\frac{1}{\gamma} = 1$$

$$T_3 V_3^{\gamma-1} = T_4 V_4^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3} \right)^{\gamma-1} = \left(\frac{V_1}{V_2} \right)^{\gamma-1}$$

$$\frac{T_3}{T_4} = (\gamma)^{\gamma-1} \quad (18)$$

$$T_4 = \frac{T_3}{(\gamma)^{\gamma-1}}$$

$$W = C_v \left[T_3 - T_1 \gamma^{\gamma-1} - \frac{T_3}{(\gamma)^{\gamma-1}} + T_1 \right]$$

Differentiate above eqn.

$$\frac{dw}{d\gamma} = 0$$

$$\frac{dw}{d\gamma} = C_v \left[0 - T_1 (\gamma-1) \gamma^{\gamma-2} - T_3 \left\{ (-\gamma+1) \gamma^{-\gamma+1} \right\} + 0 \right]$$

$$C_v \left[-T_1 (\gamma-1) \gamma^{\gamma-2} - T_3 \left\{ (-\gamma+1) \gamma^{-\gamma+1} \right\} \right] = 0$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$\frac{\sqrt{T_1 T_3}}{T_1} = \frac{T_3}{T_4}$$

$$T_4 = \frac{T_3 T_1}{\sqrt{T_1 T_3}} = \sqrt{T_1 T_3} \quad (T_2 = T_4)$$

$$T_2 = \sqrt{T_1 T_3}$$

(19)

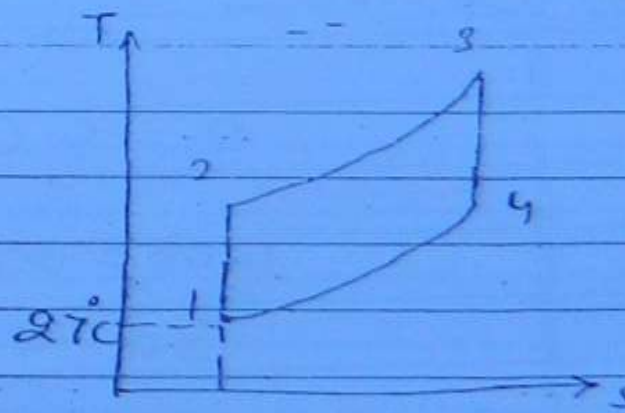
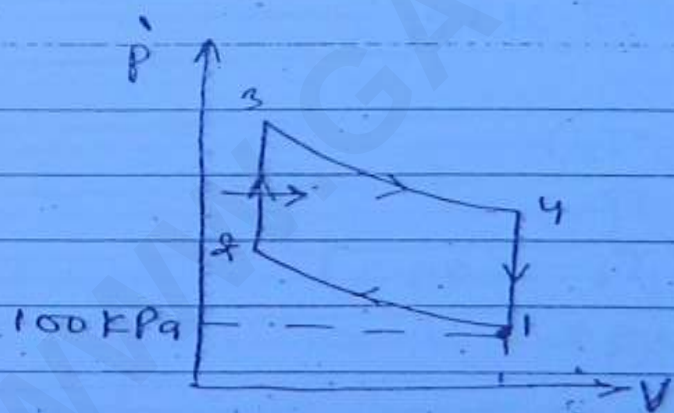
Note: For max. work output in otto cycle temp. after compression is equal to temp. after expansion.

(19)

(9)

The min. pressure & Temp. in otto cycle are 100 kPa and 27°C [which at Point 1] the amount of heat added per cycle is 1500 kJ/kg. Determine the pressure & temp. at all points & also calculate air standard efficiency and work output. Comp. ratio ($r = 8$), $C_v = 0.72 \text{ kJ/kg-K}$ & $\gamma = 1.4$

Solⁿ:



$$Q_a = 1500 \text{ kJ/kg}$$

$$P_1 = 100 \text{ kPa}$$

$$T_1 = 27 + 273 = 300 \text{ K}$$

$$P_2/P_1^{\gamma} = P_3/P_4^{\gamma}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

For process 1-2

(20)

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$$

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

$$T_2 = 300 (8)^{1.4-1}$$
$$= 689.2 \text{ K}$$

1-2 (adiabatic)

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

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

$$P_2 = 100 (8)^{1.4}$$
$$= 1837.91 \text{ kPa}$$

$$Q_5 = C_v (T_3 - T_2)$$

$$1500 = 0.72 (T_3 - 689.2)$$

$$T_3 = 2772.5 \text{ K}$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{or} \quad \left\{ \begin{array}{l} T_2 = T_3 \\ T_1 = T_4 \end{array} \right\}$$

$$T_1 T_3 = T_2 T_4$$

$$T_4 = \frac{T_1 T_3}{T_2} = \frac{300 \times 2772.5}{689.2}$$

$$T_4 = 1207.91 \text{ K}$$

Process 2-3 ($V=C$)

$$PV = mRT$$

$$P \propto T$$

(2)

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$$P_3 = P_2 \cdot \frac{T_3}{T_2} = 1837.9 \left[\frac{2772.5}{689.2} \right]$$

$$P_3 = 7395.6 \text{ KPa}$$

Process 4-1 ($V=C$)

$$\frac{P_4}{P_1} = \frac{T_4}{T_1}$$

$$P_4 = P_1 \cdot \frac{T_4}{T_1}$$

$$= 100 \times \frac{1207}{300} = 402.33 \text{ KPa}$$

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

$$= 1 - \frac{1}{(8)^{1.4-1}}$$

$$= 0.561$$

$$W = \eta \times Q_s$$

$$= 0.561 \times 1500$$

$$= 847 \text{ KJ/Kg}$$

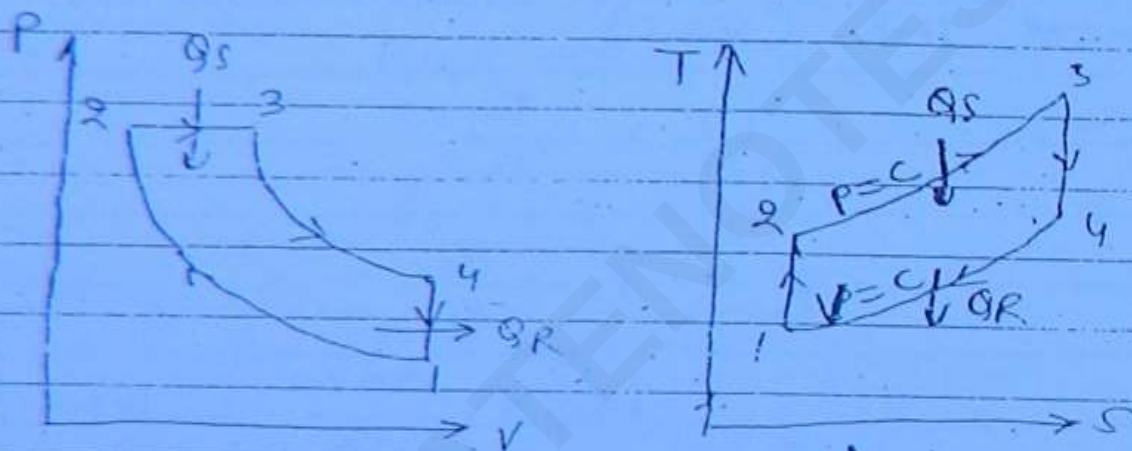
Ans:- Otto cycle is used in spark ignition (S.I.) Engine or petrol engines.

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$$\eta = 1 - \frac{Q_R}{Q_S}$$

Diesel cycle

(2) (Const. pressure cycle)



Cut-off Ratio (γ_c)

It is the ratio of volume after heat addition to the volume before heat addition.

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

Air standard eff

Air standard effⁿ of Diesel cycle: η diesel

$$\eta = 1 - \frac{Q_R}{Q_S} \quad (23)$$

$$Q_S = m C_p (T_3 - T_2)$$

$$Q_R = m C_v (T_4 - T_1)$$

$$\eta = 1 - \frac{m C_v (T_4 - T_1)}{m C_p (T_3 - T_2)}$$

$$\eta = 1 - \frac{1}{\gamma} \frac{T_1 \left(\frac{T_4}{T_1} - 1 \right)}{T_2 \left(\frac{T_3}{T_2} - 1 \right)} \quad \text{--- (1)}$$

For 1-2

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

For 2-3

$$\frac{V_3}{V_2} = \frac{T_3}{T_2} = r_c$$

For 3-4 (adiabatic)

$$T_3 V_3^{\gamma-1} = T_4 V_4^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3} \right)^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_1}{V_3} \right)^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_1}{V_2} \cdot \frac{V_2}{V_3} \right)^{\gamma-1}$$

$$\frac{T_3}{T_4} = \left(\gamma \cdot \frac{1}{r_c} \right)^{\gamma-1} \quad (24)$$

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

$$\frac{T_3}{T_4} = \frac{\gamma^{\gamma-1}}{r_c^{\gamma-1}}$$

$$\frac{T_3}{T_4} = \frac{T_2/T_1}{r_c^{\gamma-1}} \quad \therefore \frac{T_2}{T_1} = r_c^{\gamma-1}$$

$$\frac{T_3}{T_4} = \frac{T_2}{T_1} \cdot \frac{1}{r_c^{\gamma-1}}$$

$$\frac{T_3}{T_4} = \frac{T_3}{T_2} \cdot r_c^{\gamma-1}$$

$$\frac{T_4}{T_1} = r_c' \cdot r_c^{\gamma-1} = r_c^{\gamma}$$

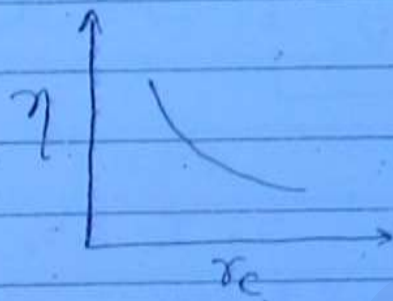
from eqn. (i)

$$\eta = 1 - \frac{1}{\gamma} \cdot \frac{1}{(r_c)^{\gamma-1}} \left[\frac{r_c^{\gamma}-1}{r_c-1} \right]$$

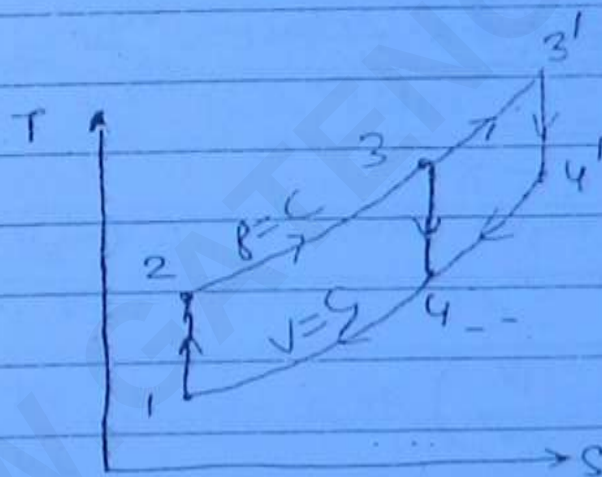
** The eff. of diesel cycle depends on
Compression ratio (r)
Cut-off ratio (r_c) and
Adiabatic index (γ)

(23)

** As the cut-off ratio increases the
effⁿ. of diesel cycle decreases.



**



With increasing in cut-off ratio the
heat rejection also increases because
heat rejection is occurring at const.
volume and the slope of const. volun
lines is more than the slope of
const. pressure lines.

② Expansion ratio (r_e)

It is defined as the ratio of volume after expansion (V_4) to the volume before expansion (V_3)

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

(26)

Relationship b/w r , r_c & r_e

$$r_c r_e = \frac{V_3}{V_2} \times \frac{V_4}{V_3} = \frac{V_4}{V_2} = \frac{V_1}{V_2}$$

$$r_c r_e = r$$

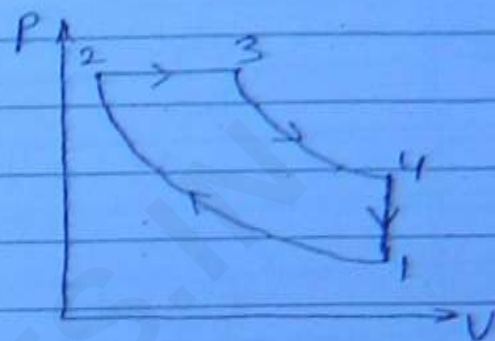
Prob. ① A diesel engine has a comp. ratio of $r = 20$, and cut-off volume is 5% of stroke volume. then find the η of air standard cycle.

Solⁿ:

(27)

* Cut-off volume = $V_3 - V_2$

$r = 20 \quad \gamma = 1.4$



$$V_3 - V_2 = 0.05 (V_1 - V_2)$$

$$V_3 - V_2 = 0.05 V_1 - 0.05 V_2$$

divide V_2 both sides

$$\frac{V_3 - V_2}{V_2} = \frac{0.05 (V_1 - V_2)}{V_2}$$

$$\frac{V_3}{V_2} - 1 = 0.05 \left(\frac{V_1}{V_2} - 1 \right)$$

$$r_c = \frac{V_3}{V_1} = 1 + 0.05 (20 - 1) = 1.95$$

$$\eta = 1 - \frac{1}{r} \cdot \frac{1}{r^{\gamma-1}} \left[\frac{r_c^{\gamma} - 1}{r_c - 1} \right]$$

$$= 1 - \frac{1}{1.4} \cdot \frac{1}{(20)^{1.4-1}} \left[\frac{(1.95)^{1.4} - 1}{1.95 - 1} \right]$$

$$= 1 - \frac{1}{1.4} \cdot \frac{1}{(20)^{1.4-1}} [1.6285]$$

$$= 0.6490$$

$$\eta = 64.90\%$$

Q2) Determine the air standard effⁿ of diesel cycle if the cylinder bore is 250 mm, Stroke length is 375 mm, and clearance volume is 1500 CC with a cut-off volume of 5% of stroke ($\gamma = 1.4$)

D=

28

$$\eta = 1 - \frac{1}{\gamma} \cdot \frac{1}{r^{\gamma-1}} \left[\frac{r_c^{\gamma} - 1}{r_c - 1} \right]$$

$$r = 1 + \frac{V_s}{V_c} = 1 + \frac{\frac{\pi}{4} D^2 L}{V_c}$$

$$r = 1 + \frac{\frac{\pi}{4} (25)^2 (37.5)}{1500} = 13.27$$

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

$$V_3 - V_2 = 0.05 (V_1 - V_2)$$

$$\frac{V_3 - V_2}{V_2} = \frac{0.05 (V_1 - V_2)}{V_2}$$

$$\frac{V_3}{V_2} - 1 = 0.05 \left(\frac{V_1}{V_2} - 1 \right)$$

$$r_c = 1 + 0.05 (13.27 - 1) = 1.6135$$

$$\eta = 1 - \frac{1}{1.4} \cdot \frac{1}{(13.27)^{1.4-1}} \left[\frac{1.6135^{1.4} - 1}{1.6135 - 1} \right]$$

$$= 0.6052$$

$$\eta = 60.52\%$$

- (3) An engine working on diesel cycle the inlet pressure & temp are 1 bar & 17°C the pressure at the end of compression is 35 bar the expansion ratio is 5. Calculate - Heat addition, heat rejection and effⁿ. of cycle ($C_p = 1.004 \text{ kJ/kg-K}$, $C_v = 0.717 \text{ kJ/kg-K}$ & $\gamma = 1.4$)

Solⁿ:-

$$P_1 = 1 \text{ bar} ; T_1 = 17 + 273 = 290 \text{ K}$$

$$P_2 = 35 \text{ bar}$$

(29)

Process 1-2 (adiabatic)

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

$$\gamma = \frac{V_1}{V_2} = \left(\frac{P_2}{P_1} \right)^{1/\gamma}$$

$$\gamma = \left(\frac{35}{1} \right)^{1/1.4} = 12.67$$

$$r_e = 5$$

$$r_c r_e = \gamma$$

$$r_c = \frac{\gamma}{r_e} = \frac{12.67}{5} = 2.53$$

$$\eta = 1 - \frac{1}{1.4} \cdot \frac{1}{(12.67)^{1.4-1}} \left[\frac{2.534^{1.4} - 1}{2.534 - 1} \right]$$

$$= 0.549$$

$$\eta = 54.9 \%$$

109-51
2-10
5

$$Q_R = C_p(T_3 - T_2)$$

1-2 (adiabatic)

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$$

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

$$T_2 = 290 (18.67)^{1.4-1} = 807.26 \text{ K}$$

2-3 (const. p process)

$$\frac{V_3}{V_2} = \frac{T_3}{T_2} = r_c = 2.51$$

$$T_3 = T_2 \times 2.51$$

$$= 807.26 \times 2.51 = 2042.36 \text{ K}$$

$$Q_3 = C_p(T_3 - T_2) = 1.004 (2042.36 - 807.26)$$

$$Q_3 = 1240 \text{ kJ/kg}$$

$$\eta = 1 - \frac{Q_R}{Q_3}$$

$$0.559 = 1 - \frac{Q_R}{1240}$$

$$Q_R = 555 \text{ kJ/kg}$$

A diesel cycle operates at a pres. of 1 bar at the beginning of compression and the volume is compressed to $1/18$ of initial volume. Heat is supplied until the volume is

twice the clearance volume. Calculate mep of the cycle ($\gamma = 1.4$).

Soln:

$$P_1 = 1 \text{ bar}$$

$$V_2 = \frac{V_1}{16}$$

$$\frac{V_1}{V_2} = 16 \quad \text{or} \quad r = 16$$

$$V_3 = 2V_2$$

$$\frac{V_3}{V_2} = 2 \quad \text{or} \quad r_c = 2$$

$$r_e = \frac{r}{r_c} = \frac{16}{2} = 8$$

$$r_e = \frac{V_4}{V_3} = 8$$

$$V_4 = 8V_3$$

$$W_{net} = W_{12} + W_{23} + W_{34} + W_{41}$$

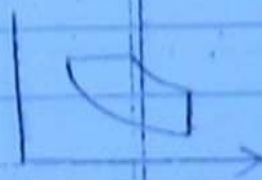
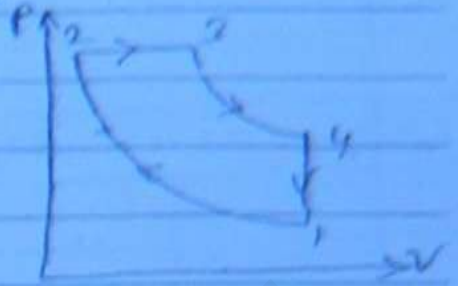
$$= \frac{P_1 V_1 - P_2 V_2}{\gamma - 1} + \frac{P_2 (V_2 - V_3) + P_3 V_3 - P_4 V_4}{\gamma - 1}$$

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

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

$$P_2 = 1 (16)^{1.4} = 48.5 \text{ bar}$$

$$P_3 = P_2 = 48.5 \text{ bar}$$



31

8-4

$$P_3 V_3^\gamma = P_4 V_4^\gamma$$

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

(22)

$$= 48.5 \left(\frac{1}{8} \right)^{1.4}$$

$$P_4 = 2.63 \text{ bar}$$

$w_{\text{net}} =$

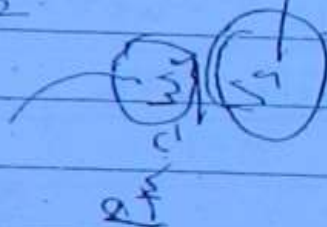
$$\frac{V_1}{V_2} = 16 \Rightarrow V_1 - V_2 = 16V_2 - V_2 = 15V_2$$

$$mep = \frac{1}{V_1 - V_2} w_{\text{net}}$$

$$= \frac{1}{15V_2} \times \left[\frac{1 \times 16V_2 - 48.5V_2}{1.4 - 1} + \frac{48.5(2V_2 - V_2)}{1.4 - 1} - \frac{48.5(2V_2) - 2.63(16V_2)}{1.4 - 1} \right]$$

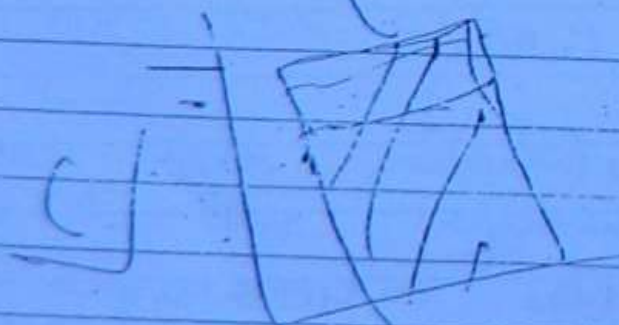
$$= \frac{1}{15V_2} \times 104.5 V_2$$

$$= 6.966 \text{ bar}$$



$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3} \right)^\gamma$$

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

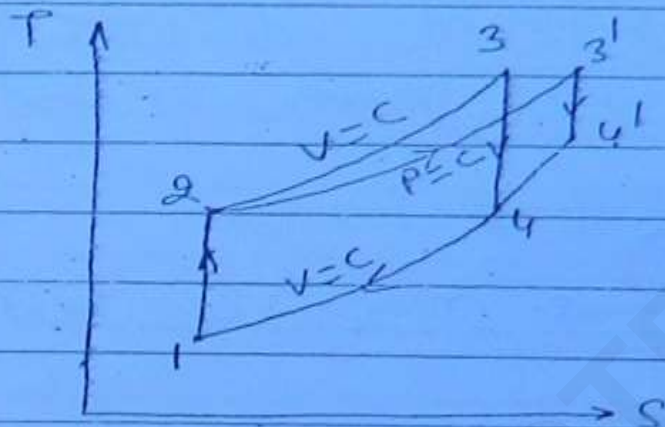


$$\eta = \frac{w}{q_p(T_3 - T_1)}$$

Comparison of Otto & diesel cycle:-

Case-I :-

Same Compression ratio and heat addition



1-2-3-4 — Otto cycle

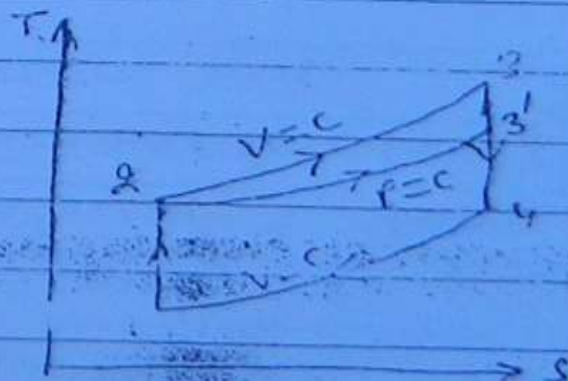
1-2-3'-4' — Diesel cycle

$$\eta = 1 - \frac{Q_R}{Q_S}$$

For same compression ratio and heat addition, heat rejection is more in diesel cycle and hence it is less efficient.

Case-II

Same comp ratio & heat rejection



1-2-3-4 — Otto cycle

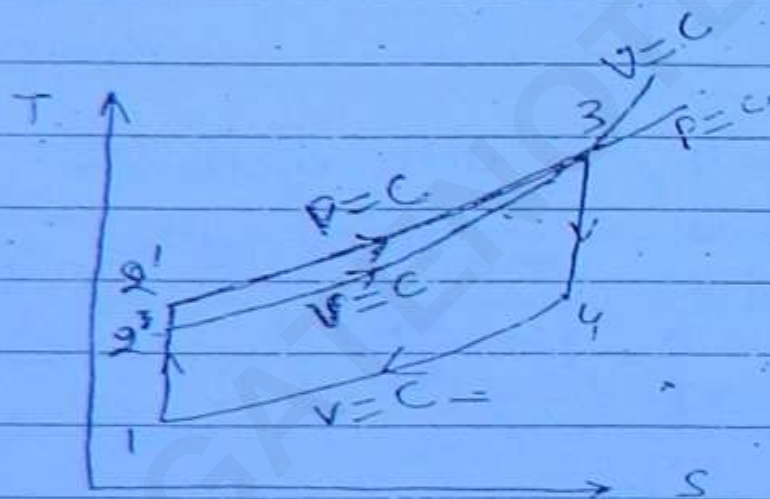
1-2'-3'-4 — Diesel cycle

(34)

For same C.R. & heat rejection
heat supplied is more in Otto
cycle and hence it is efficient.

Case III

Same max. Temp. & heat rejection

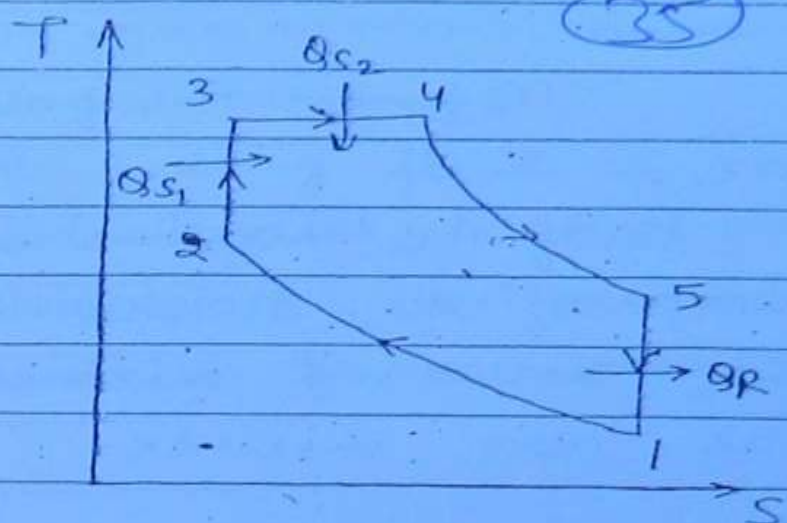


1-2-3-4 — Otto cycle

1-2'-3'-4 — Diesel cycle

For same max. temp. & heat rejection
heat supplied is more in diesel
cycle and hence it is efficient.

Dual Cycle



$$\eta = 1 - \frac{Q_R}{Q_{S1} + Q_{S2}}$$

$$Q_{S1} = m C_v (T_3 - T_2)$$

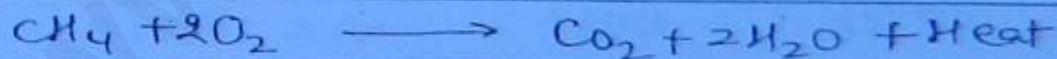
$$Q_{S2} = m C_p (T_4 - T_3)$$

Fuel - Air Cycle

chemically correct Air-fuel - ratio

(a) (Stoichiometric ratio)

Fuel + oxy. $\rightarrow$ Heat + Combustion product



16 Kg of fuel $\rightarrow$ 64 Kg of O_2

1 Kg of fuel $\rightarrow$ 4 Kg of O_2

but we know that 100 Kg of air contains 23 Kg of O_2 therefore for 4 Kg of O_2

$$\text{air required} = \frac{100 \times 4}{23} = 17.39$$

$$\left(\frac{A}{F}\right)_{st} = 17.39$$

(36)

1 kg fuel — 17.4 of air

Lean mixture :

If the mixture contains more air compare to stoichiometric requirements then that mixture is known as lean mixture.

Rich mixture :-

If the mixture contains less air compare to stoichiometric requirements then that mixture is known as rich mixture.

	Air	Fuel
lean	— more	less
Rich	— less	more

Equivalence ratio :-
(ϕ)

It is define as

$$\phi = \frac{(F/A)_{act}}{(F/A)_{st}}$$

$\phi = 1$ — stoichiometric

$\phi < 1$ — lean mixture

$\phi > 1$ — rich mixture

Reasons for variation of actual efficiency with respect to air standard effⁿ.

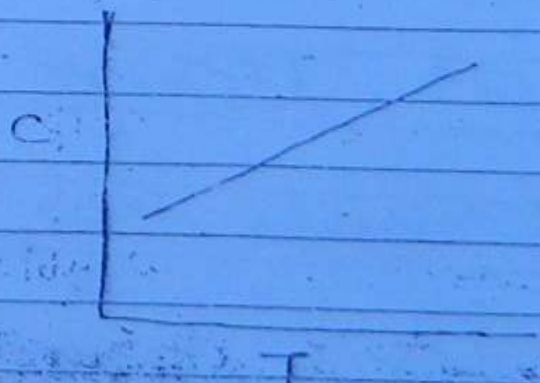
(37)

1. Non-instantaneous burning of fuel (Time losses)
2. The working fluid is not air but it is residual gases & (fuel + Air)
3. Variation of specific heat with temp.
4. dissociation of burnt gases.

Variation of specific heat with Temp.

with rise in temp. specific heat increases because larger fraction of heat input is required to produce motion of atoms within the molecules since temp. is the result of motion of molecules.

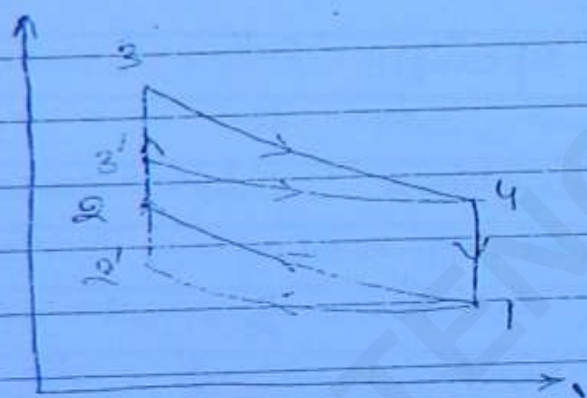
$$\text{upto } 1500\text{K} \begin{cases} C_p = a + KT \\ C_v = b + KT \\ C_p - C_v = a - b \end{cases} \quad \text{when } T > 1500\text{K} \begin{cases} C_p = a + K_1T + K_2T^2 \\ C_v = b + K_1T + K_2T^2 \end{cases}$$



with rise in temp. C_p & C_v increase
 but the difference b/w C_p & C_v
 will remain constant and hence
 with rise in temp. $\gamma = C_p/C_v$
 decreases.

(38)

$$T \uparrow, C_p \uparrow, C_v \uparrow, C_p - C_v = R = \text{const.}$$



$$T_1 v_1^{\gamma-1} = T_2 v_2^{\gamma-1}$$

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

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

$$Q = m C_v (T_3 - T_2)$$

$$1000 = m C_v (T_3 - T_2)$$

$$(T_3 - T_2) = \frac{1000}{m C_v}$$

$$C_v \uparrow, (T_3 - T_2) \downarrow$$

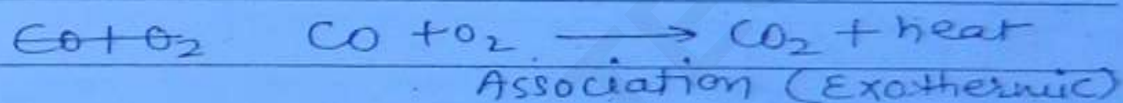
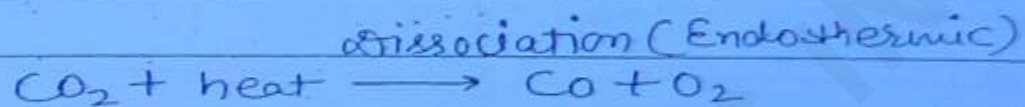
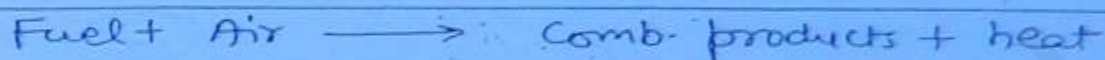
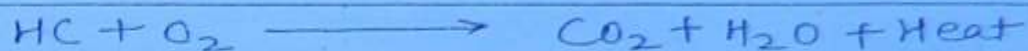
with rise in Temp. $\gamma \downarrow$ hence $T_2 \uparrow$

The effect of variable specific heat
 is to reduce peak temp. & peak
 pressure and hence the effⁿ. and

power output decreases.

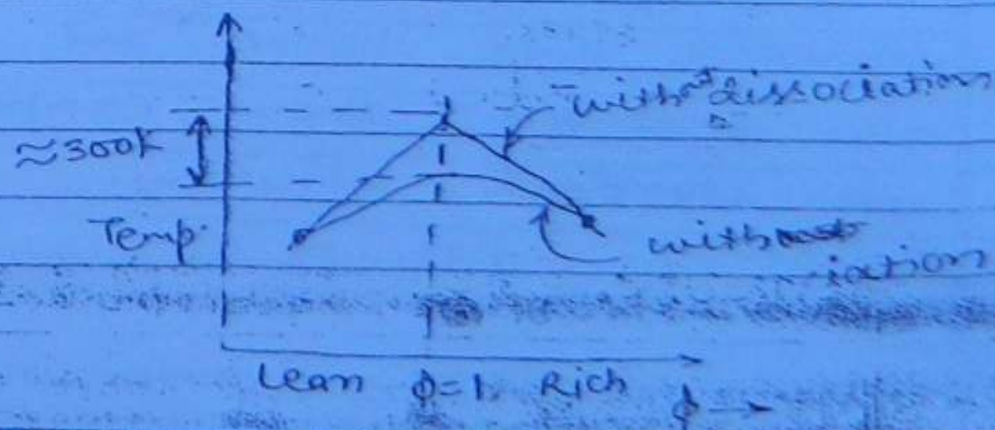
(39)

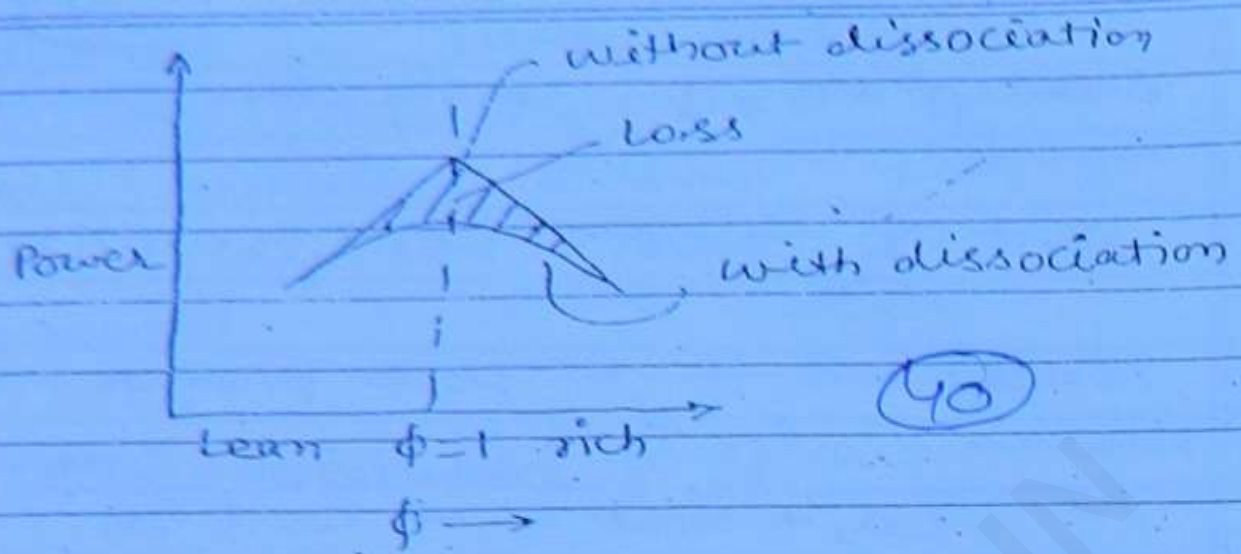
Dissociation of Combustion product:-



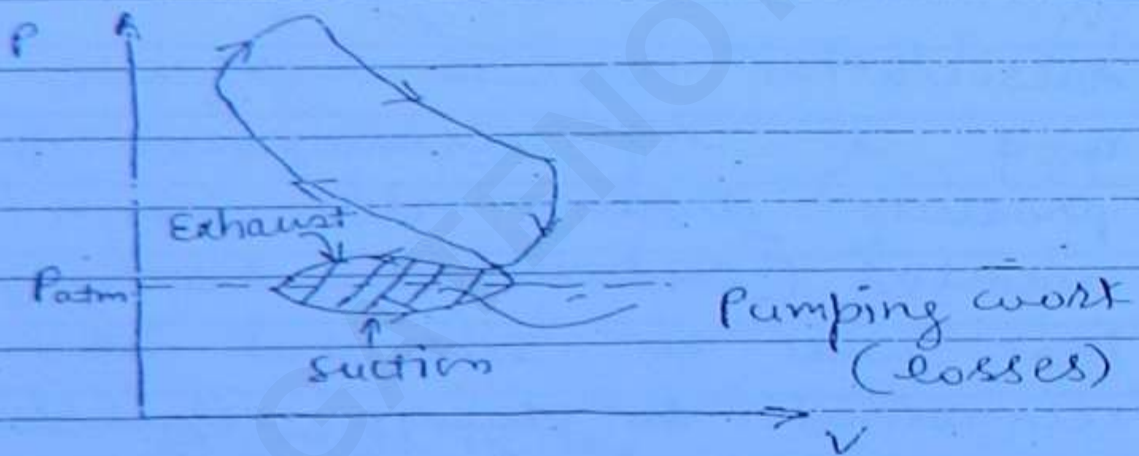
Dissociation is a reversible process and it is disintegration of combustion products. during dissociation heat is absorbed (Endothermic reaction) and this heat will be liberated when the elements recombined ~~to~~ during expansion as the temp. falls.

The effect of dissociation is to reduce part of heat liberated during combustion which is identical with the effect produced by change in specific heat.





Pumping loss :-



Pumping work is difference b/w Exhaust work and suction work. this is a negative work and because of this the net output decreases and hence the efficiency also decreases when compare to air standard efficiency.

Blow by loss :-

During Compression a portion of combustion products leaves through space b/w piston & cylinder and comes to crankcase since these products do not take part in combustion these are known as Blow by losses.

(4)

- ① Find the percentage change in the effⁿ of Otto cycle having a Comp. ratio of 7 if the specific heat at const. volume increases by 1% (initial value of $\gamma = 1.4$).

$$\eta = 1 - \frac{1}{(\gamma)^{\gamma-1}}$$

$$\frac{1}{(\gamma)^{\gamma-1}} = (1-\eta)$$

$$(\gamma)^{-(\gamma-1)} = (1-\eta)$$

$$\ln(1-\eta) = \ln(\gamma)^{-(\gamma-1)}$$

$$\ln(1-\eta) = -(\gamma-1) \ln \gamma$$

$$\ln(1-\eta) = -\left(\frac{R}{C_v}\right) \ln \gamma$$

$$\frac{1}{1-\eta} d\eta = -R \left(-\frac{1}{\gamma}\right) d\gamma \ln \gamma$$

$$\frac{-d\eta}{(1-\eta)} = \frac{R}{C_v^2} dC_v \ln r$$

(12)

$$d\eta = -(1-\eta) \frac{R}{C_v^2} dC_v \ln r$$

$$\frac{d\eta}{\eta} = \frac{-(1-\eta)}{\eta} \frac{R}{\eta} \frac{dC_v}{C_v} \ln r$$

$$\frac{d\eta}{\eta} = \frac{-(1-\eta)}{\eta} (\gamma-1) (0.01) \ln(7)$$

$$\eta = 1 - \frac{1}{(\gamma)^{\gamma-1}} = 1 - \frac{1}{(7)^{1.4-1}} = 0.5408$$

$$\begin{aligned} \frac{d\eta}{\eta} &= \frac{-(1-0.54)}{0.54} (1.4-1) 0.01 \times \ln 7 \\ &= -6.63 \times 10^{-3} \\ &= -0.663\% \end{aligned}$$

IInd method :-

$$\frac{\eta_f - \eta_i}{\eta_i} \times 100$$

$$\left[\frac{1}{(\gamma)^{\gamma-1}} - \frac{1}{(\gamma)^{\gamma-1}} \right]$$

$$\eta_i = 1 - \frac{1}{(\gamma)^{1.4-1}} = 0.5408$$

$$C_v = 0.717$$

(13)

$$C_v' = 0.717 (1 + 0.01) = 0.724$$

$$C_p' - C_v' = R = 0.287$$

$$C_p' = R + C_v' = 0.287 + 0.724 = 1.011$$

$$\eta_f = 1 - \frac{1}{(\gamma)^{\gamma'-1}}$$

$$\eta_f = 1 - \frac{1}{(7)^{1.396-1}} = 0.5376$$

$$= \frac{\eta_f - \eta_i}{\eta_i} \times 100$$

$$= \frac{0.5376 - 0.5408}{0.5408} \times 100$$

$$= -0.65\%$$

(2) A diesel cycle has a $\gamma = \frac{20}{20}$ the cut off volume 6% of stroke volume initial $C_v = 0.71$ 0.71 kJ/kg-K and

$R = 0.287 \text{ kJ/kg-K}$ find the % change in

offⁿ. if C_v increases by 3%.

Solⁿ:-

$$\eta_{li} = 1 - \frac{1}{(20)^{\gamma-1}}$$

$$= 1 - \frac{1}{(20)^{1.4-1}}$$

(44)

$$\eta_i = 1 - \frac{1}{r} \cdot \frac{1}{(r)^{\gamma-1}} \cdot \frac{(r_c)^{\gamma} - 1}{r_c - 1}$$

$$= 1 - \frac{1}{1.4} \cdot \frac{1}{(20)^{1.4-1}} \cdot \frac{(2.14)^{1.4} - 1}{2.14 - 1}$$

$$V_3 - V_2 = 0.06 (V_1 - V_2)$$

$$\frac{V_3 - V_2}{V_2} = 0.06 \frac{V_1 - V_2}{V_2}$$

$$\frac{V_3}{V_2} - 1 = 0.06 \left(\frac{V_1}{V_2} - 1 \right)$$

$$r_c - 1 = 0.06 (r - 1)$$

$$r_c - 1 = 0.06 (20 - 1)$$

$$r_c = 2.14$$

$$\eta_i = 0.642$$

$$C_v' = 1.03 \times 0.71 = 0.7313$$

$$C_p' - C_v' = R = 0.286$$

(45)

$$C_p' = C_v' + 0.286 = 0.7313 + 0.286 = 1.017$$

$$\gamma' = \frac{C_p'}{C_v'} = \frac{1.017}{0.7313} = 1.391$$

$$\eta_f = 1 - \frac{1}{\gamma'} \cdot \frac{1}{(\gamma)^{\gamma'-1}} \cdot \frac{(\gamma_c)^{\gamma'-1} - 1}{\gamma_c - 1}$$

$$= 1 - \frac{1}{1.391} \cdot \frac{1}{(20)^{1.391-1}} \cdot \frac{(2.14)^{1.391-1} - 1}{2.14 - 1}$$

$$= 0.6322$$

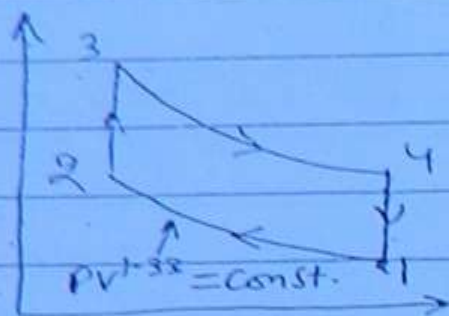
$$= \frac{0.682 - 0.6422}{0.642} \times 100$$

$$= -1.58$$

- (3) An otto cycle has a $\gamma = 10$ and uses fuel having a calorific value of 42000 KJ/kg. Air/Fuel = 15/1 the temp & pressure at the start of combustion compression is 350K and 1 bar determine the max. pressure in the cycle where is index of compression is 1.35 and $C_v = (0.717 + 2 \times 10^{-5} T)$ KJ/kg

Solⁿ:

$$\gamma = 1.33$$



(46)

2-3 (V=c)

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

1-2

$$P_1 V_1^{1.33} = P_2 V_2^{1.33}$$

$$P_2 = P_1 \left(\frac{V_1}{V_2} \right)^{1.33}$$

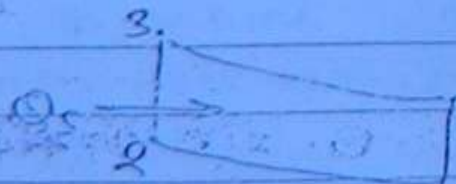
$$P_2 = 1 (10)^{1.33} = 21.38 \text{ bar}$$

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$$

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

$$T_2 = T_1 (10)^{1.33-1}$$

$$T_2 = 748.28 \text{ K}$$



$$Q_s = m_f \times CV$$

$$m_f \times CV = \int_{T_2}^{T_3} (m_a + m_f) C_v dT \quad (47)$$

$$m_f \times CV = (m_a + m_f) \int_{T_2}^{T_3} (0.717 + 2 \times 10^{-4} T) dT$$

$$42000 = \frac{m_a + m_f}{m_f} \left[0.717(T_3 - T_2) + \frac{2 \times 10^{-4}}{2} (T_3^2 - T_2^2) \right]$$

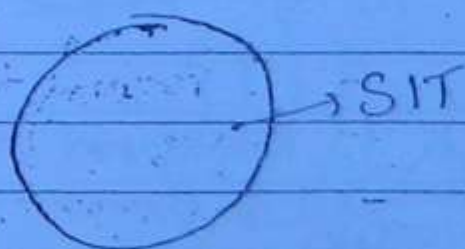
$$42000 = 15 + 1 \left[0.717(T_3 - 748.28) + 10^{-4} (T_3^2 - 748.28^2) \right]$$

$$10^{-4} T_3^2 \mp 0.717 T_3 = 3217.51$$

$$T_3 = 3125.4 \text{ K}$$

$$P_3 = 21.38 \times \frac{3125.4}{748.28}$$

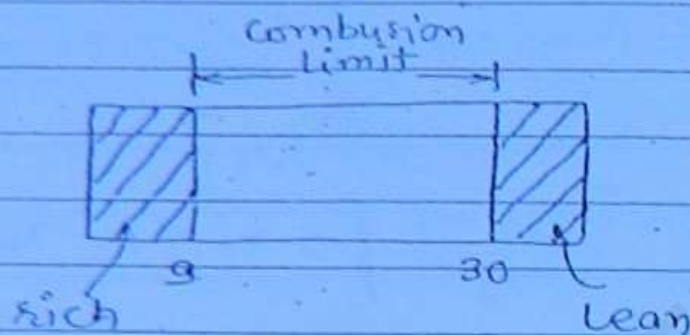
$$= 89.29 \text{ bar}$$



Ignition by
flame Speed - 15-30 m/s

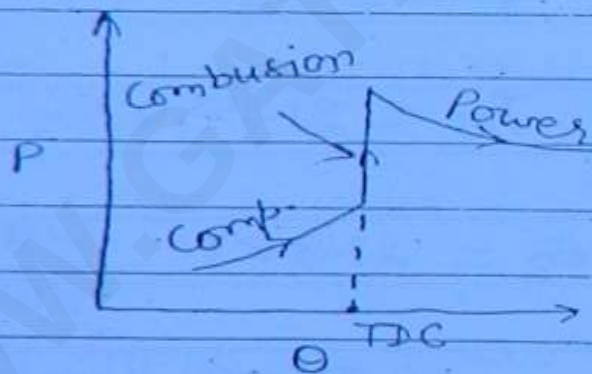
Combustion in S.I. Engine

Combustion Limits for S.I. Engines :- (48)

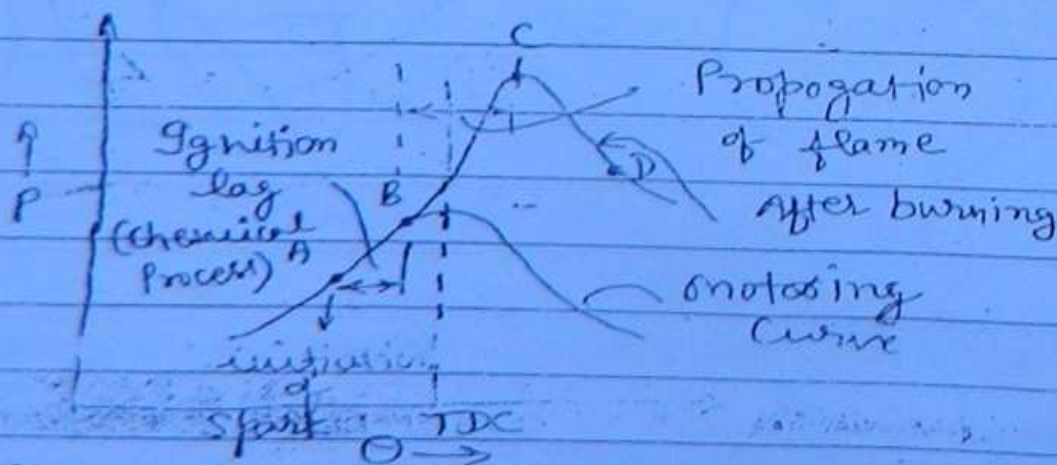


Ignition limits :-

Theoretical P- θ diagram :-



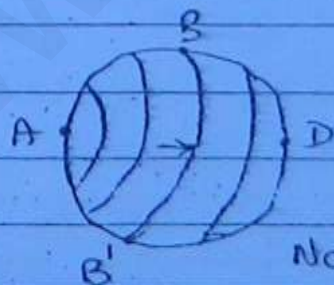
Stages of Combustion in S.I. Engine



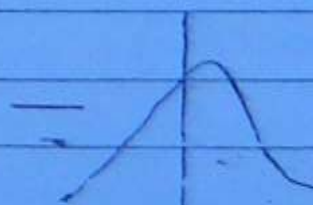
- * Point A is passage of spark (initiation of spark) it is generally 20° before TDC.
- * Point B it is the point at which the beginning of pressure rise can be detected generally 8° before TDC.
- * Point C is attainment of peak pressure
- * AB is first stage of combustion, BC is second stage of combustion and CD is third stage of combustion.
- * AB is called ignition lag or preparation phase in which the growth and development of flame takes place this is a chemical process.
- * BC It is consistent with spreading of flame throughout the combustion chamber the starting point of second phase is where the first measurable pressure rise is seen on diagram.

(49)

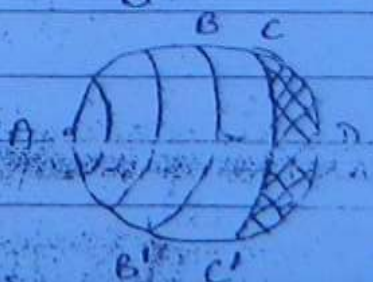
Normal Combustion Vs Abnormal Combustion:-



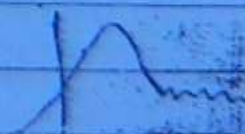
Normal Combustion



(Fig-I)



Abnormal Combustion



(Fig-II)

* Fig. J shows a flame front traveling across combustion chamber from A to D as the flame propagates the unburnt mixture called as 58 end gas is compressed causes its temp to increase. The temp. of the end gas increases further when it receives heat by $\propto$ radiation from burning charge.

* Thus some of the end charge may undergo preflame reactions increasing its temp. further if the temp. exceeds self-ignition temp. and if the unburnt gas remains at or above self ignition temp. during the ignition delay periode and advancing flame front does not reach the end gas spontaneous ignition or auto ignition will occur this is known as Detonation or Knocking.

*
** Effects of Detonation:-

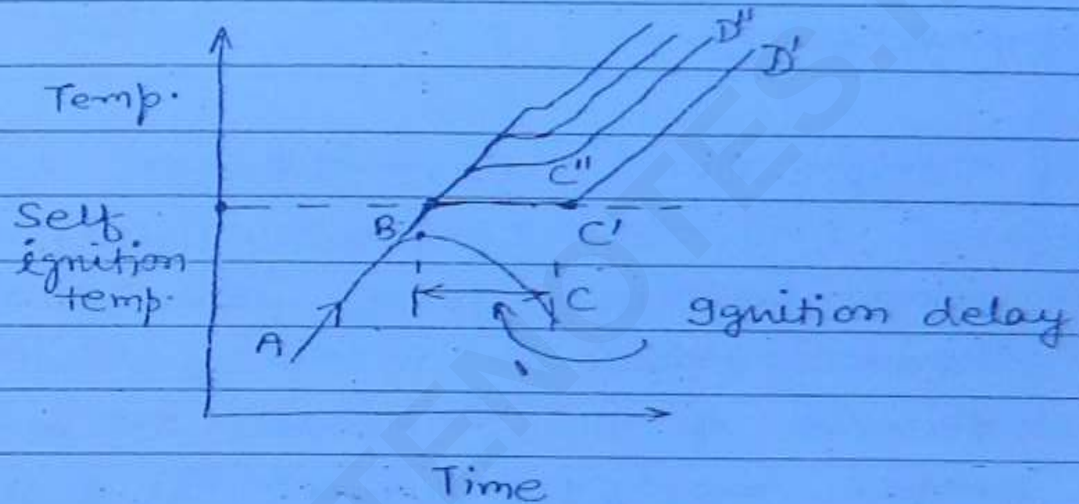
1. Noise & Roughness
2. Mechanical damage
3. Loss of power

4. Loss in efficiency.
5. Pre-ignition.

(51)

Factors controlling / affecting detonation:-

Auto-Ignition Theory:-



Auto ignition does not occur immediate as the self ignition temp. is reached some ignition delay periode is required before reaction becomes explosive. Suppose that a homogeneous fuel/air mixture when they rapidly compressed and held at that temp. and if this temp. is less then self ignition temp. (Point B) the mixture will not autoignite but will slowly cool down. If the self ignition temp. is reached

Certain ignition delay. The delay period is shortened if the temp is raised due to greater molecular activities.

(52)

Factors effecting Detonation (or)

Knocking :-

1. Compression ratio :- Increase in compression ratio, increases temp. reduces delay period and hence knocking tendency increases. Because of this reason the compression ratios in S.I engines is restricted to about 10. (6-10)

2. Supercharging :- Supercharging is the process of allowing the charge to enter at higher pressure into the cylinder. Supercharging results in increase in temp and hence the chances of detonation also increase.

Note :- Supercharging is generally used in C.I engines.

3. Raising inlet Temp :-

It results in increase in detonation

(53)

** 4. Increasing load :-

Increase in load results in increase in temp of cylinder thereby raising the temp of end charge and hence knocking tendency increases

*** 5. Advancing the spark :-

when the spark is advanced the burning gas is compressed by rising piston and therefore the temp is increased this increases knocking tendency and hence spark must be retarded.

6. Flame travel distance :-

Increasing flame travel distance results in increase in detonation

7. Spark Plug location :-

A spark plug which is located centrally has more tendency for knocking because the flame travel distance decrease.

8. Location of exhaust valve:-

The exhaust valve should be located close to spark plug so that the exit is not in the end gas region otherwise detonation will occur.

9. Engine Size:- Flame requires longer time to travel across the combustion chamber of a large engine. therefore larger engines have greater tendency for knocking than small engines. therefore S.I. engine is generally limited to 100 mm bore.

10. Turbulence:- Increase in turbulence increases flame speed and hence knocking tendency decreases.

11. Engine speed:- Increase in engine speed increases turbulence and hence knocking tendency decreases.

12. Octane Rating of Fuel:- Lower the self ignition temp. more the knocking.

Tendency. A higher octane no. means lower knocking tendency.

13. Fuel/Air ratio:-

when the mix. is slightly richer (say 10%) the ~~max~~ temp. reached are very high and because of this higher temp. the ignition lag is minimum and also the flame propagation

$T \uparrow \rightarrow \text{ignition lag} \downarrow \rightarrow \text{detonation} \uparrow$

$T \uparrow \rightarrow \text{flame speed} \uparrow \rightarrow \text{detonation} \downarrow$

velocity is max. But the effect of former is more and hence knocking tendency is found to be more at about 10% richer mixture.

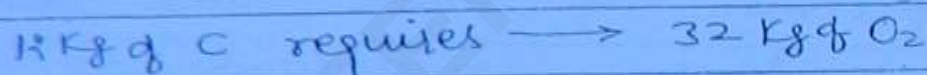
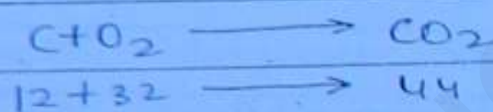
Pre-Ignition

Ignition of homogeneous mixture ~~of~~ in the cylinder before the initiation of spark is known as pre-ignition. It is caused by local overheating of combustible mixture.

Note: - There is no pre-ignition in C.I. engines due to the absence of fuel during compression.

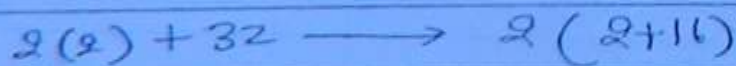
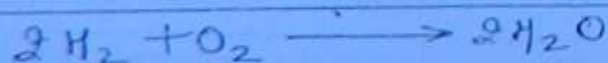
(56)

Minimum Air required for Combustion

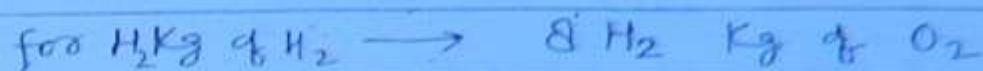
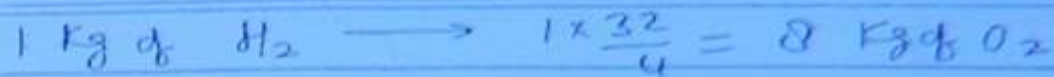


For C Kg of Carbon oxygen required is $\frac{8}{3} \times C$

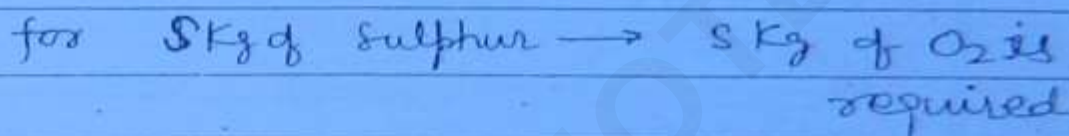
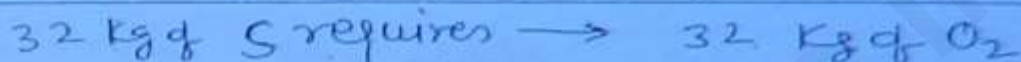
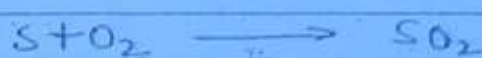
Hydrogen Combustion Equation:-



4 Kg H₂ requires 32 Kg of O₂



If sulphur is present in fuel



Total O_2 required for the Combustion of fuel

$$= \left[\frac{8}{3} C + 8 H_2 + S \right]$$

If some O_2 is available in fuel the net oxygen required for Combustion

$$= \left[\frac{8}{3} C + 8 H_2 + S - O_2 \right]$$



$$= \left[\frac{8}{3} C + 8 H_2 + S - O_2 \right] \text{ Kg of } O_2$$

$$* \text{ Air required for 1 kg fuel } = \left[\frac{8}{3} C + 8 H_2 + S - O_2 \right] \frac{100}{23}$$

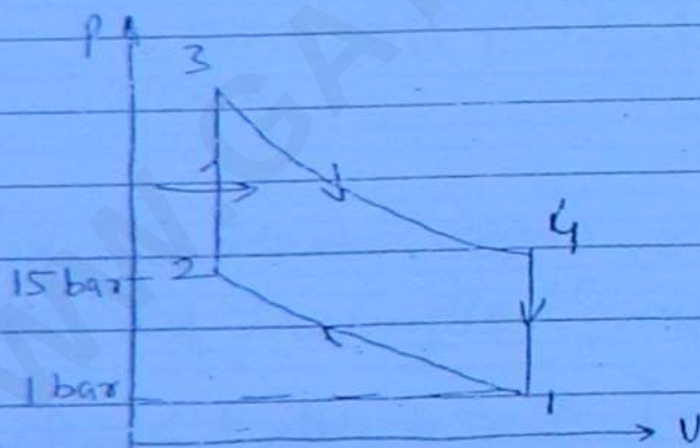
This is known as stoichiometric
Air/fuel ratio.

58

Prob: ① In an Otto cycle air at 17°C and
1 bar is compressed adiabatically
untill the pressure is 15 bar
heat is added at const. volume
untill the pressure rises to
40 bar Calculate

- effⁿ.
- Compⁿ. ratio
- mep ($C_v = 0.717 \text{ kJ/kg-K}$, $R = 0.287$)

Solⁿ: -



$$P_1 = 1 \text{ bar} \quad P_2 = 15 \text{ bar} \quad P_3 = 40 \text{ bar}$$

$$T_1 = 17 + 273 = 290 \text{ K}$$

(1-2)

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

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

$$\frac{15}{1} = (r)^{1.4}$$

$$r = 6.919$$

(59)

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

$$= 1 - \frac{1}{(6.919)^{1.4-1}} = 0.5387$$

$$= 53.87\%$$

$$mep = \frac{w_{net}}{V_1 - V_2}$$

for 2-3

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$$T_3 = \frac{40}{15} \times 628.6$$

$$= 1676.4$$

(1-2)

$P_1 V$

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$$

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

$$= 290 (6.919)^{1.4-1}$$

$$= 628.65 \text{ K}$$

$$Q_s w_{net} = C_v (T_3 - T_2)$$

(Q2)

$$w_{net} = \eta \times Q_s$$

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

$$= 0.5387 \times 0.717 (1676.4 - 628.65)$$

$$w_{net} = 404.69$$

$$S_o = 0.717 (1676.4 - 628.65)$$

$$= 751.25 \text{ kJ/kg}$$

$$P_1 V_1 = 1 \times R T_1$$

$$V_1 = \frac{R T_1}{P_1}$$

66

$$V_1 = \frac{0.287 \times 290}{100}$$

$$V_1 = 0.823$$

$$\frac{V_1}{V_2} = 6.91$$

$$V_2 = \frac{0.823}{6.91} = 0.1204$$

$$\begin{aligned} mep &= \frac{W_{net}}{V_1 - V_2} \\ &= \frac{(404.69)}{0.823 - 0.1204} \\ &= 575.9 \end{aligned}$$

$$mep = 5.75 \text{ bar}$$

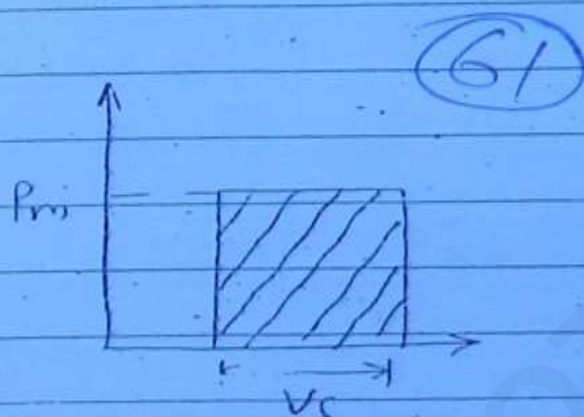
IInd methode

$$\begin{aligned} mep &= \frac{\eta_{th} \times \Delta P}{(\gamma - 1)(\gamma - 1)} \\ &= \frac{0.5387 \times (40 - 15)}{(6.91 - 1)(1.4 - 1)} \end{aligned}$$

$$mep = 5.69 \text{ bar}$$

Performance parameters :-

1. Indicated Power (I.P.) :- The power developed inside the cylinder is known as indicated power (I.P.)



$$W_{net} = \frac{P_m \times V_s}{\text{Cycle}}$$

$$IP = \frac{P_m V_s}{\text{Cycle}} \times \frac{N_{\text{cycle}}}{60 \text{ sec}}$$

$$IP = \frac{P_m \times A \times L \times N}{60} \quad \left(\text{For two stroke engine} \right)$$

For four stroke engine

$$I.P. = \frac{P_m L A N}{2 \times 60}$$

If there are K no. of cylinders then

$$IP = \frac{P_m L A N K}{2 \times 60} \quad \left(\text{for 4-stroke} \right)$$

$$I.P. = \frac{P_m L A N K}{60} \quad \left(\text{for 2-stroke} \right)$$

20. Brake Power (B.P.) :- It is the power available at output shaft.

(62)

$$\text{Brake power} = \frac{T \times 2\pi N}{60}$$

T = Brake Torque

3. Frictional Power (F.P.) :- The difference between I.P. & B.P. is known as frictional power.

4. Mechanical efficiency (η_m) :-

$$\eta_m = \frac{BP}{IP} = \frac{\eta_{bth}}{\eta_{ith}}$$

5. Indicated Thermal efficiency (η_{ith}) :-

$$\eta_{ith} = \frac{IP}{m_f \times C.V.}$$

m_f = mass of fuel

C.V. = Calorific value

6. Brake Thermal efficiency (η_{bth}):-

$$\eta_{bth} = \frac{BP}{m_f \times CV}$$

(63)

7. Indicated specific fuel consumption (ISFC):-

It is the amount of fuel consume to develop I.P.

$$* \quad ISFC = \frac{m_f}{IP}$$

m_f = mass of fuel consume in Kg/hr. and IP is in Kw

Note:- Generally specific fuel consumption in $\frac{Kg}{Kw-hr}$.

$$* \quad \eta_{th} = \frac{IP}{m_f CV} = \frac{Kw}{\frac{Kg \times \frac{KJ}{Kg}}{Sec}} = \frac{Kw}{Kw}$$

Brake specific fuel consumption (BSFC):-

It is the fuel consume to develop brake power.

$$BSFC = \frac{m_f}{BP} \quad \frac{Kg}{Kw-hr}$$

$$\eta_m = \frac{BP}{IP} = \frac{\frac{mf}{BSFC}}{\frac{mf}{ISFC}} = \frac{ISFC}{BSFC}$$

(69)

Combustion in C.I. Engines

- (a) Compression ignition engines
- (b) Diesel engines

Stages of Combustion in C.I. Engines:-

- (1) Delay periode
 - a. Physical delay
 - b. chemical delay
- (2) Uncontrolled Combustion
- (3) Controlled Combustion
- (4) After burning

Physical Delay:-

The time between beginning of injection and attainment of chemical reaction condition is known as physical delay. During this periode fuel is atomised

⑤) vapourised and mixed with air.

⑥5

Chemical delay:-

Reaction starts slowly an ignition takes place. Ignition delay in S.I. engine is equivalent to chemical delay for C.I. engines.

As soon as the fuel is injected in to the combustion chamber it will vapourised and mixes with air. The time spend in this process is known as physical delay. The free flame reactions start and after some time fuel burns automatically. And the time consume in this is known as Chemical delay.

During total delay periode more fuel droplets comes out from the injector and group of droplets burn together. This produces uncontrolled combustion known as Detonation. At the beginning of combustion.

After some periode of time the temp. of combustion chamber is so high then as soon as the fuel droplets enter in the combustion chamber it burns instantaneously and this is called as Controlled Combustion.

Since diesel is less volatile there are some pockets of air/fuel mixture which will burn during expansion known as After burning.

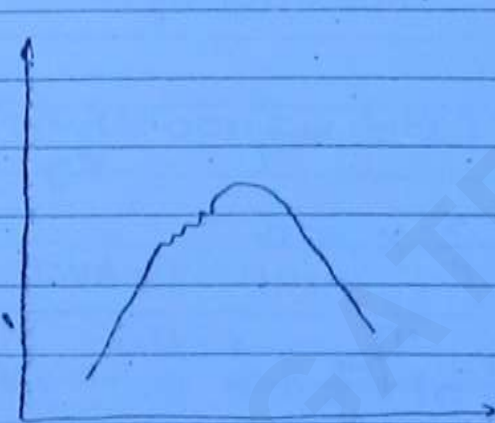
66

- Ques:-
- ① The parameters which control detonation in S.I. engines increase detonation in C.I. engines and hence a good S.I. engine fuel is a bad C.I. engine fuel.
 - ② If petrol is used in C.I. engines uncontrolled combustion or detonation occurs because the self ignition temp. of petrol is very high than diesel so that the delay period will be more resulting in violent knocking and due to this they can be mechanical failure of engine parts.
 - ③ If diesel is used in S.I. engines and as diesel is less volatile black fumes come out from the engine and thereby large no. of carbon atoms deposit at the spark plug and hence the spark plug will be blocked.

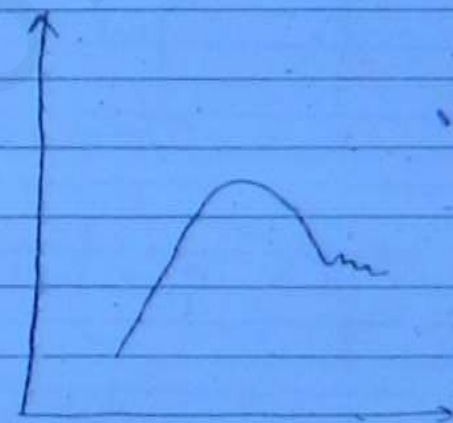
Factors Controlling Detonation

(67)

Factors	S.I.	C.I.
S.I. T	high	low
delay period	high	low
CR	Low	high
Inlet Temp.	Low	high
Inlet pressure	Low	high
Combustion wall temp.	Low	high



C.I. Engines



S.I. Engines

6796

Prob ①

A 4-Cylinder, four stroke S.I. engine has a Comp. ratio 8 and bore of 100 mm with a stroke = bore the volumetric effⁿ of each cylinder is 75% the engine operates at a speed of 4800 rpm with an Air/Fuel ratio of 15. The CV of fuel is 42000 kJ/kg

Density of air is 1.12 Kg/m^3
and mep in the cylinder is
10 bar and mechanical effⁿ.
is 80%. Determine

- Indicated Thermal effⁿ.
- Brake power.

(68)

Solⁿ:-

$$K=4$$

four stroke engine

$$r=8$$

$$D=L=100 \text{ mm}$$

$$\eta_{vol} = 0.75$$

$$N=4800 \text{ rpm}$$

$$A/F=15$$

$$CV = 42000 \frac{\text{KJ}}{\text{Kg}}$$

$$\rho_a = 1.12 \text{ Kg/m}^3$$

$$mep = 10 \text{ bar}$$

--

$$\eta_m = 0.80$$

$$I.P. = \frac{P_m L A N K}{60 \times 2}$$

$$= \frac{(10 \times 100) \times (0.1) \pi (0.1)^2 \times 4800 \times 4}{4 \times 60 \times 2}$$

$$= 125.66 \text{ kW}$$

$$\eta_m = \frac{B.P.}{I.P.}$$

$$B.P. = \eta_m \times I.P.$$

$$= 0.8 \times 125.66$$

$$= 100.528 \text{ kW}$$

$$\eta_{\text{vol.}} = \frac{\text{Act. Volume of air taken/cycle}}{\text{Swept Volume}}$$

(69)

$$4800 \text{ rpm} = 2400 \text{ Cycles/min.}$$

(As it is a four stroke engine each cycle is completed in 2 revolutions)

$$\eta_{\text{vol.}} = \frac{V_a/\text{cycle}}{V_s}$$

$$V_a/\text{cycle} = \eta_{\text{vol.}} \times V_s$$

$$= 0.75 \times \frac{\pi}{4} (0.1)^2 (0.1) \quad \text{m}^3/\text{cycle}$$

$$V_a = 5.89 \times 10^{-4} \text{ m}^3/\text{cycle}$$

$$V_a = 5.89 \times 10^{-4} \frac{\text{m}^3}{\text{cycle}} \times \frac{2400 \text{ cycle}}{60 \text{ sec.}}$$

$$V_a = 0.02356 \frac{\text{m}^3}{\text{sec.}}$$

As there are 4-cylinders to

$$\text{Total Volume taken} = 4 \times 0.02356$$

$$= 0.09424 \frac{\text{m}^3}{\text{sec.}}$$

$$\rho_a = \frac{m_a}{V_a}$$

$$m_a = \rho_a \times V_a$$

$$= 1.12 \frac{\text{kg}}{\text{m}^3} \times 0.09424 \frac{\text{m}^3}{\text{sec}}$$

$$= 0.1055 \frac{\text{kg}}{\text{sec}}$$

$$\frac{m_a}{m_f} = 15$$

$$m_f = \frac{m_a}{15} = \frac{0.1055}{15}$$

$$m_f = 7.036 \times 10^{-3} \frac{\text{kg}}{\text{sec}}$$

$$\eta_{\text{ith}} = \frac{\text{IP}}{m_f \times C \cdot V}$$

$$= \frac{125.66 \text{ kW}}{7.036 \times 10^{-3} \frac{\text{kg}}{\text{sec}} \times 42 \times 10^3 \frac{\text{kJ}}{\text{kg}}}$$

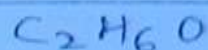
$$= 0.425$$

$$\eta_{\text{ith}} = 42.5 \%$$

(2)

The chemical formula of Fuel is C_2H_6O . Calculate stoichiometric Air/Fuel ratio and % Composition of products of composition per kg alcohol.

Soln:-



(77)

$$12 \times 2 \rightarrow C = 24$$

$$1 \times 6 \rightarrow H = 6$$

$$16 \rightarrow O = 16$$

$$46$$

$$\% C = \frac{24}{46} = 0.52$$

$$\% H_2 = \frac{6}{46} = 0.13$$

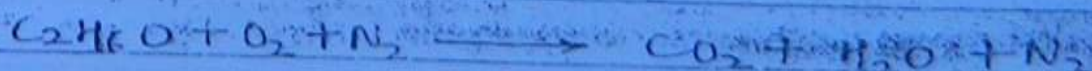
$$\% O_2 = \frac{16}{46} = 0.35$$

$$\left(\frac{A}{F}\right) = \left[\frac{8}{3}C + 8H_2 + S - O_2 \right] \times \frac{100}{23}$$

$$\left(\frac{A}{F}\right)_{st.} = \left[\frac{8}{3} \times (0.52) + 8(0.13) + 0 - (0.35) \right] \times \frac{100}{23}$$

$$\left(\frac{A}{F}\right)_{st.} = 9.03$$

For 1 kg of fuel burns, 9.03 kg air required.





(72)

12 kg C produces 44 kg of CO_2

1 kg C ——— 2.

$$1 \text{ kg C produces } = \frac{44}{12} = \frac{11}{3} \text{ kg of } CO_2$$

$$0.52 \text{ kg C produces } = \frac{11}{3} \times 0.52$$

$$= 1.906 \text{ kg of } CO_2$$

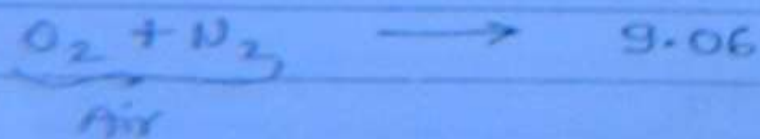


2 kg H_2 produces 18 kg of H_2O vapour

1 kg H_2 " 9 kg of H_2O vapour

0.13 kg H_2 " 9×0.13

$$= 1.17 \text{ kg of } H_2O \text{ vapour}$$



100 kg air ——— 77 kg N_2

9.06 kg air

$$\frac{77 \times 9.06}{100}$$

100

(73)

$$= 6.97 \text{ kg of } N_2$$

$$CO_2 \quad 1.906 \quad \frac{1.906}{10.05} \times 100 = 18.96\%$$

$$H_2O \quad 1.17 \quad \frac{1.17}{10.05} \times 100 = 11.64\%$$

$$N_2 \quad 6.97 \quad \frac{6.97}{10.05} \times 100 = 69.35\%$$

2 4

4-5 gpm
air

TM motor

heavy duty

Power for
Saw & Sma

Low voltage
for
luminaires

Valves

Light Input

Not much

$\eta_H \uparrow$

Barrels

2

1-5 hr

low

high

low

low

Power

Power

more like

part

low input

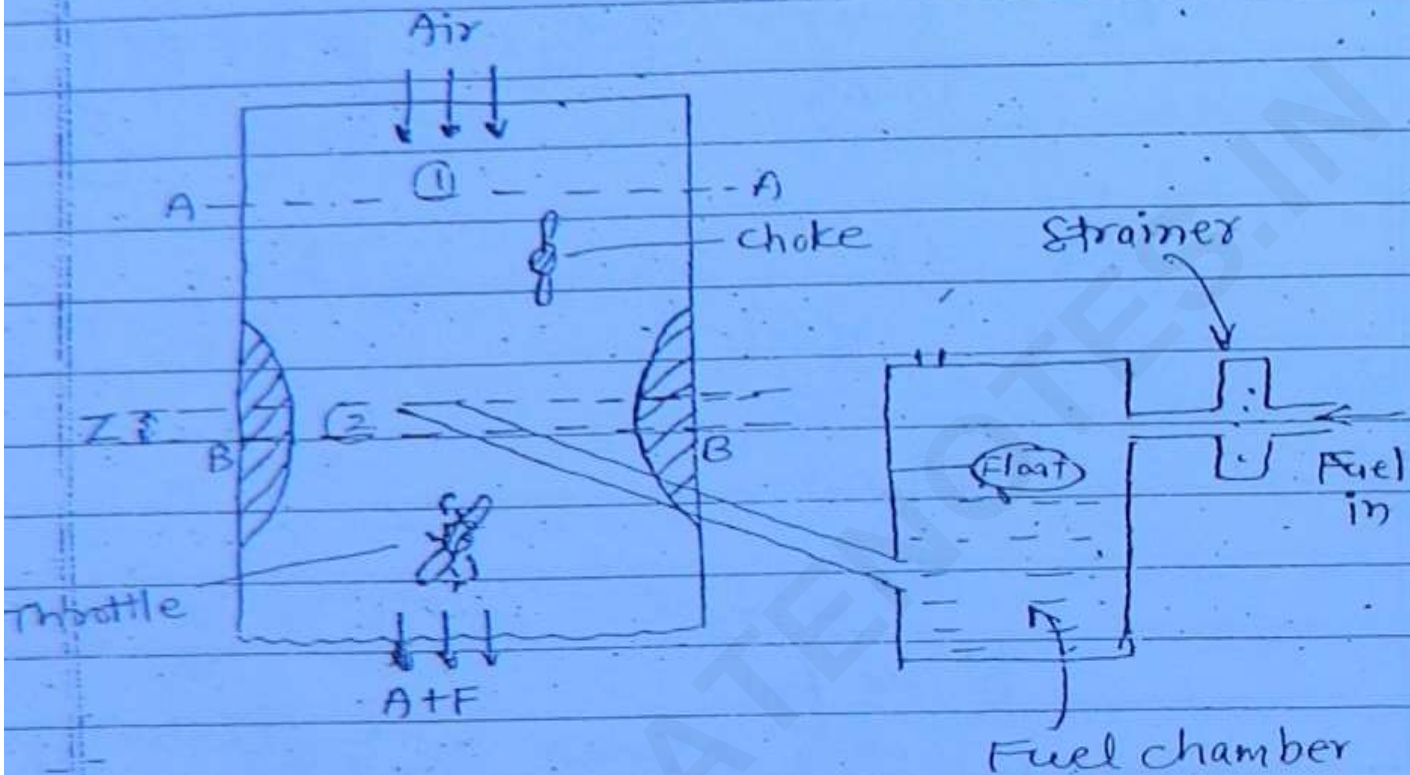
Not less

$\eta_H \downarrow$

Carburetion & Fuel Injection

Simple Carburetor :-

(74)



Carburetor is a device which supplies combustible mixture to the cylinder

Exact analysis to find Air/Fuel ratio:-

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

$$C_1 \ll C_2$$

$$h_1 = h_2 + \frac{C_2^2}{2}$$

$$C_2 = \sqrt{2(h_1 - h_2)}$$

C_2 = velocity of air at throat

$$C_2 = \sqrt{2(C_p T_1 - C_p T_2)} \quad (75)$$

$$C_2 = \sqrt{2 C_p T_1 \left(1 - \frac{T_2}{T_1}\right)}$$

Note: Assuming isentropic flow.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

P_2 = Throat pressure

P_1 = Atm. pressure

$$C_2 = \sqrt{2 C_p T_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

$$m_a = P_2 A_2 C_2$$

$$\therefore m = \rho A C$$

$$m_a = \rho_2 Q$$

$$m_a = \text{Theoretical mass} = \rho_2 A_2 C_2$$

Actual mass

$$\dot{m}_a = C_d \cdot \rho_2 A_2 C_2$$

Theoretical mass flow rate $\Rightarrow$

$$\dot{m}_2 = \rho_2 A_2 C_2$$

where $A_2 =$ Area of venturi at the float

$\rho_2 =$ density of air at

(76)

Actual mass flow rate of Air

$$\dot{m}_2 = C_d \rho_2 A_2 C_2$$

$C_d =$ coefficient of discharge for venturi

$$pV = nRT$$

$$p = \left(\frac{m}{V} \right) RT$$

$$p = \rho RT$$

$$\rho = \frac{p}{RT}$$

$$\rho_1 = \frac{p_1}{RT_1}$$

p_1, T_1 - atm. pressure & temp.

isentropic

$$pV^\gamma = \text{const.}$$

$$\frac{p_1}{\rho_1^\gamma} = \frac{p_2}{\rho_2^\gamma}$$

$$\frac{\rho_2}{\rho_1} = \left(\frac{p_2}{p_1} \right)^{1/\gamma}$$

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

∴

$$m_a = C_d P_2 A_2 C_2$$

(77)

$$m_a = C_{da} P_1 \left(\frac{P_2}{P_1} \right)^{1/\gamma} A_2 C_2$$

$$m_a = C_{da} P_1 \left(\frac{P_2}{P_1} \right)^{1/\gamma} A_2 \sqrt{2 C_f T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} \right]}$$

Mass of Fuel :-

Assuming fuel to be incompressible ($P_f = \text{const.}$)

* The nozzle tip is kept at a height with respect to fuel surface to avoid Spilling of fuel.

$$\frac{P_1}{\rho_f g} + \frac{v_1^2}{2g} + z_1 = \frac{P_2}{\rho_f g} + \frac{v_2^2}{2g} + z_2$$

$$\frac{P_1}{\rho_f g} + \frac{0^2}{2g} + 0 = \frac{P_2}{\rho_f g} + \frac{C_f^2}{2g} + z$$

$$\frac{P_1}{\rho_f g} - \frac{P_2}{\rho_f g} - z = \frac{C_f^2}{2g}$$

$$\frac{P_1 - P_2 - \rho_f g z}{\rho_f g} = \frac{C_f^2}{2g}$$

(78)

$$C_f = \sqrt{\frac{2[(P_1 - P_2) - \rho_f g z]}{\rho_f}}$$

Theoretical

$$m_f = \rho_f A_f C_f$$

Actual mass flow rate of fuel

$$m_f = C_{df} \rho_f A_f C_f$$

$$m_f = C_{df} \rho_f A_f \sqrt{\frac{2[(P_1 - P_2) - \rho_f g z]}{\rho_f}}$$

$$m_f = C_{df} \cdot A_f \sqrt{2 \rho_f [(P_1 - P_2) - \rho_f g z]}$$

Approximate analysis to find A/F ratio :-

Air is assumed to be incompressible.

$$m_a = \rho_a C_1 A_1$$

(79)

$$m_a = C_{da} \rho_a A_2 C_2$$

$$\frac{P_1}{\rho g} + \frac{C_1^2}{2g} = \frac{P_2}{\rho g} + \frac{C_2^2}{2g}$$

$$C_1 \lll C_2$$

$$C_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_a}}$$

$$m_a = C_{da} \rho_a A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho_a}}$$

$$m_a = C_{da} A_2 \sqrt{2 \rho_a (P_1 - P_2)}$$

$$\frac{A}{F} = \frac{m_a}{m_f} = \frac{C_{da} A_2 \sqrt{2 \rho_a (P_1 - P_2)}}{C_{df} A_f \sqrt{2 \rho_f [(P_1 - P_2) - \rho_f g z]}}$$

Note:- If nozzle tip height z is neglected

$$\frac{A}{F} = \frac{C_{da} A_2 \sqrt{P_0}}{C_{df} A_f \sqrt{P_f}} \quad (83)$$

$$\frac{A}{F} = \frac{C_{da} A_2 \sqrt{P_0}}{C_{df} A_f \sqrt{P_f}}$$

At higher altitudes the density of air decreases and hence from the above equation A/F will be less that is the mixture becomes rich and hence the drawback of simple Carburetor is it supplies rich mixture at higher altitudes.

- (1) A simple carburetor has to supply 5 kg of air/min. the atm. air is at a pressure of 1.013 bar and at the temp. of 27°C. Calculate throat diameter of venturi if the air flow velocity at the throat (throat) is 90 m/sec. & velocity coefficient is 0.8. Assume isentropic flow and treat air as compressible flow.

Soln:-

(8)

$$m_a = 5 \text{ kg/min}$$

$$P_1 = 1.013 \text{ bar}$$

$$T_1 = 27 + 273$$

$$(2) P_1 = 101.3 \text{ kPa}$$

$$= 300 \text{ K}$$

$$C_2 = V_2 = 90 \text{ m/sec}$$

$$C_v = 0.8$$

$$P_2 = P_1 \left[\frac{P_2}{P_1} \right]^{1/\gamma}$$

$$P_1 = \frac{P_1}{RT_1}$$

$$= \frac{1.013 \times 10^5}{0.287 \times 300}$$

$$P_2 = 0.01176 \left[\frac{P_2}{P_1} \right]^{1/\gamma}$$

$$P_1 = 1.176 \text{ kPa}$$

$$C_2 = C_v \times \sqrt{2 C_p T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} \right]}$$

$$C_p = 1.005 \frac{\text{kJ}}{\text{kg-K}}$$

$$C_p = 1005 \frac{\text{J}}{\text{kg-K}}$$

$$90 = 0.8 \times \sqrt{2 \times 1005 \times 300 \left[1 - \left(\frac{P_2}{1.013} \right)^{\frac{1.4-1}{1.4}} \right]}$$

$$P_2 = 0.01176 \text{ bar}$$

$$\rho_2 = \rho_1 \left[\frac{\rho_2}{\rho_1} \right]^{1/\gamma}$$

$$= 1.176 \left[\frac{0.941}{1.013} \right]^{1/1.4}$$

$$\rho_2 = 1.1054 \text{ kg/m}^3$$

assume $C_d = 1$

(82)

$$m_q = \rho_2 A_2 C_2$$

$$\frac{5}{60} = 1.1054 \times A_2 \times 90$$

$$A_2 = 8.37 \times 10^{-4} \text{ m}^2$$

$$A_2 = \frac{\pi}{4} d_2^2$$

$$d_2 = \sqrt{\frac{4A}{\pi}}$$

$$= \sqrt{\frac{4 \times 8.37 \times 10^{-4}}{\pi}}$$

$$d_2 = 0.0325 \text{ m}$$

$$d_2 = 3.25 \text{ cm}$$

(or)

$$d_2 = 32.5 \text{ mm}$$

(2)

Determine Air/Fuel ratio at 6000 m altitude in a carburetor adjusted to give an air/fuel ratio of 15 at sea level where air temp is 27°C and pressure is 1.013 bar.

The temp of air decreases with altitude and is given by

$$t = t_s - 0.0065h$$

(82)

where h = height in meter

and t_s = sea level temp in $^\circ\text{C}$

The air pressure decreases with altitude as per the relation

$$h = 19220 \log_e \left(\frac{1.013}{P} \right)$$

where P is in bar

Solⁿ:

$$\frac{\text{air}}{\text{fuel}} = 15$$

$$m_a = 15$$

$$P_1 = 1.013 \text{ bar} = 101.3 \text{ kPa}$$

$$T_1 = t_1 = 27 + 273 = 300 \text{ K}$$

$$\rho_a = \frac{P_1}{RT_1} = \frac{101.3}{0.287 \times 300}$$

$$(\rho_a)_{sl} = 1.176$$

$$\frac{A}{F} = \frac{m_a}{m_f} = \frac{C_d a A_2}{C_d f A_f} \sqrt{\frac{\rho_a}{\rho_f}}$$

$$\left(\frac{A}{F} \right) \propto \sqrt{\rho_a}$$

SL - Sea Level

$$\frac{\left(\frac{A}{F}\right)_{SL}}{\left(\frac{A}{F}\right)_h} = \sqrt{\frac{(p_a)_{SL}}{(p_a)_h}}$$

(84)

$$\left(\frac{A}{F}\right)_h = \left(\frac{A}{F}\right)_{SL} \sqrt{\frac{(p_a)_h}{(p_a)_{SL}}}$$

$$t = t_s - 0.0065 h$$

at $h = 6000$

$$t = 27 - 0.0065 (6000)$$

$$t = -12^\circ \text{C}$$

$$T = 273 - 12 = 261 \text{ K}$$

$$h = 19220 \log_e \left(\frac{1.013}{p} \right)$$

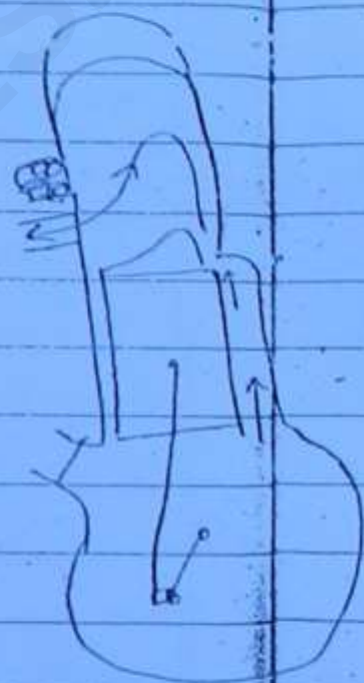
$$6000 = 19220 \log_e \left(\frac{1.013}{p} \right)$$

$$p = 0.742 \text{ bar}$$

(or)

$$p = 74.2 \text{ KPa}$$

$$\left(\frac{A}{F}\right)_h = 15 \sqrt{\frac{0.991}{1.176}} = 13.75$$



Requirements of A/F ratio under various requirements :-

There are two types of operations

(1) Steady state operation

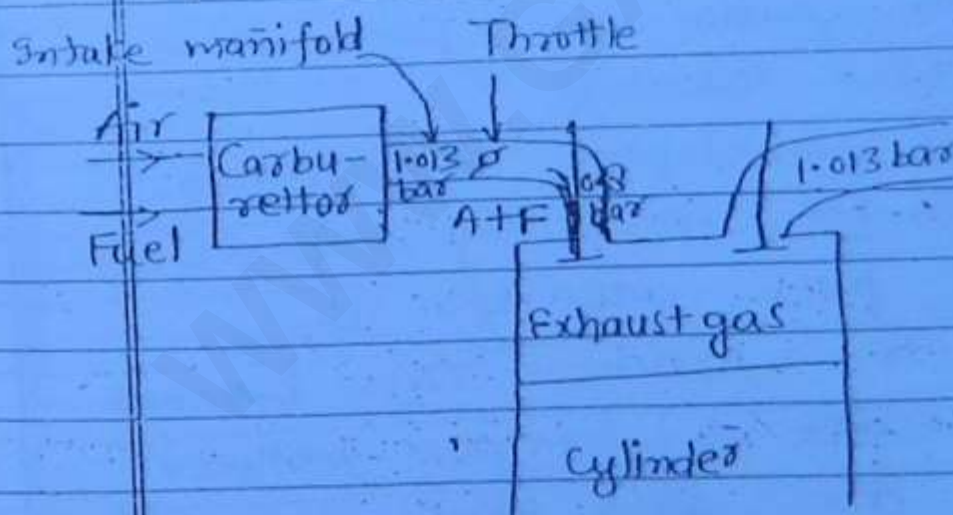
- | | | |
|------------------|----|----|
| a. Idling | SS | CS |
| b. Cruising | SS | CS |
| c. Maximum power | SS | CS |

(2) Transient operation

- | | | | |
|-----------------|----|------|-------|
| a. Starting | LR | 6-10 | 16-12 |
| b. Acceleration | LR | 6-10 | 16-12 |

Idling :-

Idling is no load low speed operation.



Under idling the engine operates at no load and with nearly closed throttle. Under idling case engine requires

rich mixture. The amount of fuel

brought in during idling is much less than during full throttle (96) operation due to very small opening of throttle. This results in much larger proportion of exhaust gas being mixed with fresh charge.

Under idling conditions the pressure in the intake manifold is less than or below atm. pressure due to restriction of air flow.

When the intake valve opens the pressure differential b/w combustion chamber & intake manifold results in initial backward flow of exhaust gases as a result the mixture of Fuel/Air ratio in combustion chamber is diluted.

This results in poor combustion. It is therefore necessary to provide more flow fuel particles by enriching the mixture. This increases the probability of contact between fuel and Air and hence improves combustion.

The general air/Fuel ratio under idling conditions are
 $12 - 12.5 \cdot (\phi > 1)$

Cruising :- (97) (Normal Range)

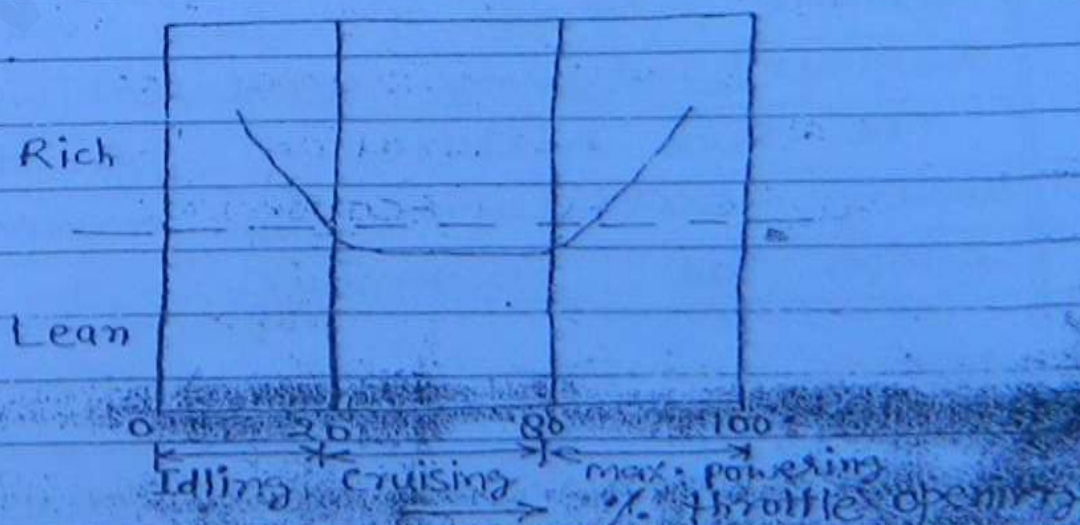
The exhaust gas dilution problem is insignificant in cruising and the primary aim is better fuel economy.

20-80 % of throttle opening operation is known as cruising.

For better fuel economy slightly lean mixture is supply because more oxygen would result in complete combustion of fuel and hence the general Air/Fuel ratios are $16 - 16.5$. ($\phi < 1$)

Maximum power range:

Engine requires during-peak power operation Richer mixture. To provide better power and to prevent overheating of exhaust valve. The general A/F ratios are around 13 . ($\phi > 1$).



(2) Transient Operation

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Starting :- During starting the temp. are low and the fuel remains in liquid form also fuel may condense on coming in contact with cold cylinder walls even though air/fuel ratio is within normal combustion range the ratio of evaporated fuel to air is less and hence during starting very rich mixture must be supplied. (A/F of 3-5).

Acceleration:- The purpose of opening throttle is to provide an increase in torque. When throttle is suddenly open during acceleration the liquid fuels lags behind. And thus cylinder receives lean mixture. While the rich mixture is required during acceleration.

The rich mixture is provided during acceleration by means of a device as Economiser.

1. A venturi of a simple carburettor has a throat dia. of 20 mm (D_2) and fuel orifice dia. of 1.12 mm. The level of petrol surface in the float chamber is 6 mm below the throat. The coefficient of discharge (C_{d_v}) for venturi, and coeff. of fuel orifice are 0.85 & 0.78 resp. specific gravity of petrol is 0.75.

Calculate

- A/F ratio for a pressure drop of 0.08 bar
- Petrol consumption in kg/hr.
- Critical air velocity

The intake conditions are

$$P_1 = 1 \text{ bar} \quad \& \quad T_1 = 290 \text{ K}$$

$$\text{For air } C_p = 1.005 \text{ kJ/kg}\cdot\text{K}$$

$$C_v = 0.718 \text{ kJ/kg}\cdot\text{K}$$

neglect compressibility of air

Soln:-

$$D_2 = 20 \text{ mm} \quad , \quad D_f = 1.12 \text{ mm}$$

$$z = 6 \text{ mm} \quad , \quad C_{d_v} = 0.85 \quad , \quad C_{d_f} = 0.78$$

$$S = 0.75 \quad \Delta P = 0.08$$

$$P_1 = 1 \text{ bar} = 100 \text{ kPa}$$

$$T_1 = 290 \text{ K}$$

$$\rho_a = \frac{P_1}{RT_1}$$

$$= \frac{100}{0.287 \times 290} = 1.201 \text{ kg/m}^3$$

$$P_1 - P_2 = 0.08 \text{ bar}$$

$$100 - P_2 = 8 \text{ kPa}$$

$$P_2 = 92 \text{ kPa}$$

(100)

$$\frac{A}{F} = \frac{C_{d_a} A_2 \sqrt{2 \rho_a (P_1 - P_2)}}{C_{d_f} A_f \sqrt{2 \rho_f [(P_1 - P_2) - \rho_f g z]}}$$

$$= \frac{0.85 \times (20)^2 \sqrt{1.2 (0.08 \times 10^5)}}{0.78 \times (1.12)^2 \sqrt{750 [0.08 \times 10^5 - 750 \times 9.81 (6 \times 10^{-3})]}}$$

$$= \frac{0.85 \times (20)^2 \sqrt{1.2 (0.08 \times 10^5)}}{0.78 \times (1.12)^2 \sqrt{750 [0.08 \times 10^5 - 750 \times 9.81 (6 \times 10^{-3})]}}$$

$$\frac{A}{F} = 13.94$$

$$m_f = C_{d_f} A_f \sqrt{2 \rho_f [(P_1 - P_2) - \rho_f g z]}$$

$$m_f = 0.78 \times \frac{\pi}{4} (1.12 \times 10^{-3})^2 \times$$

$$\sqrt{2 \times 750 [(0.08 \times 10^5) - 750 \times 9.81 \times 6 \times 10^{-3}]}$$

$$m_f = 2.65 \times 10^{-3} \text{ Kg/sec.}$$

$$m_f = 2.5 \text{ kg/hr}$$

$$C_{air} = \sqrt{\frac{2(P_1 - P_2)}{\rho_a}}$$

$$(P_1 - P_2) > \rho_f g z$$

(101)

$$P_1 - P_2 = \rho_f g z$$

$$C = \sqrt{\frac{2 \rho_f g z}{\rho_a}}$$

$$C = \sqrt{\frac{2 \times 750 \times 9.81 \times 6 \times 10^{-3}}{1.2}}$$

$$C_{air} = 8.5 \text{ m/s}$$

2) The following observations has been made from the test of a 4-cylinder two stroke petrol engine.

Diameter 10 cm

Stroke 15 cm

Speed 1600 rpm

Area of positive loop of indicator diagram 5.75 cm²

Area of negative loop of indicator diagram 0.25 cm²

Length of the diagram (l_d)

5.5 cm

Spring constant (k_s)

3.5 bar/cm

Find

Indicated power in kW.

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Mean effective pressure

$$P_m = \frac{k_s a_d}{l_d}$$

where

k_s = Spring constant

a_d = area of indicator diagram

l_d = length of the diagram

$$P_m = \frac{3.5 \text{ bar}}{\text{cm}} \times \frac{(5.75 - 0.25) \text{ cm}^2}{5.5 \text{ cm}}$$

$$P_m = 3.5 \text{ bar}$$

or

$$P_m = 350 \text{ kPa}$$

P_m L A N K

IP =

60

$$350 \times 0.15 \times \frac{\pi}{4} \times (0.1)^2 \times 1600 \times 4$$

=

60

~~IP = 4.48~~

$$P = 43.98 \text{ kW}$$

3) A single cylinder engine running at 1800 rpm develops a Torque of 8 N-m. The indicated power of the engine is 1.8 kW. Find friction power.

$$N = 1800 \text{ rpm}$$

$$IP = 1.8 \text{ kW}$$

(103)

$$\text{Frictional Power} = IP - BP$$

$$BP = \frac{2\pi NT}{60}$$

$$= \frac{2\pi \times 1800 \times 8}{60} \quad \frac{\text{Nm}}{\text{sec}}$$

$$BP = 1508 \text{ W}$$

(or)

$$BP = 1.508 \text{ kW}$$

$$FP = 1.8 - 1.508$$

$$FP = 0.292 \text{ kW}$$

4) Find brake specific fuel consumption in Kg/kWh of a diesel engine whose fuel consumption is 5 gm/sec and power output (BP) is 80 kW. If the mechanical eff^m is 75%. Calculate indicated specific fuel consumption in Kg/kWh.

Solⁿ -

$$BP = 80 \text{ kW}$$

$$\eta_m = 0.75$$

(104)

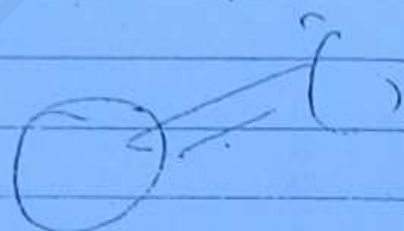
W₂ h₂ -

$$\eta_m = \frac{BP}{IP}$$

(η_m)

$$IP = \frac{BP}{\eta_m} = \frac{80}{0.75} = 106.6 \text{ kW}$$

$$BSFC = \frac{m_f}{BP}$$



$$m_f = \frac{5 \text{ gm}}{\text{sec}} = \frac{5 \times 10^{-3} \text{ kg}}{\frac{1}{3600} \text{ hr.}}$$

$$m_f = 18 \text{ kg/hr.}$$

$$BSFC = \frac{18 \text{ kg/hr.}}{80 \text{ kW}}$$

$$BSFC = 0.225 \frac{\text{kg}}{\text{kW-hr.}}$$

$$ISFC = \frac{m_f}{IP}$$

$$IP = \frac{m_f}{ISFC}$$

$$\eta_m = \frac{BP}{IP} = \frac{m_f / BSFC}{m_f / ISFC} = \frac{ISFC}{BSFC}$$

$$\eta_m = 0.75 = \frac{ISFC}{0.225} \quad (105)$$

$$ISFC = 0.1687 \text{ Kg/Kw-hr.}$$

Prob. ⑤ A 6-cylinder petrol engine operates on 4-stroke cycle. The bore of each cylinder is 80 mm and stroke is 100 mm. The clearance volume per cylinder is 70 cc. At a speed of 4000 rpm the fuel consumption is 20 kg/hr and Torque developed is 150 N-m. Find.

- i. Brake power
- ii. Brake thermal effⁿ. (CV = 43000 KJ/kg)
- iii. Relative effⁿ. based on brake power bases

Assume the cycle to be working on otto cycle and $\gamma = 1.4$

Solⁿ:-

$$D = 80 \text{ mm} \therefore L = 100 \text{ mm}$$

$$= 0.08 \text{ m} \quad L = 0.1 \text{ m}$$

$$V_c = 70 \text{ cc}$$

$$N = 4000 \text{ rpm}$$

$$T = 150 \text{ N-m}, \text{ CV} = 43000 \text{ KJ/Kg}$$

Exel 1

$$BP = \frac{2\pi NT}{60}$$

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$$= \frac{2\pi \times 4000 \times 150 \text{ N-m}}{60 \text{ sec.}}$$

$$BP = 62.83 \times 10^3 \text{ W}$$

(2)

$$BP = 62.83 \text{ KW}$$

$$\eta_{bth} = \frac{BP}{m_f \times \text{CV}}$$

$$= \frac{62.83}{\frac{20 \text{ Kg}}{3600 \text{ sec.}} \times 43000 \frac{\text{KJ}}{\text{Kg}}}$$

$$\eta_{bth} = 0.2628$$

(3)

$$\eta_{bth} = 26.28 \%$$

$$\eta_{rel} = \frac{\eta_{bth}}{\eta_{air std.}}$$

$$\eta_{air std.} = \frac{1}{(r)^{\gamma-1}}$$

$$\gamma = 1 + \frac{V_s}{V_c}$$

$$\gamma = 1 + \frac{\frac{\pi}{4} D^2 L}{V_c}$$

(107)

$$\gamma = 1 + \frac{\frac{\pi}{4} (8)^2 \times 10}{70}$$

$$\gamma = 8.18$$

$$\eta_{\text{air sd.}} = 1 - \frac{1}{(8.18)^{1.4-1}}$$

$$\eta_{\text{air sd.}} = 0.5685$$

$$\eta_{\text{rel.}} = \frac{0.2628}{0.5685}$$

$$\eta_{\text{rel.}} = 0.4622$$

(or)

$$\eta_{\text{rel.}} = 46.22\%$$

Fuel Injection Equations

* *

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C_d

Volume of fuel injection/cycle = $C_d \times \text{Area of orifice} \times \text{velocity of injection} \times \text{time of injection}$

velocity of injection

$$V_f = \sqrt{\frac{2(P_{inj} - P_{cyl})}{\rho_f}}$$

P_{inj} — injection pressure

P_{cyl} — cylinder pressure

Time of injection

$$\omega = \frac{\phi}{t} \quad \begin{matrix} \text{rad/sec} & \text{rad} \end{matrix}$$

$$t = \frac{\phi}{\omega}$$

$$180^\circ = \pi$$

$$90^\circ = \frac{\pi}{2}$$

$$\frac{\pi \theta}{180} \text{ rad.}$$

$$t = \frac{\pi \theta}{180} \times \frac{1}{\frac{2\pi N}{60}}$$

$$t = \frac{\pi \theta \times 60}{180 \times 2\pi N}$$

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$$t = \frac{\theta \times 60}{360 \times N}$$

$$\omega = \frac{\theta}{t}$$

$$t = \frac{\theta \times 60}{360 \times N}$$

here N in rpm.

Q. 1 A single cylinder 4-stroke diesel engine running at 1500 rpm uses 2.5 kg of fuel per hr. The specific gravity of fuel is 0.88. The injection period is 25° crank angle, if the injection pressure is 150 bar and cylinder pressure is 30 bar. Find the diameter of fuel orifice. Take C_d for fuel orifice 0.88.

Soln: 4-stroke
 $N = 1500 \text{ rpm}$
 $m_f = 2.5 \text{ kg/hr.}$
 $\rho_f = 880 \text{ kg/m}^3$
 $\theta = 25^\circ$
 $P_{inj} = 150 \text{ bar}$
 $P_{cyl} = 30 \text{ bar}$
 $C_d = 0.88$

$$\text{Vol. of fuel injection/cycle} = C_d \times A \times V \times t$$

(110)

$$P_f = \frac{m_f}{V_f}$$

$$V_f = \frac{m_f}{P_f} = \frac{2.5 \text{ Kg}}{3600 \text{ sec}} \times \frac{1}{880 \frac{\text{Kg}}{\text{m}^3}}$$

$$V_f = 7.89 \times 10^{-3} \frac{\text{m}^3}{\text{sec.}}$$

for 4-stroke engine

$$\frac{1500}{2} = 750 \text{ cycle/min.}$$

$$\frac{V_f}{\text{cycle}} = \frac{7.89 \times 10^{-3} \text{ m}^3/\text{sec.}}{\frac{750}{60} \frac{\text{cycles}}{\text{sec.}}}$$

$$\text{Volume of fuel injection/cycle} = 6.313 \times 10^{-8} \text{ m}^3/\text{cycle}$$

$$\text{Velocity} = \sqrt{2(P_{inj} - P_{atm})}$$

$$V = \sqrt{\frac{2(150-30) \times 10^5}{880}}$$

(11)

$$V = 165.14 \text{ m/sec.}$$

Time of injection

$$t = \frac{\theta \times 60}{360 \times N}$$

$$t = \frac{25 \times 60}{360 \times 1500}$$

$$t = 2.77 \times 10^{-3} \text{ sec.}$$

$\therefore$

$$6.313 \times 10^{-8} = 0.88 \times A \times 165.14 \times 2.77 \times 10^{-3}$$

$$A = 1.5688 \times 10^{-7} \text{ m}^2$$

$$A = \frac{\pi}{4} d^2 = 1.5688 \times 10^{-7}$$

$$d = 4.4 \times 10^{-4} \text{ m.}$$

(12)

$$d = 0.44 \text{ mm.}$$

- Q. The specific fuel consumption of 4-stroke diesel engine (112) producing 25 kW while running at 3000 rpm is 0.3 kg/kW-hr. If the injection pressure is 160 bar and pressure in the cylinder is 30 bar. Coefficient of velocity is 0.88 and coefficient of discharge is 0.65. Jet dia. is 0.8 mm find crank angle travelled during injection period and take density of fuel as 875 kg/m³.

Soln:-

$$BP = 25 \text{ kW}$$

$$N = 3000 \text{ rpm}$$

$$BSFC = 0.3 \text{ kg/kW-hr}$$

$$P_{inj} = 160 \text{ bar}$$

$$P_{cyl} = 30 \text{ bar}$$

$$C_v = 0.88$$

$$C_d = 0.65$$

$$\text{Jet dia} = 0.8 \text{ mm}$$

$$\rho_f = 875 \text{ kg/m}^3$$

$$V_f = \frac{m_f}{\rho_f}$$

$$= \frac{0.3 \text{ kg/kW-hr}}{875 \text{ kg/m}^3}$$

$$V_f = 9.52 \times 10^{-8} \text{ m}^3/\text{sec.}$$

$$v = C_v \sqrt{\frac{2(P_{inj} - P_{cyl.})}{\rho_f}} \quad (113)$$

$$= 0.88 \sqrt{\frac{2(160 - 30) \times 10^5}{875}}$$

$$V = 151.69 \text{ m/sec.}$$

$$\text{BSFC} = \frac{m_f}{BP}$$

$$m_f = \text{BSFC} \times BP$$

$$m_f = \frac{(BP) \times (\text{BSFC})}{\text{Kwhr}} \quad \text{Kg}$$

$$m_f = 7.5 \text{ Kg/hr.}$$

(or)

$$m_f = \frac{7.5 \text{ Kg}}{3600 \text{ sec.}} = 2.08 \times 10^{-3} \frac{\text{Kg}}{\text{sec.}}$$

$$\rho_f = \frac{m_f}{\text{Vol.}_f}$$

$$\text{Vol.}_f = \frac{m_f}{\rho_f} = \frac{2.08 \times 10^{-3} \text{ Kg/sec.}}{875 \text{ Kg/m}^3}$$

$$n \text{ rpm} = 3000$$

(114)

$$\Rightarrow 1500 \text{ cycles/min} = \frac{1500 \text{ cycle}}{60 \text{ sec}}$$

$$\text{Vol.}_f = \frac{2.38 \times 10^{-6} \text{ m}^3/\text{sec}}{\frac{1500 \text{ cycles/sec}}{60}}$$

$$\text{Vol.}_f = 9.52 \times 10^{-8} \frac{\text{m}^3}{\text{cycle}}$$

$$\text{Area} = \frac{\pi}{4} d^2$$

$$= \frac{\pi}{4} (0.8 \times 10^{-3})^2$$

$$A = 0.502 \times 10^{-6} \text{ m}^2$$

$$\text{Volume of fuel injected/cycle} = C_d \times A \times \text{vel.} \times t$$

$$9.52 \times 10^{-8} = 0.65 \times 0.502 \times 10^{-6} \times 151.69 \times t$$

$$t = 0.00192 \text{ sec.}$$

$$t = \frac{\theta \times 60}{360 \times N}$$

$$\theta \times 60$$

$$0.00192 = \frac{\theta \times 60}{360 \times 3000} \Rightarrow \theta = 34.57^\circ$$

③ When the pressure inside the Combustion chamber is 30 bar and injection pressure is 160 bar. The fuel penetrates distance of 24 cm. in 20 milli sec. Estimate the time taken by the fuel to penetrate the same distance when the injection pressure is changed to 250 bar. other variables remaining same.

Solⁿ:-

$$P_{inj} = 160 \text{ bar}$$

$$P_{cyl} = 30 \text{ bar}$$

(115)

$$\text{Velocity} = \frac{\text{Distance}}{\text{Time}}$$

$$V = \frac{D}{t}$$

$$D = Vt$$

$$\therefore V = \sqrt{\frac{2(\Delta P)}{\rho_f}}$$

$$D \propto (\sqrt{\Delta P})t$$

$$V \propto \sqrt{\Delta P}$$

$$D = K \sqrt{\Delta P} t$$

$$(\sqrt{\Delta P} t)_1 = (\sqrt{\Delta P} t)_2$$

$$\Delta P_1 = P_{inj} - P_{cyl} = 160 - 30$$

$$\Delta P_2 = 250 - 30$$

$$\Delta P_2 = 250 - 30 = 220 \text{ bar}$$

$$\frac{t_2}{t_1} = \frac{\sqrt{\Delta P_1}}{\sqrt{\Delta P_2}} \quad (1/6)$$

$$t_2 = t_1 \sqrt{\frac{\Delta P_1}{\Delta P_2}}$$

$$t_2 = 20 \sqrt{\frac{130}{220}}$$

$$t_2 = 15.37 \text{ milli sec.}$$

A 6-cylinder, 4-stroke diesel engine develops 100 kW at 3000 rpm when Brake thermal $\eta_b = 25\%$.

Calculate quantity of fuel injected per cylinder per cycle.

Also calculate the fuel orifice dia. if the injection pressure is 160 bar, pressure inside the combustion chamber is 50 bar and crank travel during injection period is 25°.

Take C_d for fuel orifice 0.65

$$d_f = ?$$

$$V_f = 7.47 \times 10^{-8} \text{ m}^3 / \text{cycle/cylinder}$$

4-stroke diesel engine

$$K = 6$$

$$BP = 100 \text{ kW}$$

$$N = 3000 \text{ rpm}$$

$$\eta_{bth} = 25\%$$

(117)

$$P_{inj} = 160 \text{ bar}$$

$$P_{cyl} = 50 \text{ bar}$$

$$\theta = 25^\circ$$

$$CV = 42000 \text{ kJ/kg}$$

$$\rho_f = 850 \text{ kg/m}^3$$

$$\eta_{bth} = \frac{B.P.}{m_f \times CV}$$

$$m_f = \frac{B.P.}{\eta_{bth} \times CV}$$

$$m_f = \frac{100 \text{ kW}}{0.25 \times 42000 \text{ kJ/kg}}$$

$$m_f = 9.523 \times 10^{-3} \text{ kg/sec}$$

$$\therefore V_f = \frac{m_f}{\rho_f}$$

$$= \frac{9.523 \times 10^{-3} \text{ kg/sec}}{850 \text{ kg/m}^3}$$

$$V_f = 1.12 \times 10^{-5} \text{ m}^3/\text{sec}$$

and

$$V = \sqrt{\frac{2 (P_{inj} - P_{cyl})}{\rho_f}}$$

$$= \sqrt{\frac{2 (160 - 50) \times 10^5}{850}} \quad \text{118}$$

$$V = 160.87 \quad \text{m/s.}$$

$$rpm = 3000 \text{ rpm}$$

$$\therefore \frac{3000}{60 \times 2} = \frac{1500}{60} \text{ cycle/sec.}$$

$$\frac{V_f}{\text{cycle}} = \frac{1.12 \times 10^{-5}}{\frac{1500}{60}} \text{ m}^3/\text{sec.}$$

$$= 4.48 \times 10^{-7} \text{ m}^3/\text{sec.}$$

$$4.48 \times 10^{-7} = C_d \times A \times V_{el.} \times t$$

$$t = \frac{0 \times 60}{360 \times N} = \frac{25 \times 60}{360 \times 3000}$$

$$= 1.38 \times 10^{-3} \text{ sec.}$$

$$4.48 \times 10^{-7} = 0.65 \times A \times 160.87 \times 1.38 \times 10^{-3}$$

$$\frac{A \pi D^2}{4} = \frac{4.48 \times 10^{-7}}{0.65 \times 160.87 \times 1.38 \times 10^{-3}}$$

$$D = 2.5 \text{ mm}$$

Engine Emission & Controls

The emissions are

CO_2

CO

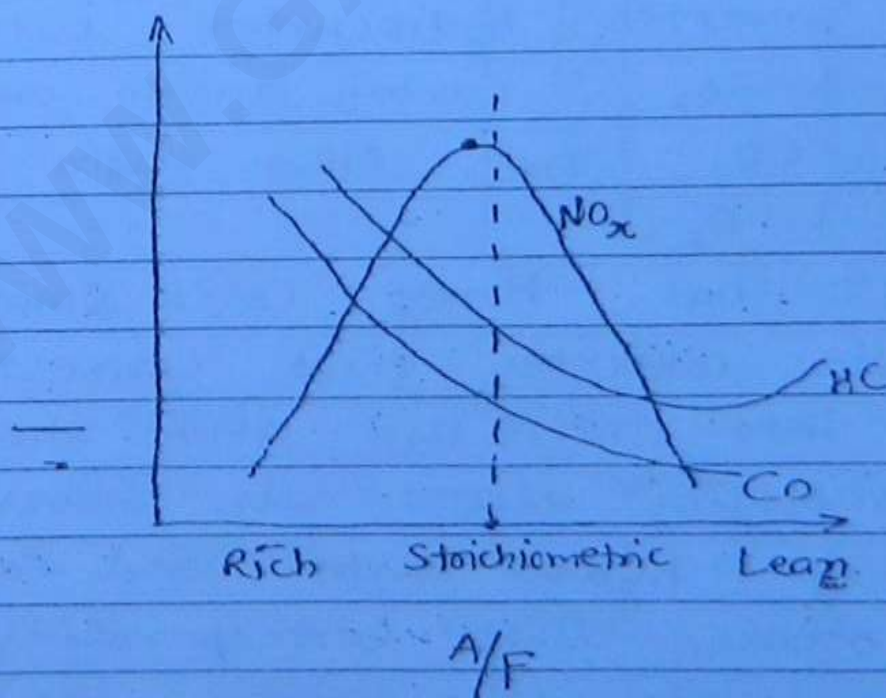
Unburnt HC

Oxides of Nitrogen (NO_x)

119

Of these Carbon mono oxide, hydrocarbons & oxides of Nitrogens are main pollutants.

Oxide of N_2 are formed when the temp.s are very-very high and when there is sufficient availability of oxygen.



Emission Control :-

Change in engine design so as to keep lower compression ratios.

Slightly leaner mixture must be provided.

120

An addition of after burner to the exhaust can completely burn partially burnt Hydro Carbons in exhaust gases.

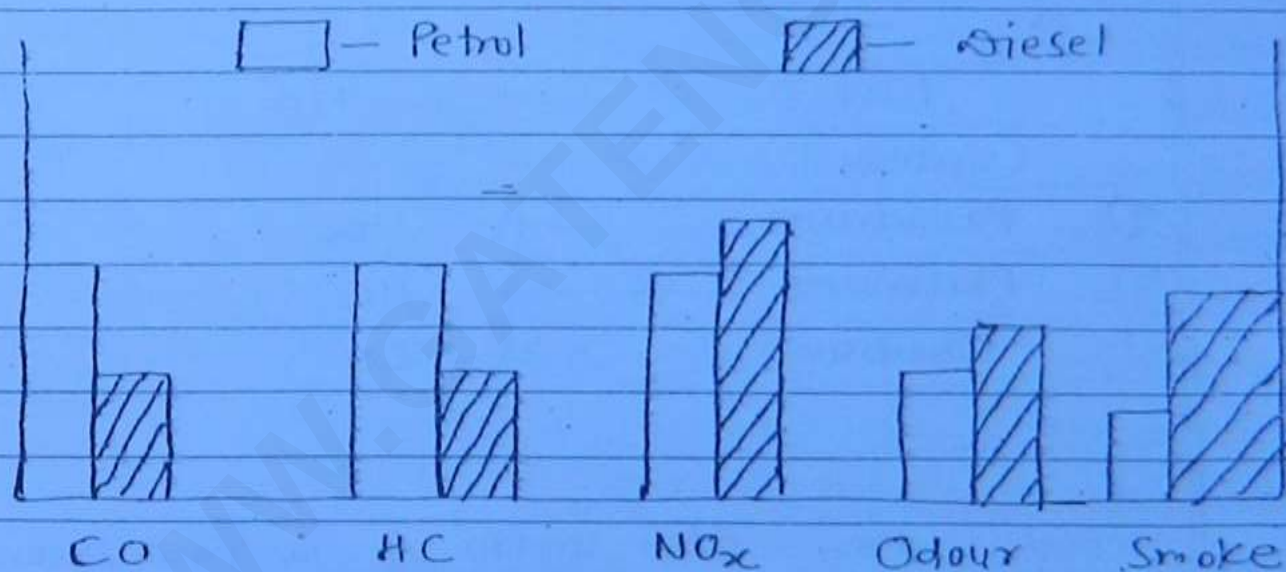
Catalytic Converter consists of Platinum, Palladium & Rhodium and it converts hydrocarbon into H_2O & CO_2 , Carbon mono oxide into CO_2 and NO_x into N_2 & O_2 .

It has Honey Comb Structure. Catalytic converter first converts NO_x into N_2 & O_2 and then by supplying excess air conversion of Hydro Carbon and Carbon monoxide into corresponding oxides takes place.

Drawbacks of Catalytic Converter:-

(12)

- * The exhaust temp. are more when catalytic converter is fitted.
- * If lead is used in fuel this lead attacks the converter and hence decreases conversion efficiency.
- * If sulphur is present in fuel catalytic converter converts sulphur into oxides of sulphur and this may result in acid rain.



Diesel engines if properly maintain have very little CO in their exhaust. And a small amount of smoke. While petrol engine exhaust significant amount of CO and unburnt HC. So, diesel engine is cleaner as compared to petrol engine and pollute less.

However this fact is not generally noted because a small increase in smoke becomes highly visible. while CO and Unburnt HC can not be noticed so easily.

122

match the following

List-I

(Catalyst)

A) Palladium

B) Platinum

C) Rhodium

List-II

1) NO_x

2) HC

3) CO

Find the A/F ratio of a Four stroke single cylinder engine with fuel consumption time for 10 cc is 20.4 sec. and air consumption time for 0.1 m^3 is 16.3 sec.

Assume density of air as 1.175 kg/m^3 and specific gravity of fuel 0.7

Solⁿ:

$$V_f = 10 \text{ cc} / 20.4 \text{ sec.}$$

$$V_a = 0.1 \text{ m}^3 / 16.3 \text{ sec.}$$

(123)

$$\rho_a = m_a / V_a$$

$$m_a = \rho_a V_a$$

$$= 1.175 \frac{\text{kg}}{\text{m}^3} \times \frac{0.1 \text{ m}^3}{16.3 \text{ sec.}}$$

$$m_a = 7.2 \times 10^{-3} \text{ kg/s}$$

$$\rho_f = \frac{m_f}{V_f}$$

$$m_f = \rho_f \times V_f$$

$$= \frac{700 \times 10 \times 10^{-6}}{20.4}$$

$$m_f = 0.34 \times 10^{-3}$$

$$\therefore \frac{A}{F} = \frac{m_a}{m_f}$$

$$= \frac{7.2 \times 10^{-3}}{0.34 \times 10^{-3}}$$

$$\frac{A}{F} = 21$$

③

An 8-cylinder, Four stroke engine of 9 cm Bore and 8 cm stroke with a compression ratio of 7 is tested at 4500 rpm on a dynamometer which has 54 cm radius.

(124)

During a 10 min ^{beam} test the dynamometer scale reading was 42 Kg and the engine consume 4.4 Kg of fuel having a calorific value of 44000 KJ/Kg. Air at 27°C and 1 bar was supplied to carburetor at the rate of 6 Kg/min. Find

- i. Brake power
- ii. Brake sp. fuel consumption
- iii. Brake sp. Air consumption
- iv. Brake thermal effⁿ.
- v. Volumetric effⁿ &
- vi. A/F ratio

Soln:-

4-stroke

$$K = 8$$

$$D = 9 \text{ cm}$$

$$L = 8 \text{ cm}$$

$$r = 7$$

$$N = 4500 \text{ rpm}$$

$$\text{Dyna. radius} = 54 \text{ cm}$$

$$t = 10 \text{ min}$$

$$m = 42 \text{ kg}$$

$$m_f = 4.4 \text{ kg}$$

$$m_f = \frac{4.4}{10} \text{ kg/sec}$$

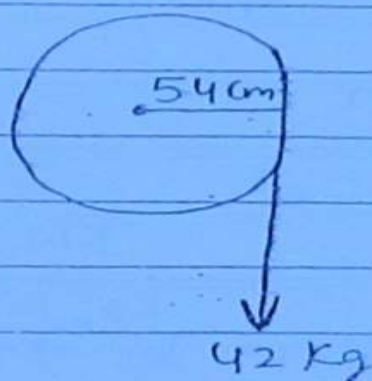
$$CV = 44000 \text{ kJ/kg}$$

$$T_1 = 300\text{K}$$

$$P_1 = 1 \text{ bar}$$

$$m_a = 6 \text{ kg/min}$$

(125)



$$T = F \times r$$

$$= 42 \times 9.81 \times 0.54$$

$$= 222.49$$

$$42 \text{ kg}$$

$$F = 42 \times 9.81$$

$$BP = \frac{2\pi NT}{60}$$

$$= \frac{2\pi \times 4500 \times 222.49}{60}$$

$$= 104.8 \times 10^3 \text{ W}$$

(2)

$$BP = 104.8 \text{ kW}$$

BSFC

$$BSFC = \frac{m_f}{B.P.}$$

m_f in kg/hr.

$$= \frac{4.4 \times 60}{104.8 \times 10} = 0.251$$

$$m_f = \frac{4.4 \times 60}{10}$$

$$= 0.251 \text{ kg/kwhr}$$

(iii)

$$\begin{aligned}
 BSAC &= \frac{m_a}{B.P.} \\
 &= \frac{6 \times 60}{104.8} \quad (126) \\
 &= 0.0572 \times 60 \\
 BSAC &= 3.435 \text{ Kg/kwhr.}
 \end{aligned}$$

(iv)

Brake Thermal effⁿ.

$$\begin{aligned}
 \eta_{bth} &= \frac{B.P.}{m_f \times C.V.} \quad \frac{KW}{\frac{Kg}{Sec} \times \frac{KJ}{Kg}} \\
 &= \frac{104.8 \text{ KW}}{\frac{4.4 \text{ Kg}}{10 \times 60 \text{ Sec}} \times 44000 \frac{KJ}{Kg}}
 \end{aligned}$$

$$\eta_{bth} = 0.3248$$

(or)

$$\eta_{bth} = 32.48 \%$$

(v)

$$\frac{A}{F} = \frac{m_a}{m_f}$$

$$\begin{aligned}
 &= \frac{6 \text{ Kg/min.}}{4.4 \text{ Kg/min.}} \\
 &= 1.36
 \end{aligned}$$

$$\frac{A}{F} = 13.63$$

$$(127)$$

(vi) Volumetric effⁿ.

$$\eta_v = \frac{\text{Actual volume of Air/cycle} = V_a}{\text{Swept volume} = V_s}$$

$$\rho_a = \frac{m_a}{V_a}$$

$$V_a = \frac{m_a}{\rho_a}$$

$$P = \rho R T$$

$$\rho_a = \frac{P_a}{R_a T_a} = \frac{100}{0.287 \times 300}$$

$$\rho_a = 1.1614 \text{ Kg/m}^3$$

$$V_a = \frac{6 \text{ Kg/min}}{1.1614 \text{ Kg/m}^3}$$

$$V_a = 5.166 \text{ m}^3/\text{min.}$$

For 4-st engine

$$4500 \text{ rpm} = \frac{4500}{2} = 2250 \text{ cycle/min}$$

$$\begin{aligned} V_a / \text{cycle} &= \frac{5.166 \text{ m}^3/\text{min}}{2250 \text{ cycle/min}} \\ &= 2.296 \times 10^{-3} \text{ m}^3/\text{cycle} \end{aligned}$$

$$\begin{aligned} V_s &= \frac{\pi}{4} D^2 L K \cdot (128) \\ &= \frac{\pi}{4} (0.09)^2 (0.08) 8 \text{ m}^3 \\ &= 4.07 \times 10^{-3} \text{ m}^3 \end{aligned}$$

$$\eta_{\text{vol}} = \frac{V_a}{V_s}$$

$$= \frac{2.296 \times 10^{-3}}{4.07 \times 10^{-3}}$$

$$\eta_{\text{vol}} = 0.5639$$

(or)

$$\eta_{\text{vol}} = 56.39 \%$$

Note: The volumetric effⁿ. of above engine is very low and volumetric effⁿ. of a well design engine is in the range of 75 - 85 %

(129)

Gate
(4)

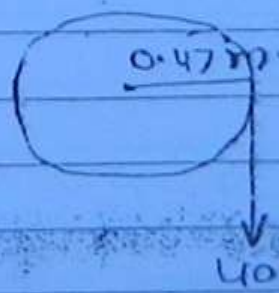
The following details were noted in a test on a 4-cylinder, 4-stroke engine

Bore	100 mm
Stroke	120 mm
Speed	1600 rpm
Fuel Consumption	0.2 Kg/min
C.V. of fuel	44000 KJ/Kg
Brake load	40 Kg
Brake circumference	300 cm (or) 3 m
η_m	80 %

Calculate

- Brake thermal effⁿ.
- Indicated thermal effⁿ.
- Indicated mean effective pressure
- Brake specific Fuel Consumption.

Soln:-



$$F = 9.81 \times 40 \\ = 392.4$$

$$\text{Circumference} = 2\pi r \\ 3 = 2\pi r \\ r = 0.477 \text{ m}$$

$$T = F \times r$$

$$= 392.4 \times 0.477$$

$$= 187.35$$

(130)

$$\text{Brake power} = \frac{2\pi NT}{60}$$

$$= \frac{2\pi \times 1600 \times 187.35}{60}$$

$$BP = 31392 \text{ W}$$

(22)

$$BP = 31.39 \text{ kW}$$

i) Brake thermal effⁿ.

$$\eta_{bth} = \frac{BP}{m_f \times C \cdot V}$$

$$= \frac{31.39 \text{ kW}}{0.2 \frac{\text{kg}}{60 \text{ sec}} \times 44000 \frac{\text{KJ}}{\text{kg}}}$$

$$\eta_{bth} = 0.2156$$

(23)

$$\eta_{bth} = 21.56 \%$$

(ii) Indicated Thermal effⁿ.

$$\eta_{ith} = \frac{\eta_{bth}}{\eta_m}$$

(131)

$$= \frac{0.2156}{0.8}$$

$$\eta_{ith} = 0.269$$

(or)

$$\eta_{ith} = 26.9 \%$$

(iii) Indicated mean effective pressure

$$IP = \frac{P_m \text{ IANK}}{2 \times 60}$$

$$\eta_m = \frac{BP}{IP}$$

$$0.8 = \frac{31.36}{IP}$$

$$IP = 39.24 \text{ kW}$$

$$39.24 = \frac{P_m \times (0.12) \times \frac{\pi}{4} (0.1)^2 \times 1600 \times 4}{60 \times 2}$$

$$P_m = 780.65 \text{ kPa}$$

(or)

$$P_m = 780.65 \times 10^3 \text{ N/m}^2$$

(or)

$$P_m = 7.8065 \text{ bar}$$

(iv)

Brake Specific Fuel Consumption

$$BSFC = \frac{m_f}{B.P.}$$

(132)

$$= \frac{0.2 \times 60}{31.39} \frac{\text{kg}}{\text{kW hr.}}$$

$$BSFC = 0.3826 \text{ kg/kW hr.}$$

⑤ The power output of a 6-cylinder, 4-stroke engine by measure means of a dynamometer for which the law is

$$b.p. = \frac{WN}{2000} \text{ kW}$$

where

W = brake load in N

N = speed in rpm

The following data may be used

Throat diameter	30 mm
Bore	100 mm
Stroke	120 mm
Brake load	560 N
Speed	2000 rpm

$\frac{C}{H}$ by mass

(1)

83

17

133

Coeff. for discharge for venturi

0.6

atmp. pressure

1 bar

pressure drop across venturi

14.5 cm of H₂O

Time taken for 100 cc fuel

Consumption

20 sec.

atmp. Temp.

27°C

Fuel density

831 kg/m³

calculate

(i) Brake power

(ii) Brake Torque

(iii) BSFC

(iv) Volumetric eff.

(v) Percentage excess of air

Soln:- * Assume Air as incompressible

(i) Brake power

$$bP = \frac{WN}{20000}$$

$$bP = \frac{560 \times 2400}{20000}$$

$$bP = 67.2 \text{ kW}$$

(ii) Brake Torque

$$bP = \frac{2\pi NT}{60} \text{ kW}$$

$$67.2 \times 10^3 = \frac{2\pi \times 2400 \times T}{60}$$

$$T = \frac{67.2 \times 10^3 \times 60}{2\pi \times 2400}$$

$$T = 267.3 \text{ N-m}$$

(iii) ~~8.87%~~ % excess of Air

$$\left(\frac{A}{F}\right)_{st.} = \frac{100}{23} \left(\frac{8}{3} C + 8H_2 \right)$$

$$\left(\frac{A}{F}\right)_{st.} = \frac{100}{23} \left[\frac{8}{3} \times 0.83 + 8 \times (0.17) \right]$$

$$\left(\frac{A}{F}\right)_{st.} = 15.536$$

Actual Air/Fuel ratio

$$V_f = 100 \text{ cm}^3 / 20 \text{ sec.}$$

$$\rho_f = 831 \text{ kg/m}^3$$

$$P = \frac{m_f}{V_f}$$

$$V_f = 15$$

$$m_f = \rho_f \times V_f$$

$$= \frac{831 \text{ kg}}{\text{m}^3} \times \frac{100 \times 10^{-6} \text{ m}^3}{20 \text{ sec.}}$$

(135)

$$m_f = 4.155 \times 10^{-3} \text{ kg/sec.}$$

$$m_a = C_{da} A_2 \sqrt{2 \rho_a (P_1 - P_2)}$$

$$(P_1 - P_2) = 14.5 \text{ cm. of Hg}$$

$$P = \rho R T$$

$$\rho_a = \frac{P}{R T}$$

$$P = \rho g h$$

$$\rho_a = \frac{100}{0.287 \times 300} \quad (P_1 - P_2) = 13.6 \times 10^3 \times 9.81 \times \frac{14.5}{100}$$

$$\rho_a = 1.16$$

$$P_1 - P_2 = 19345.3 \text{ N/m}^2$$

$\therefore$

$$m_a = 0.6 \times \frac{\pi}{4} (0.03)^2 \times \sqrt{2 \times 1.16 (19345.3)}$$

$$m_a = 0.089 \text{ kg/sec.}$$

$$\left(\frac{A}{F} \right)_{act.} = \frac{0.089}{4.155 \times 10^{-3}}$$

$$= 21.62$$

$$\% \text{ Excess} = \frac{\left(\frac{A}{F}\right)_{\text{act.}} - \left(\frac{A}{F}\right)_{\text{st.}}}{\left(\frac{A}{F}\right)_{\text{st.}}}$$

$$(136)$$

$$\% \text{ excess} = \frac{21.62 - 15.536}{15.536}$$

$$\% \text{ excess} = 0.391$$

(a)

$$\% \text{ excess} = 39.1 \%$$

(iv) Brake specific fuel consumption (BSFC)

$$\text{BSFC} = \frac{m_f}{bP}$$

$$\text{BSFC} = \frac{4.155 \times 10^{-3} \times 3600}{67.2}$$

$$\text{BSFC} = 0.222 \text{ Kg/kwhr.}$$

(v) Volumetric effⁿ.

$$\eta_v = \frac{V_a / \text{cycle}}{V_s}$$

$$\rho_a = \frac{m_a}{V_a}$$

(13%)

$$V_a = \frac{m_a}{\rho_a}$$

$$V_a = \frac{0.089 \text{ kg/sec.}}{1.16 \text{ kg/m}^3}$$

$$V_a = 0.0766 \text{ m}^3/\text{sec.}$$

For 4-s engine 2400 rot./min.

$$= \frac{2400}{2} = 1200 \text{ cycle/min.}$$

$$\frac{V_a}{\text{cycle}} = \frac{0.0766 \text{ m}^3/\text{sec.}}{\frac{1200}{60} \text{ cycle/sec.}}$$

$$\frac{V_a}{\text{cycle}} = \frac{60 \times 0.0766 \text{ m}^3}{1200}$$

$$V_a/\text{cycle} = 3.8315 \times 10^{-3} \text{ m}^3$$

$$V_s = \frac{\pi}{4} D^2 L K = \frac{\pi}{4} (0.1)^2 (0.12) \times 6$$

$$V_s = 5.652 \times 10^{-3} \text{ m}^3$$

$$\eta_v = \frac{V_a}{V_s}$$

(138)

$$= \frac{3.8315 \times 10^{-3}}{5.652 \times 10^{-3}}$$

$$\eta_v = 0.6877$$

Ans

$$\eta_v = 68.77\%$$

Q. A 6-cylinder, 6-stroke engine delivers a brake power of 120 kW at 1600 rpm. The fuel to be used has a calorific value of 43000 kJ/kg. And its percentage composition by mass is C = 86%, H = 13% & 1% non-combustibles. The volumetric effⁿ assume to be 80%. Indicated thermal effⁿ is 40%. mechanical effⁿ 80%. The air consumption is 110 excess with resp. to stoichiometric conditions.

(i) Estimate volumetric composition of dry exhaust gases.

(ii) Determine bore & stroke of

the engine. Taking Stroke to Bore ratio $L/D = 1.5$.

Assume density of air as $1.298 \frac{\text{kg}}{\text{m}^3}$

and air contains 23 Kg of oxygen in 100 Kg.

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Soln.

4-Stroke

$$K = 4$$

$$N = 1600 \text{ rpm}$$

$$b.p. = 120 \text{ kW}$$

$$CV = 43000 \text{ KJ/kg}$$

$$\%C = 0.86, \quad \%H_2 = 0.13$$

$$\eta_{vol.} = 0.8$$

$$\eta_{ith.} = 0.4$$

$$\eta_m = 0.8$$

$$\left(\frac{A}{F}\right) = 110\% \text{ excess}$$

$$\rho_a = 1.298 \text{ kg/m}^3$$

$$\frac{L}{D} = 1.5$$

Stoichiometric A/F ratio

$$\left(\frac{A}{F}\right)_{st.} = \frac{100}{23} \left[\frac{8}{3} (0.86) + 8 (0.13) \right]$$

$$\left(\frac{A}{F}\right)_{st.} = 14.5$$

$$\left(\frac{A}{F}\right)_{act.} = 14.5 + 1.1(14.5)$$

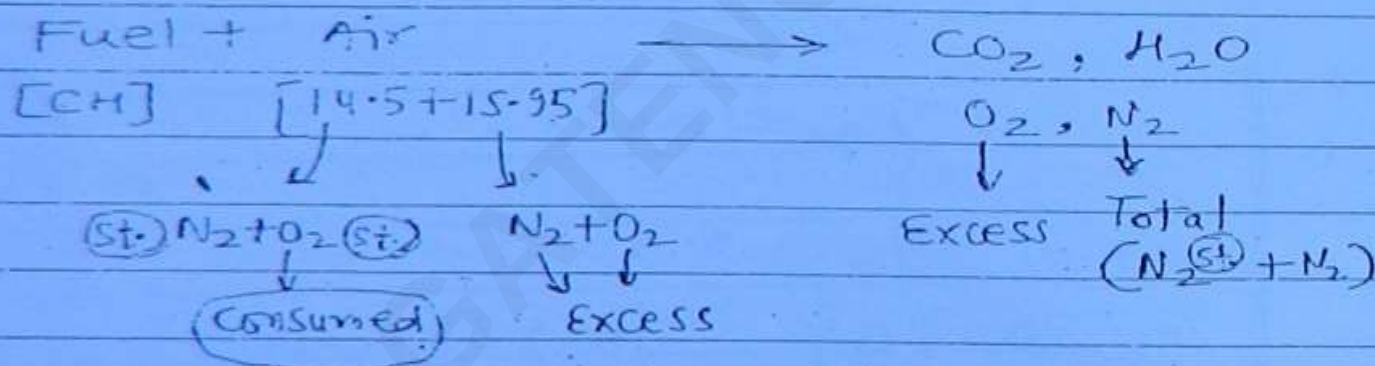
$$\left(\frac{A}{F}\right)_{act.} = 30.45$$

140

Excess air supply per kg fuel

$$= 30.45 - 14.5$$

$$= 15.95$$



ie: As excess air is used stoichiometric oxygen will react with fuel resulting in CO₂ & H₂O whereas excess O₂ will come out in exhaust gases. as N₂ is inert it does not undergo any reaction and hence stoichiometric N₂ & ~~excess~~ and excess N₂ will come out in exhaust gases.

$$\eta_m = \frac{b.p.}{i.p.}$$

$$i.p. = \frac{b.p.}{\eta_m} \quad (141)$$

$$= \frac{1.20 \text{ kW}}{0.8}$$

$$i.p. = 150 \text{ kW}$$

$$\eta_{ith} = \frac{i.p.}{m_f \times C.V.}$$

$$0.4 = \frac{150 \text{ kW}}{m_f \times 43000 \frac{\text{kJ}}{\text{kg}}}$$

$$m_f = 8.72 \times 10^{-3} \text{ kg/sec.}$$

For actual

$$\frac{m_a}{m_f} = 30.45$$

$$m_a = 30.45 \times m_f$$

$$m_a = 30.45 \times 8.72 \times 10^{-3}$$

$$m_a = 0.2655 \text{ kg/sec.}$$

$$\eta_{vol} = \frac{V_a / \text{cycle}}{V_s}$$

(142)

$$\rho_a = \frac{m_a}{V_a}$$

$$V_a = \frac{m_a}{\rho_a} = \frac{0.2655 \text{ Kg/sec}}{1.298 \text{ Kg/m}^3}$$

$$V_a = 0.2045 \text{ m}^3/\text{sec}$$

For 4-stroke engine

$$1600 \text{ rpm} \Rightarrow 800 \text{ cycle/min}$$

$$= \frac{800 \text{ cycle}}{60 \text{ sec}}$$

$$\frac{V_a}{\text{cycle}} = \frac{0.2045 \text{ m}^3/\text{sec}}{\frac{800 \text{ cycle}}{60}}$$

$$\frac{V_a}{\text{cycle}} = 0.0153 \text{ m}^3$$

$$0.8 = \frac{0.0153 \text{ m}^3}{\frac{\pi}{4} D^2 L K \text{ m}^3}$$

$$\frac{\pi D^2 L K}{4} = \frac{0.0153}{0.8}$$

$$\frac{\pi}{4} D^2 (1.5D) \times 6 = \frac{0.0153}{0.8}$$

$$D = 0.139 \text{ m.}$$

(143)

(09)

$$D = 13.9 \text{ cm.} \approx 14 \text{ cm.}$$

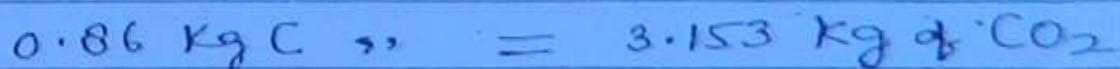
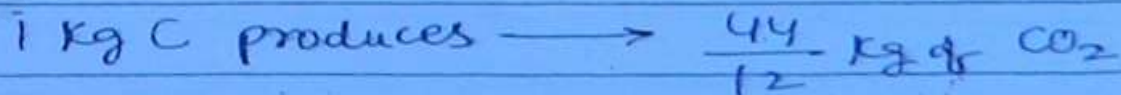
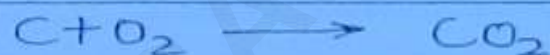
$$L = 1.5D$$

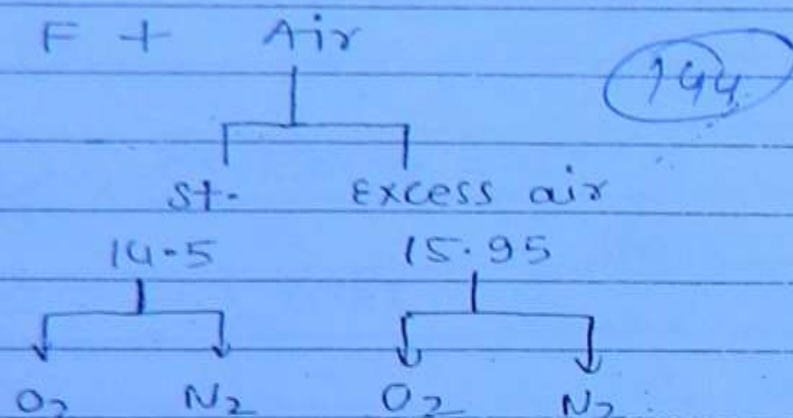
$$= 1.5 \times 14$$

$$L = 21 \text{ cm.}$$

Exhaust Gases,

$\text{CO}_2, \text{N}_2, \text{O}_2$





100 Kg of air — 23 Kg of O₂

15.95 Kg " — $\frac{15.95 \times 23}{100}$

15.95 Kg Air = 3.6685 Kg of O₂

100 Kg air — 77 Kg of N₂

30.45 — " → $\frac{77 \times 30.45}{100}$

30.45 Kg Air = 23.446 Kg of N₂

The above analysis is known as
Gravimetric (or) Mass analysis

CO₂ — 3.153

O₂ — 3.668

N₂ — 23.446

Volumetric Analysis of Exhaust Gases:-

	Grav.	Volum.
CO ₂	3.153	$3.153/44 = 0.7165$
O ₂	3.668	$3.668/32 = 0.1145$
N ₂	23.446	$23.446/28 = 0.8373$
		1.023

(145)

$$\frac{0.07165}{1.023} \times 100 = 7\%$$

$$\frac{0.1145}{1.023} \times 100 = 11.2\%$$

$$\frac{0.8373}{1.023} \times 100 = 81.8\%$$

Volumetric Analysis of Exhaust Gases :-

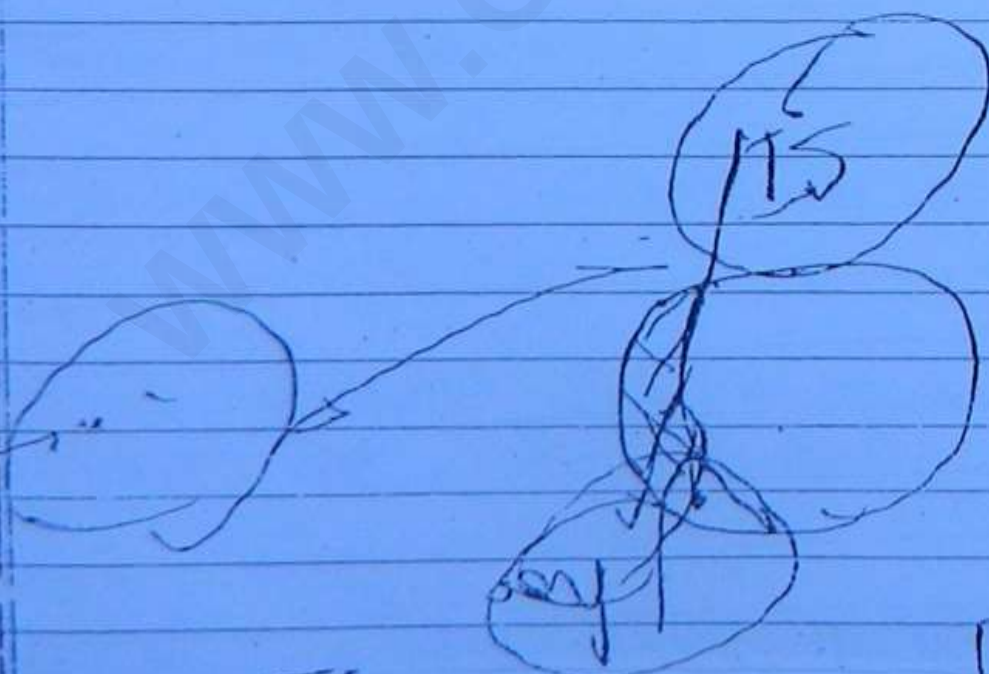
	Grav.	Volum.
CO ₂	3.153	$3.153/44 = 0.7165$
O ₂	3.668	$3.668/32 = 0.1145$
N ₂	23.446	$23.446/28 = 0.8373$
		1.023

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$$\frac{0.07165}{1.023} \times 100 = 7\%$$

$$\frac{0.1145}{1.023} \times 100 = 11.2\%$$

$$\frac{0.8373}{1.023} \times 100 = 81.8\%$$

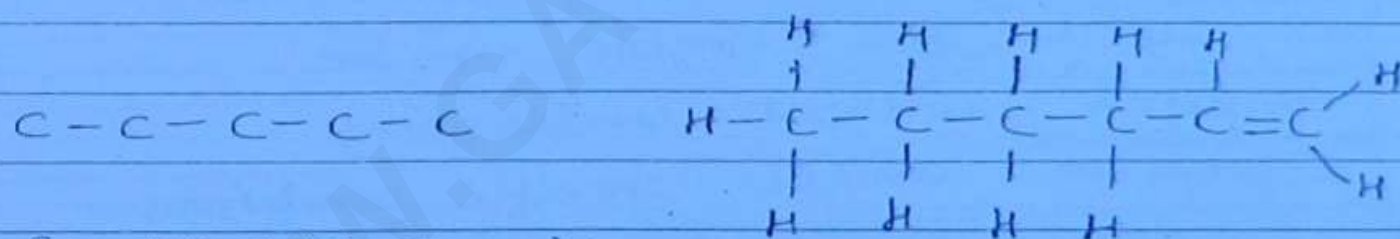


Fuels

(147)

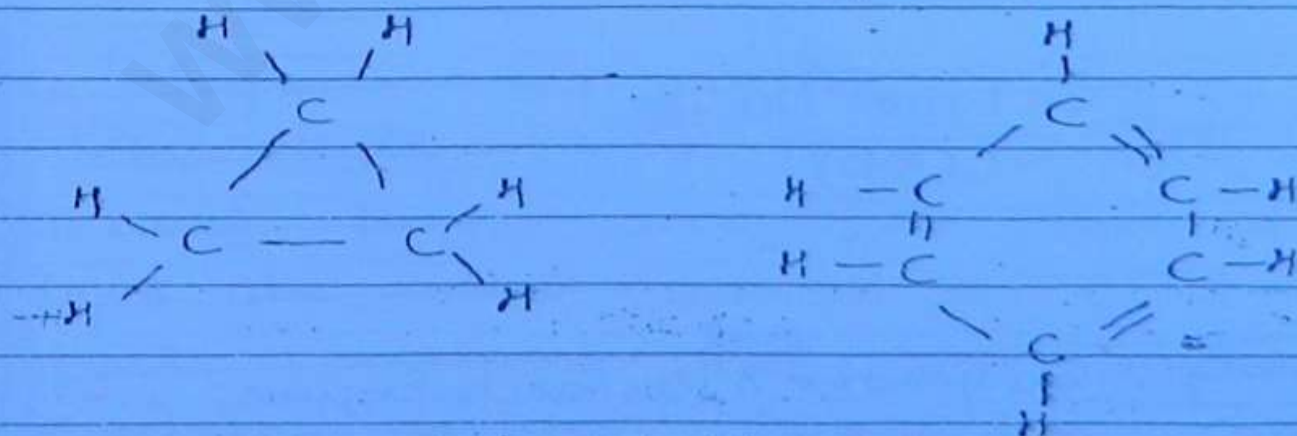
The main constituents are of fuel

1. Paraffins (C_nH_{2n+2}), they are straight chain compounds
2. Olefins (C_nH_{2n}), They are straight chain unsaturated compounds
3. Naphthenes (C_nH_{2n}), They are cyclic saturated compounds.
4. Aromatics (C_nH_{2n-6}), They are cyclic unsaturated compounds.



Paraffins (C_nH_{2n+2})

Olefins (C_nH_{2n})



Naphthenes (C_nH_{2n}) Aromatics (C_nH_{2n-6})

- * Paraffins have high knocking tendency when use in S.I. engine whereas aromatics have least knocking tendency in S.I. engine fuel.
- * alifins form gummy deposits and are readily oxydised on storage and hence these contain are kept to minimum.

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Highest Useful Compression ratio:-
(HUCR) It is the highest useful compression ratio at which the engine can be operated without undergoing detonation under specific conditions.

Octane Number (ON)

- * Initially ^{Fuel} Cooperative Research Engine (CFR) operated on test fuel and the compression ratio is adjusted such that detonation

occurs at say Top dead centre.

This compression ratio is fixed and now the engine operates on mixture of 98-octane & η -heptane.

The percentage by volume of 98-octane at which detonation again starts at TDC is known as Octane Number (ON).

Octane no. is used for Knock Rating of S.I. engine fuels. 98-octane has very little knocking tendency hence its octane number is arbitrarily assigned as 100. whereas η -heptane detonates easily. And hence its octane number is assigned as 0 (zero).

Higher Octane no. means better S.I. engine fuel.

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Dopes in S.I. engine fuels

To improve anti-knocking properties Tetra ethyle lead, Tetra methyle lead & Tertiary butyle accetylene. When these dopes are added, octane no. of S.I. engine fuels increases and hence anti-knocking properties is also improves.

Note:- When lead is used it forms deposits on spark plug and hence to remove this deposits Ethylene di-bromide is used. Sometime;

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Motor Octane number
(MON)

It measures anti-knocking performance under severe condition like large loads and high speeds.

Research Octane Number
(RON)

It measures anti-knocking performance under mild conditions like less load and low speed.

$$*** \text{ Fuel Sensitivity} = \text{RON} - \text{MON}$$

$$*** \text{ Antiknock Index} = \frac{\text{RON} + \text{MON}}{2}$$

(157)

Cetane Number

(or)

Knock Rating of C.I. engine fuels

The reference fuels used in C.I. engine

* Cetane (η -hexadecane), ($\text{C}_{16}\text{H}_{34}$) (CN=100)
Straight chain compound --

* α -methyl naphthalene (CN=0)

* hepta methyl nonane (CN=15)

$$* \text{ Cetane No.} = 60 - \frac{\text{Octane No.}}{2}$$

Note: Higher Cetane no. means lower octane no. and hence a good C.I. engine fuel is a bad S.I. engine fuel.

Dopes in C.I. engine fuels

- * Primary amyl nitrate
- * Secondary amyl nitrate
- * Butyl peroxide

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Alternate fuels

- * Alcohol
- * Hydrogen
- * CNG
- * Biogas
- * LPG etc.

Alcohol

Advantages:-

1. High octane No. (high compression ratio can be used)
2. It can be obtained from no. of sources.
3. Less emissions compared with petrol.
4. Low sulphur content.

Disadvantages:-

1. Low calorific value
2. Ignition characteristics are poor
3. As alcohol is less volatile starting characteristics are poor. when refueling automobile headaches have been experienced.

Hydrogen

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Advantages:-

1. Low emissions.
2. Easily available
3. Hydrogen burns faster
4. Fuel leakage to environment is not a pollutant.

Disadvantages:-

1. Handling of hydrogen is difficult and storage requires high capital & running cost.
2. Difficult to refuel
3. High fuel cost
4. Problem of detonation.
5. High knock emission, due to high temperatures.

CNG

(or)

Compressed Natural Gas

It contains 90% Ethane & 4% of methane.

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Advantages:-

1. High octane no. and hence it
2. is good S.I. engine
2. Low engine emissions
3. Low cost.
4. Safe in operation.
5. It is odourless
6. Fuel availability is very large.

Disadvantages:-

1. Low energy density.
2. Low volumetric effⁿ. because it is a gaseous fuel.
3. Need for large pressurised storage tank.
4. Refueling is a slow process.

Heat Balance

IAS, IES
Gate
①

A test on a two-stroke engine give following results.

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Speed	350 rpm
Brake load	65 kg
mean eff. press.	3 bar
Fuel consumption	4 kg/hr
Cooling water flow rate	500 kg/hr
water inlet temp.	20°C
water outlet temp.	40°C
Test room temp.	20°C
Temp of exhaust gases	400°C
A/F ratio	32
Cylinder dia.	22 cm
Stroke	28 cm
Brake dia.	1 m
C.V. of fuel	43000 kJ/kg
Proportion of H_2 in fuel	15%
Mean sp. heat of dry ^{exhaust} gases	1 kJ/kg·K
Mean sp. heat of steam	2.1 kJ/kg·K
Latent heat of steam	2250 kJ/kg

Find

- ① Indicated power
- ② Brake power & draw heat balance sheet for test in kJ/min and also in %

Solⁿ:-

$$I.P. = \frac{P_m \cdot L \cdot A \cdot N \cdot K}{60} \quad (2\text{-stroke})$$

$$= \frac{300 \times 0.28 \times \pi \times (0.22)^2 \times 350}{4 \times 60}$$

$$I.P. = 18.63 \text{ Kw}$$

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$$d = 1 \text{ m.}$$

$$r = 0.5 \text{ m.}$$



$$T = F \times r$$

$$= 65 \times 9.81 \times 0.5$$

$$65 \text{ kg} \times F$$

$$F = 65 \times 9.81 \text{ N}$$

$$b_p = \frac{2 \pi N T}{60}$$

$$= \frac{2 \pi \times 350 \times 65 \times 9.81 \times 0.5}{60}$$

$$b_p = 11.68 \text{ Kw.}$$

The input energy = chemical energy
of fuel

$$= m_f \times C.V.$$

mass flow rate of fuel

$$m_f = 4 \text{ kg/hr} = \frac{4}{60} \frac{\text{kg}}{\text{min}}$$

(157)

$$m_f \times C.V. = \frac{4}{60} \frac{\text{kg}}{\text{min}} \times 43000 \frac{\text{KJ}}{\text{kg}}$$

$$= 2866.67 \text{ KJ/min}$$

$$\begin{aligned} \text{Brake power in KJ/min} &= 11.68 \times 60 \\ &= 700.8 \text{ KJ/min} \end{aligned}$$

Heat absorbed by cooling water
heat carried $= m c_p \Delta t$

$$= \frac{500}{60} \frac{\text{kg}}{\text{min}} \times 4.187 \frac{\text{KJ}}{\text{kg K}} \times (40 - 20) \text{ K}$$

$$= 697.83 \text{ KJ/min}$$

Heat carried by dry exhaust Gases
 $= m_{de} C_{pde} (\Delta t)$

$$m_{de} = \text{mass of dry exhaust gases.}$$

$$\text{Exhaust} = \text{dry exh. gases} + \text{H}_2\text{O}$$

$$m_f + m_a \longrightarrow \text{Exhaust gases}$$

$$4 + 128 \longrightarrow 132 \text{ Kg/hr.}$$

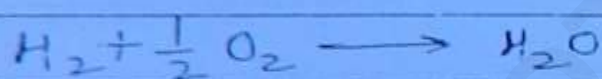
total exh. gases

$$\frac{A}{F} = 32$$

$$m_a = 32 \times 4 = 128$$

$$m_f = 4 \text{ Kg/hr.}$$

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$$2 \text{ Kg H}_2 \longrightarrow 18 \text{ Kg of H}_2\text{O}$$

$$1 \text{ Kg H}_2 \longrightarrow 9 \text{ Kg of H}_2\text{O}$$

$$0.15 \text{ Kg of H}_2 \longrightarrow 0.15 \text{ Kg H}_2$$

Kg fuel

$$1 \text{ Kg} \longrightarrow 0.15 \times 9 \frac{\text{Kg of H}_2\text{O}}{\text{Kg fuel}}$$

$$\longrightarrow 1.35 \frac{\text{Kg H}_2\text{O}}{\text{Kg fuel}}$$

As 4 Kg/hr. of fuel is used
total water vapour produce

$$= 4 \times 1.35 \text{ Kg H}_2\text{O} / \text{hr.}$$

$$= 5.4 \text{ Kg H}_2\text{O} / \text{hr.}$$

$$\begin{aligned}\text{Total dry exhaust gas} &= 132 - 5.4 \\ &= 126.6 \text{ Kg/hr.}\end{aligned}$$

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$$m_{de} = 126.6 \text{ Kg/hr.}$$

$$Q_{de} = m_{de} \cdot c_{pde} (\Delta t)$$

$$= \frac{126.6}{60} \frac{\text{Kg}}{\text{min}} \times 1 \times (40 - 20)$$

$$Q_{de} = 801.8 \text{ KJ/min.}$$

Heat carried by steam

$$= m c_{pw} (100 - 20) + mL + m c_{ps} (400 - 100)$$

$$20 - 100 \rightarrow 100 - 400$$

SH

LH

Steam

$$= m [c_{pw} \times 80 + L + c_{ps} \times 300]$$

$$= \frac{5.4 \text{ Kg}}{60 \text{ min}} \left[4.187 \times 80 + 2250 + 2.1 \times 300 \right]$$

$$= 289.4 \text{ KJ/min.}$$

Heat balance sheet

Parameters	Heat (KJ/min.)
1. Fuel	2866.67 (Total)
2. bp	700.8
3. Cooling H ₂ O	697.83
4. Dry exhaust gases	801.8
5. Steam	289.4
	2489.83

* Heat unaccounted = $2866.67 - 2489.83$
 $= 376.84 \text{ KJ/min.}$

These losses are frictional and Radiation losses.

	%
1. -	100%
2. $\frac{700.8}{2866.67} \times 100 = 24.4$	
3. $\frac{697.83}{2866.67} \times 100 = 24.34$	
4. $\frac{801.8}{2866.67} \times 100 = 27.97$	
5. $\frac{289.4}{2866.67} \times 100 = 10.09$	

MORSE TEST

- * This test is used to find out Indicated power (i.p.) of a multicylinder engine.
- * In this test speed is kept constant so that frictional power remains constant.

(16)

$$\text{All cycle firing} \rightarrow i p_1 + i p_2 + i p_3 + i p_4 = \sum_{i=1}^4 b p_i + f p$$

$$\text{First cut-off} \rightarrow i p_2 + i p_3 + i p_4 = \sum_{i=2}^4 b p_i + f p$$

$\therefore$

$$i p_1 = \sum_{i=1}^4 b p_i - \sum_{i=2}^4 b p_i$$

--

$$i p = i p_1 + i p_2 + i p_3 + i p_4$$

Note: Indicated power of each cylinder is obtain by cutting of spark plug (in case of S.I. engine) of each cylinder and cutting of fuel supply (in case of C.I. engines.)

Prob. ①

during morse test on a
4-cylinder engine the following
measurements of brake power
were taken at constant speed.

All cylinder firing

Brake power (bp)

3037 kW

No. ① cylinder not firing

bp

2102 kW

No. ② cylinder not firing

bp

2102 kW

No. ③ cylinder not firing

bp

2100 kW

No. ④ cylinder not firing

bp

2098 kW

then the mechanical eff'n.

Solⁿ:-

$$\begin{aligned} i p_1 &= 3037 - 2102 \\ &= 935 \text{ kW} \end{aligned}$$

$$i p_2 = 935 \text{ kW}$$

$$i p_3 = 937 \text{ kW}$$

$$i p_4 = 939 \text{ kW}$$

$$\dot{P} = \dot{P}_1 + \dot{P}_2 + \dot{P}_3 + \dot{P}_4$$

$$= 935 + 935 + 937 + 939$$

$$\dot{P} = 3746 \text{ kW}$$

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$$\eta_m = \frac{3037}{3746} \times 100$$

$$\eta_m = 81.07\%$$

(2) For S.I. engine the equivalence ratio ϕ of mixture enters the combustion chamber has values

- a) $\phi > 1$ for idling, $\phi < 1$ peak power
- b) $\phi > 1$ for idling, $\phi = 1$ peak power
- c) $\phi > 1$ for both idling & peak power
- d) $\phi < 1$ for both idling & peak power

(3) Assertion:- Air standard effⁿ of thermodynamic cycles is higher than Actual efficiency of engines.

Reason:- Air is not a perfect gas.

Solⁿ:-

Assertion (V), Reason (X)

(C)

* ④ which of the following factors increase detonation in S.I. engines

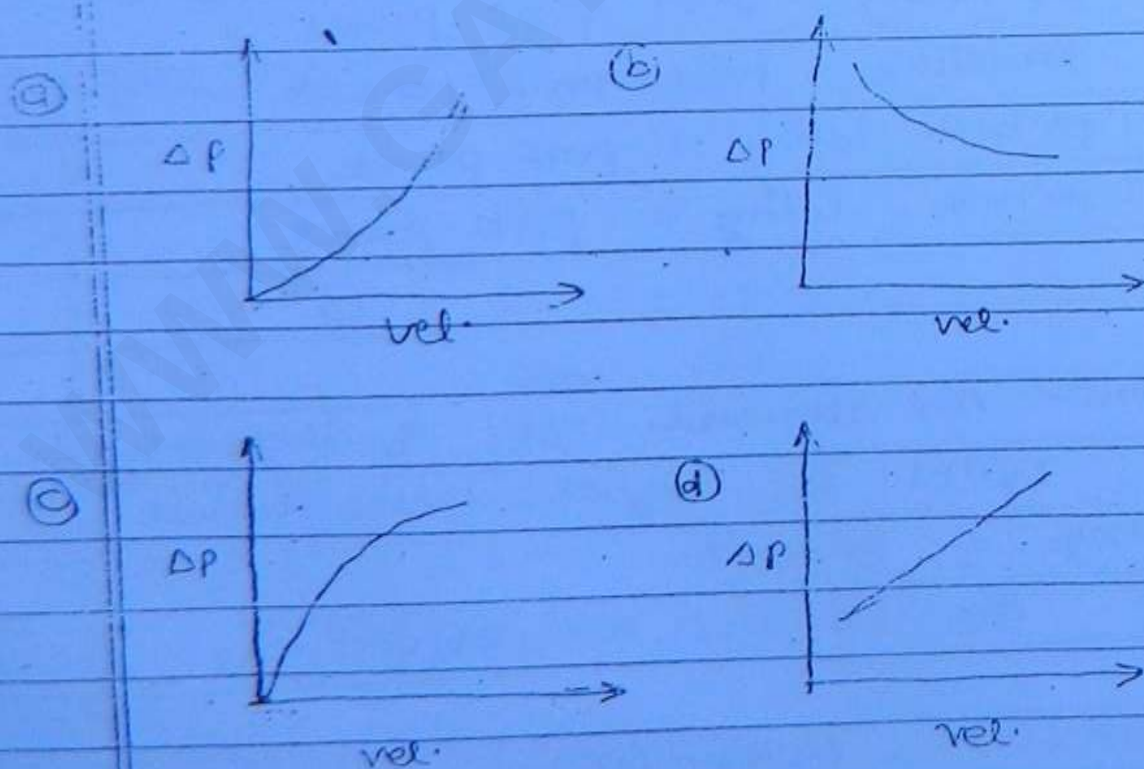
- ① Increase spark advance
- ② Increase speed
- ③ Increase A/F ratio beyond stoichiometric strength
- ④ Increased Compression ratio.

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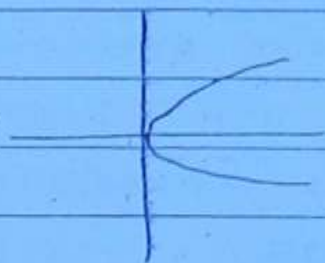
Solⁿ:- ① & ④



Prob ⑤ which of the following curve is proper representation of pressure differential (ΔP) Vs velocity of Air at the throat of Carburetor.

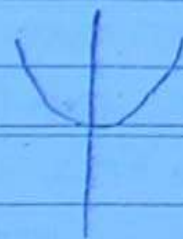


Solⁿ:-



$$y^2 = 4ax$$

(165)



$$x^2 = 4ay$$

$$v = \sqrt{\frac{2 \Delta P}{\rho_a}}$$

$$v^2 = \frac{2 \Delta P}{\rho_a}$$

$$x^2 = 4a(y)$$

**
Q. Match the following

List I

-- List II

(Elements of Carburetor)

(Rich mixture requirement)

- | | |
|-----------------------|--|
| (A) Idling System | — (1) To Compensate for dilution of charge |
| (B) Economiser | |
| (C) Acceleration pump | (2) for cold starting |
| (D) Chock | (3) For meeting power range of operation |
| | (4) For meeting Rapid opening of throttle |

Solⁿ:-

A B C D

(1) (3) (4) (2)

Prob (7) Consider the following statements

- (1) The performance of an S.I. engine can be improved by increasing the compression ratio.
- (2) Fuels of higher octane number can be employed at higher compression ratios.

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Solⁿ:- Both are correct statements

(8) Assertion:- Self ignition temp. of end charge must be higher to prevent knocking in S.I. engines

Reason:- Higher compression ratio decreases the temp. of air/fuel mixture.

Solⁿ:- Assertion (✓) ; Reason (X)

Ques (9)

List-I (mode of operation)	List-II (Approx. A/F ratio)
(A) Idling	(1) 12.5
(B) Cold starting	(2) 9
(C) Cruising	(3) 16
(D) max. power	(4) 22
	(5) 3

- (10) A Gas engine has a swept volume of 300 cc and a clearance volume of 25 cc its volumetric effⁿ. is 0.88 and mech. effⁿ. is 0.9. What is the volume of mixture taken in per cycle.

Solⁿ -

$$\eta_{vol} = \frac{V_{a/cycle}}{V_s}$$

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$$V_{a/cycle} = \eta_v \times V_s$$

$$= 0.88 \times 300$$

$$V_{a/cycle} = 264$$

- (11) The correct sequence of decreasing order of effⁿ. of three given basic engine effⁿ. are.

- (i) 2-S S.I.
(ii) 4-S S.I.
(iii) 4-S C.I.

Solⁿ -

	S.I.	C.I.
efficiency	6-10	14-22

2-stroke

4-stroke

less efficient

more efficient

$$4-S (C.I.) > 4-S (S.I.) > 2-S (S.I.)$$

The thermal effⁿ. of 4-s engine is more than 2-s engine

(168)

Volumetric effⁿ. of 4-s engine is greater than 2-s engine because of more time for induction.

The power of 2-s engines is more than the 4-s engines for same size.

Consider the following statements

Knock in S-I engine can be reduced by

- (i) Supercharging
- (ii) Retarding the spark
- (iii) Using a fuel of long straight chain structure
- (iv) Increasing engine speed.

Soln:- (ii) & (iv) are correct (✓)

Note:- Difference b/w 2-s & 4-s engines

3n 2-s → Ports

4 4-s → Valves

- (13) If methane undergoes combustion with stoichiometric quantity of air then A/F ratio on mass basis is.

Soln:-



$$\text{C} - \frac{12}{16} = 0.75$$

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$$\text{H} - \frac{4}{16} = 0.25$$

$$\left(\frac{A}{F}\right)_{\text{st.}} = \frac{100}{23} \left[\frac{8}{3} \text{C} + 8\text{H} \right]$$

$$= \frac{100}{23} \left[\frac{8}{3} (0.75) + 8(0.25) \right]$$

$$\left(\frac{A}{F}\right)_{\text{st.}} = 17.4$$

- (14) When a 4-stroke diesel engine running at 2000 rpm has a injection duration of 1.5 milli sec. What is the corresponding duration of crank angle in degree.

Soln:-

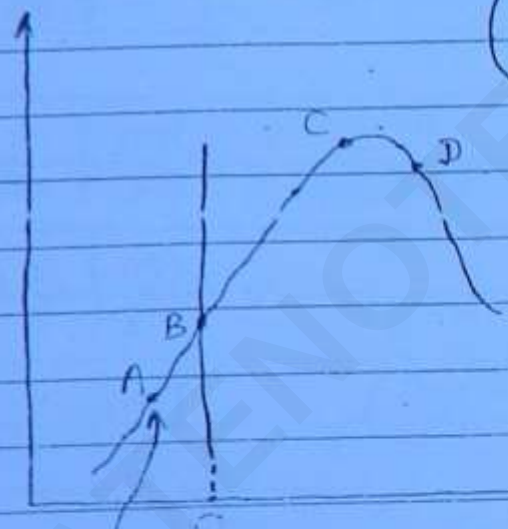
$$\theta = \frac{t \times 360 \times N}{60}$$

$$\theta = \frac{1.5 \times 10^{-3} \times 360 \times 2000}{60}$$

$$\theta = 18^\circ$$

15) A hypothetical pressure diagram for a C.I. engine is shown in the fig. The diesel knock is generated during the periode

- (a) AB
- (b) BC
- (c) CD
- (d) After D



start of ignition

HMT
TD
PAC
FM
FC
GOM

P_{01}

NE

BM BM

BM BM

OC
760mm

r_{12}

442
820
100
100
100

2400 rpm

1200 rpm

3

1200 rpm

1200 rpm

$$m(110) = (20)$$

20

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Supercharging

We know that

(171)

$$IP = \frac{P_m LAN}{60}$$

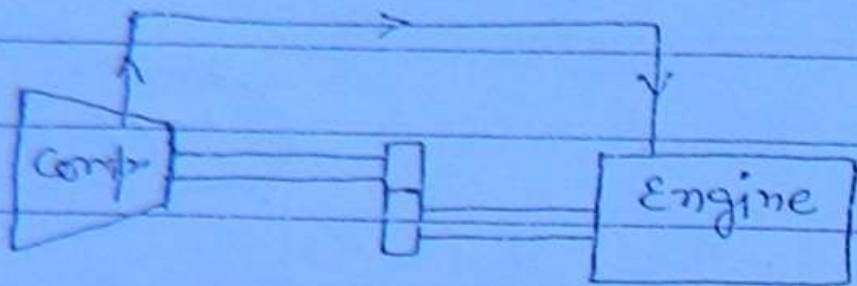
Indicated power can be increased by increasing all the terms in Numerator but increase in $L \& A$ results in larger engine and increase in N results in increasing frictional losses and hence the best option to increase I.P. is to increase mean effective pressure. This can be done by increasing the pressure of charge at the inlet to the engine cylinder.

Methods of Supercharging :-

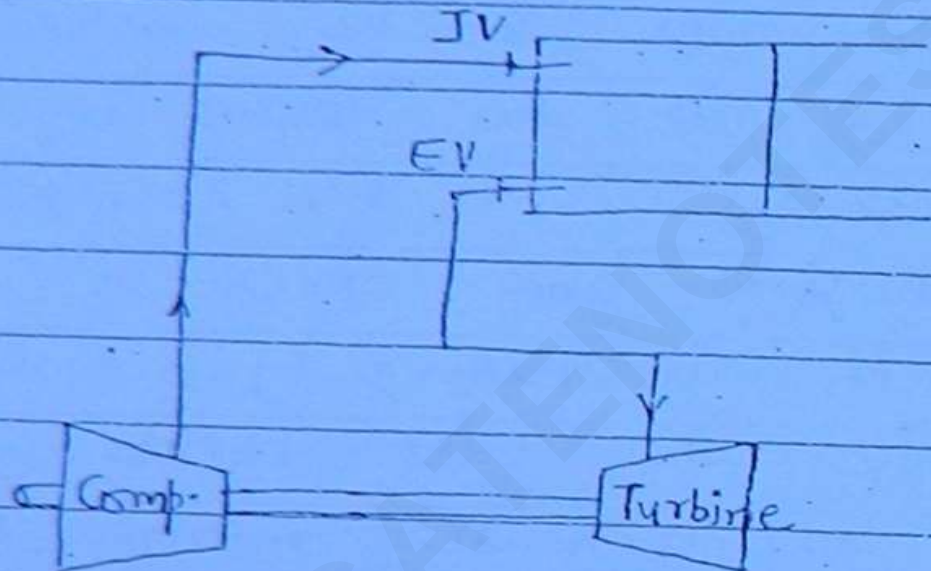
- ① Mechanical Supercharging
- ② Turbo charging

In mechanical supercharging, some engine power is utilised to run the compressor. Whereas in turbocharging exhaust gas energy is utilised in running compressor.

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mechanical Supercharger



Turbo charging

Assignment

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- ① An experimental 4-s petrol engine of $1710 \text{ cm}^3 (V_s)$ Capacity. is to develop max. power at 5400 rpm. the volumetric effⁿ. is assumed to be 70% and A/F ratio is 13. Two carburetors are to be fitted and air speed at the throat (C_2) is 107 m/s. the coefficient of discharge for venturi is assumed to be 0.85. and that of fuel jet is 0.66 and allowance must be made for a tube the dia. of which is $1/1.25$ of throat dia. The petrol surface is 6 mm below the fuel jet nozzle.

Calculate -

Size of throat and fuel jet.

The sp. gravity of petrol is 0.75. atm. pressure & temp are 1.013 bar and 27°C resp.

(Throat dia. = 2.17 cm & Jet dia. = 1.23 mm)

- ② A 4-s petrol engine of 2 Litre Capacity (V_s) is to develop a power at 4000 rpm. The Vol. effⁿ is 0.75 and $A/F = 14$. The venturi throat dia. is 30 mm. The coefficient of discharge of venturi is 0.9 and that for fuel jet is 0.6.

Calculate -

The air velocity at throat and dia. of fuel jet.

The sp. gravity of fuel is 0.76, atm. pressure & Temp. are 1 bar & 293 K.

(Air velocity = 80.5 m/s & Dia. of fuel jet = 1.9365 mm)

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15) A 5-cylinder, 4-stroke petrol engine with a bore of 120 mm & stroke of 200 mm was supplied with petrol of composition 82% carbon, 18% Hydrogen by mass. The dry exhaust composition by mass was 16.45% CO_2 , 3.85% O_2 & 79.7% N_2 .

Determine.

The actual A/F ratio and percentage excess air.

$$C = 0.82$$

$$H_2 = 0.18$$

$$\left(\frac{A}{F}\right)_{st.} = \frac{100}{23} \left[\frac{8}{3} C + 8H_2 \right]$$

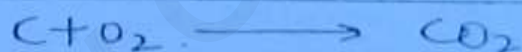
$$= \frac{100}{23} \left[\frac{8}{3} (0.82) + 8(0.18) \right]$$

$$\left(\frac{A}{F}\right)_{st.} = 15.76$$

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Actual A/F ratio

			14	20
			37	350
				455
Ex. gas	%	Kg exh.	32	455
CO ₂	16.45	0.1645	32	2 40
			41	1340
			156	
O ₂	3.85	0.0385	49	11.12
			22	12
N ₂	79.7	0.797	227	14
				25



0.1645 Kg CO₂ / Kg exhaust gas

$$12 \times 0.1645$$

$$= 44$$

Kg C

Kg Exh. gas

$$0.04486$$

Kg C

Kg exhaust gas

Page _____

We know that
0.82 Kg of C is present in
per Kg fuel.

(176)

$$\frac{\text{Kg C}}{\text{Kg fuel}} = 0.82$$

$$\frac{\text{Kg C}}{\text{Kg Ex. gas}} = 0.04486$$

$$\frac{\text{mass of Exh. gas}}{\text{Kg fuel}} = 18.28$$

We know that

1 Kg exhaust gas contains
0.797 Kg of N_2 therefore for
18.28 Kg exhaust gases N_2

$$= 0.797 \times 18.28$$

$$= 14.57$$

Fuel + Air $\longrightarrow$ Exhaust

$O_2 + N_2$

N_2

14.57

14.57

100 Kg air $\longrightarrow$ 77

2. $\longrightarrow$ 14.57