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MRG

-: HAND WRITTEN NOTES:-

OF 166

MECHANICAL ENGINEERING

1

-: SUBJECT:-

INTERNAL COMBUSTION ENGINE

6

2

Engine Nomenclature :-

Top Dead centre (TDC)

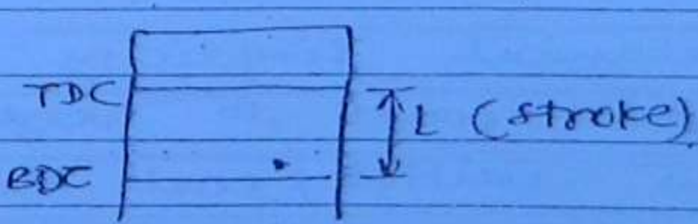
It is the dead centre when piston is ~~nearest to~~ Farthest from the crankshaft (It is & equal to I.D.C in case of horizontal engines)

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 cylinde

Date 15 oct.
Page 1

(1)

Introduction

Engine :-

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

(4)

Heat Engines → C.E → H.E. → 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 products of combustion transfer heat to working fluid whereas in IC engine the products 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, condensor etc.
4. Higher efficiency.

Clearance volume (V_c)

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

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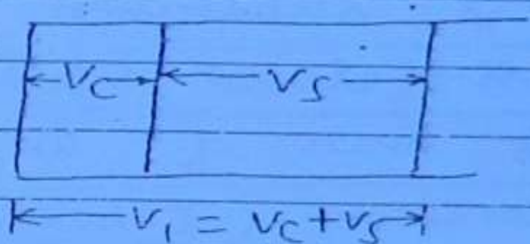
Compression ratio (CR or γ)

It is define a volume before compression to the volume after compression.



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

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

Air Standard Cycles
(or) (Ideal I.C. engine cycles)
(or) (Const. volume cycle)

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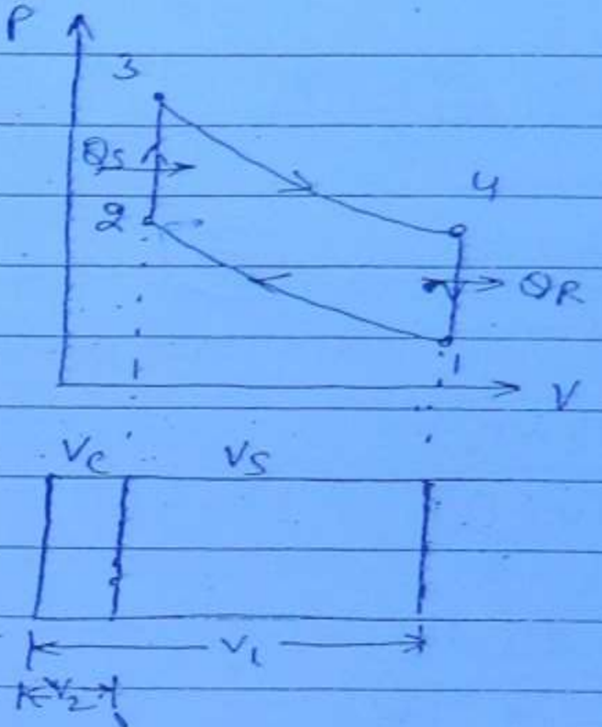
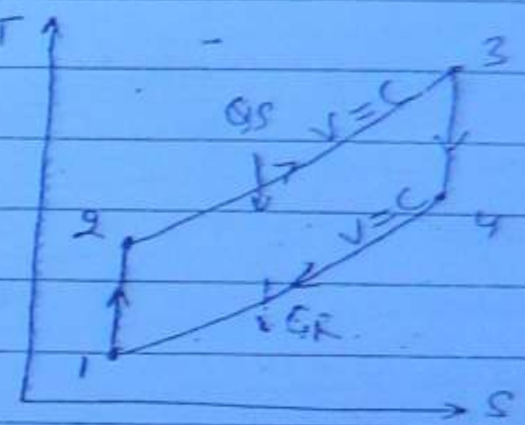
Assumptions made in Air Standard Cycles:-

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 ~~chang~~ chemical change.
5. All the processes are reversible processes.

Otto Cycle

(or) (Const. volume cycle)





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 → cycle is power producing cycle

* Anticlockwise → cycle in power consuming cycle.

Note:- Swept volume $V_S = V_1 - V_2$

Air standard efficiency (or) Ideal eff^{n} of otto cycle:-

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

$$\eta = 1 - \frac{Q_R}{Q_S} \quad (8)$$

$$Q_S = mC_v(T_3 - T_2)$$
$$Q_R = mC_v(T_4 - T_1)$$

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

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

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

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)

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

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

∴ from ① and ② we have

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

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

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

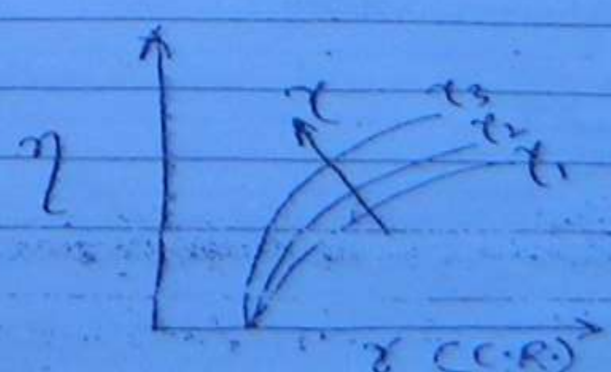
$$\therefore \quad \eta = 1 - \frac{T_1}{T_2}$$

$$\eta = 1 - \frac{1}{(T_2/T_1)}$$

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

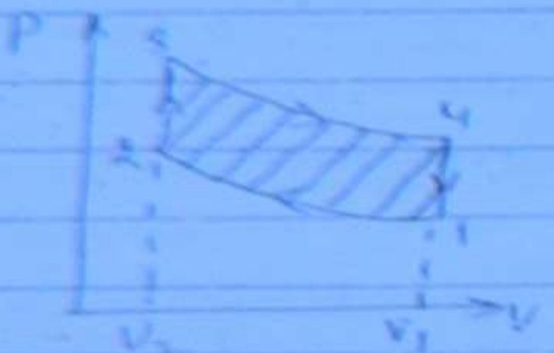
e:- The specific heat Efficiency of otto cycles is depend upon compression ratio (r) and adiabatic index (γ).

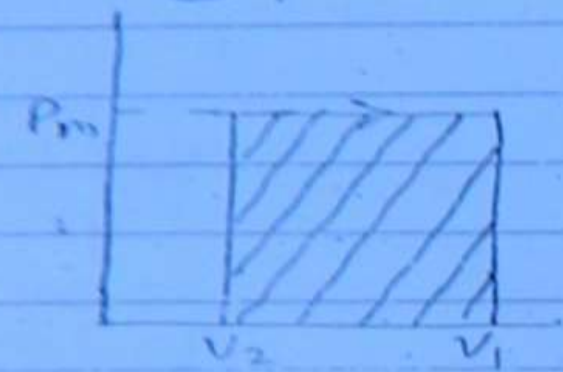
efficiency vs compression ratio (η vs r



Mean effective pressure (mep)

It is the constant pressure (the Hypothetical pressure) during expansion producing same work as the actual cycle.





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

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

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

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

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

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

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

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

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

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

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

Q) 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

Ans.

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

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

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

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

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

$$= \frac{23.625 \times 10^5}{r - 1} = \frac{23.625 \times 10^5}{}$$

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

$$= 5.25 \text{ bar}$$

Engine

(5) An I.C. 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 cm^3 and heat supply per kg air is 1800 kJ/kg (9c) then the work output is.

Sol^n:-

(13)

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

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

$$=$$

$$V_C = 196.3 \ cm^3$$

$$V_1 = V_S + V_C \qquad V_2 =$$

$$=$$

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

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

$$r = 1 + \frac{V_S}{V_C}$$

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

$$r = 1 + \frac{\frac{\pi}{4} (10)^2 \times 15 \ cm^3}{196.3 \ cm^3} = 7$$

$\eta = 1 - \frac{1}{(r)^{\gamma - 1}}$ = [illegible] $\frac{1}{(7)^{1.4}}$ [illegible]

[illegible]

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

$$\omega = \eta \times Q_s$$

(14)

$$= 0.541 \times 1800$$

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

5) The mean effective pressure of an otto cycle can be expressed as

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

where η_{th} = Thermal effn.

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

ΔP = pressure rise during heat addition

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

r = pressure ratio

γ = adiabatic index

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

Sol:-

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

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

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

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

$$\omega_{net} = \eta_{th} \times Q_S$$

(15)
$$= \eta_{th} \times m C_V (T_3 - T_2)$$

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

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

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

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

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

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

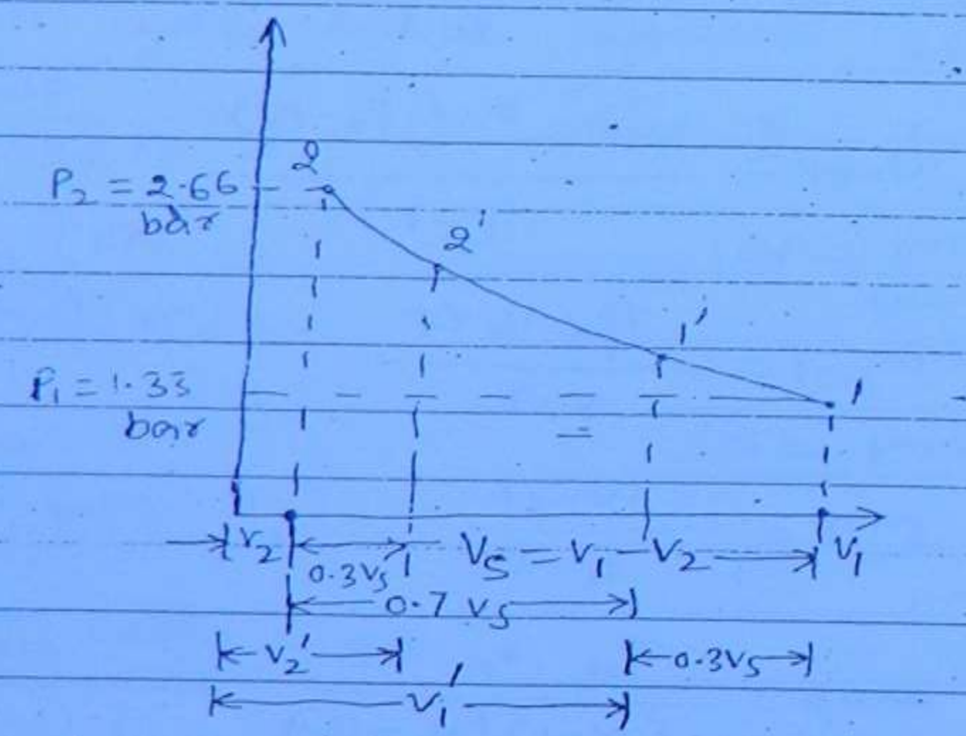
$$mep = \frac{\eta_{th} V_2 \Delta P}{(\gamma - 1)(V_1 - V_2)}$$

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

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

(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)



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

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

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

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

Similarily

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

we know

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

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

$$\frac{\cancel{V_2}[1+0.7(r-1)]}{\cancel{V_2}[1+0.3(r-1)]} = \left(\frac{2.66}{1.33}\right)^{1/1.33}$$

$$r = 4.51$$

Q. Show that for max. work per kg of air in otto cycle for a given upper and lower temp. of ~~T3~~ & ~~T1~~ 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

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

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

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

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

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

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

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

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

Differentiate above eqn.

$$\frac{dW}{dr} = 0$$

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

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

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

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

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

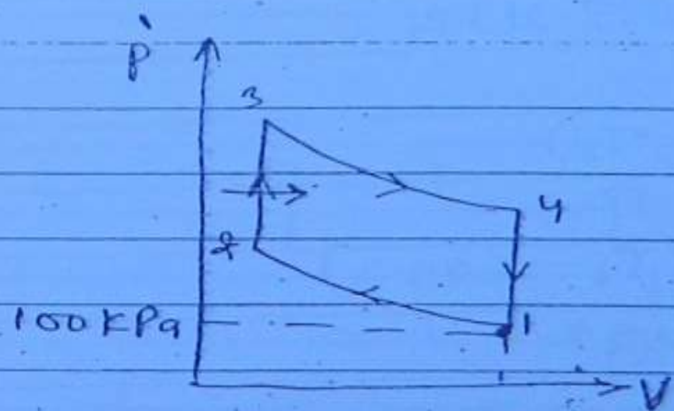
$$T_2 = \sqrt{T_1 T_3}$$

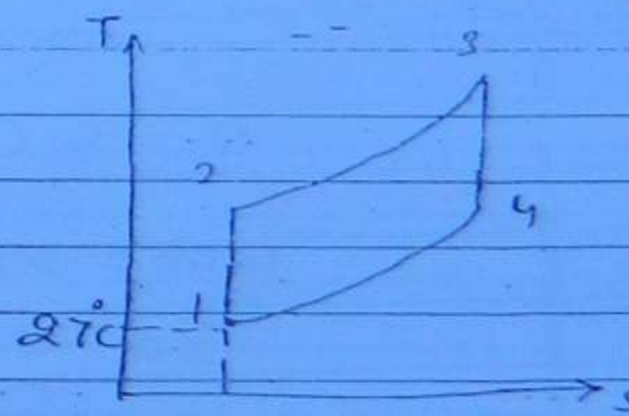
Note:- For max. workoutput in otto cycle temp. after compression is equal to temp. after expansion.

IES

(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 ($r = 8$),
$C_v = 0.72$ KJ/Kg-K & $\gamma = 1.4$

Solⁿ:-





$Q_a = 1500$ KJ/Kg $P_1 = 100$ KPa

$T_1 = 27 + 273 = 300$ K

$$P_2 V_2^{\gamma-1} = P_3 V_3^{\gamma-1}$$

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

For process 1 − 2

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

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

$$T_2 = 300(8)^{1.4-1}$$

$$= 689.2 \text{ K}$$

1 − 2 (adiabatic)

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

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

$$P_2 = 100(8)^{1.4}$$

$$= 1837.91 \text{ kPa}$$

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

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

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

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{(or)} \quad \left\{ \frac{T_2}{T_1} = \frac{T_3}{T_4} \right\}$$

$$T_1 T_3 = T_2 T_4$$

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

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

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)

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

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

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

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

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

$$= \underline{0.561}$$

$$W = \eta \times Q_s$$

$$= 0.561 \times 1500$$

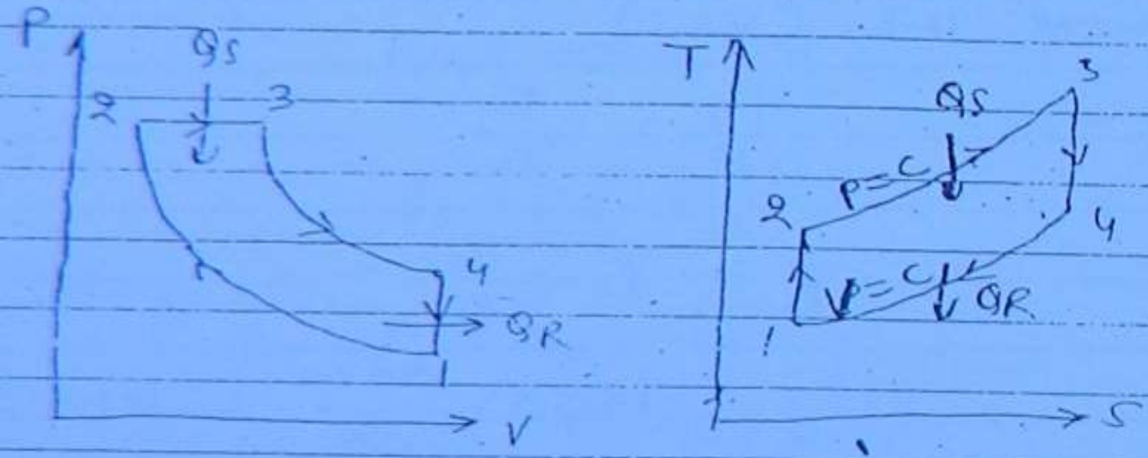
$$= 847 \text{ KJ/Kg}$$

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

(22)

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

Diesel Cycle
(or Const. pressure cycle)



Cut-off Ratio (r_c)

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

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

Air standard eff

Air standard eff^n of Diesel cycle :- η_{diese}

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

$$Q_S = mC_p(T_3 - T_2)$$
$$Q_R = mC_v(T_4 - T_1)$$

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

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

For 1-2

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

For 2-3

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

For 3-4 (adiabatic)

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

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

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

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

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

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

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

$$\frac{T_3}{T_4} = \frac{T_2/T_1}{r_c^{\gamma-1}} \qquad \because \frac{T_2}{T_1} = r^{\gamma-1}$$

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

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

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

from eqn. (i)

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

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

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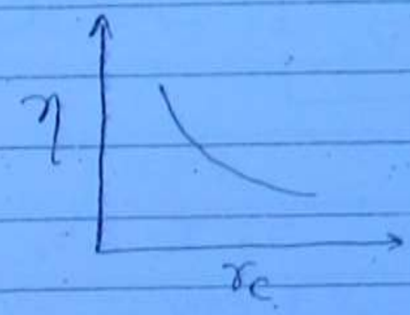
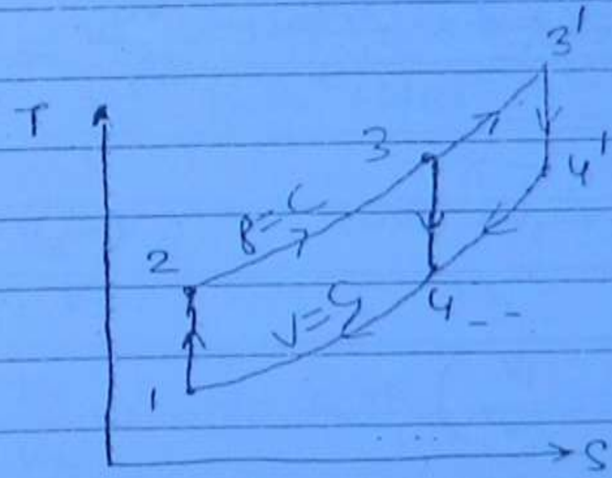
** As the cut-off ratio increases the effn. of diesel cycle decreases.



** 

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

(3) Expansion ratio (r_e)

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

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

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Relationship b/w r, r_c & r_e

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

$$\boxed{r_c r_e = r}$$

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:-

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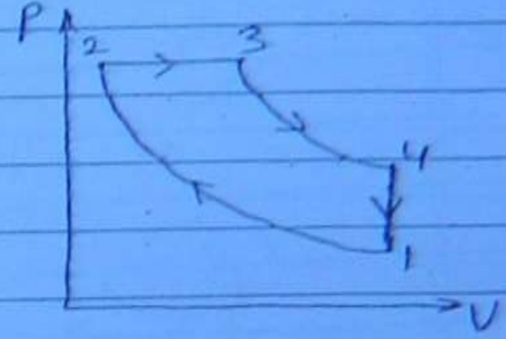

* Cut-off volume $= V_3 - V_2$

$r = 20 \qquad \gamma = 1.4$

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

$$V_3 - V_2 = 0.05\,V_1 - 0.5\,V_2$$

devide V_2 both sides

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

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

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

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

$$= 1 - \frac{1}{1.4}\cdot\frac{1}{(20)^{1.4-1}}\left[\frac{(1.95)^{1.4} - 1}{1.85 - 1}\right]$$

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

$$= 0.6490$$

$$\eta = 64.90\,\%$$

(2) Determine the air standard eff^n. 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 ($\gamma = 1.4$)

Sol^n:-

D =

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

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

$$r = 1 + \frac{\frac{\pi}{4} (2.5)^2 (37.5)}{1500} = 13.27$$

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

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

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

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

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

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

$= 0.6052$

$\eta = 60.52\%$

(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 ($C_p = 1.004$ kJ/kg-K, $C_v = 0.717$ kJ/kg-K & $\gamma = 1.4$)

Soln:- $P_1 = 1$ bar ; $T_1 = 17 + 273 = 290$ K

$P_2 = 35$ bar

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Process 1-2 (adiabatic)

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

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

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

109-51
2-10
5

$$r_e = 5$$

$$r_c r_e = r$$

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

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

$$= 0.549$$

$$\eta = 54.9\ \%$$

$Q_R = C_p(T_2 - T_2)$

1-2 (adiabatic)

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

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

$T_2 = 290 \left(12.67\right)^{1.4-1} = 807.26K$

2-3 (const. pres process)

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

$T_3 \Rightarrow T_2 \times 2.51$

$= 807.26 \times 2.51 = 2042.36K$

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

$Q_s = 1240 \; KJ/Kg$

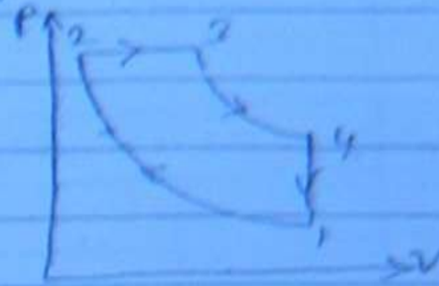
$\eta = 1 - \frac{Q_R}{Q_s}$

$0.549 = 1 - \frac{Q_R}{1240}$

$Q_R = 555 \; KJ/Kg$

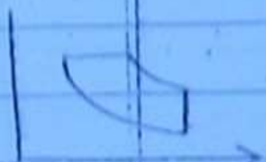
A diesel cycle operates at a pres. of 1 bar at the beggining of compression and the volume is compressed to 1/16 of initial volume. Heat is supplied untill the volume is ...

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



Soln:

$P_1 = 1$ bar



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

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

31

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

$$V_3 = 2V_2$$

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

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

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

$$V_4 = 8V_3$$

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

$$= \frac{P_1V_1 - P_2V_2}{\gamma - 1} + P_2(V_3 - V_2) + \frac{P_3V_3 - P_4V_4}{\gamma - 1}$$

$$P_1V_1^{\gamma} = P_2V_2^{\gamma}$$

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

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

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

3 - 4

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

$$P_4 = P_3 \left(\frac{V_3}{V_4}\right)^{\gamma}$$ 32

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

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

$W_{net} =$

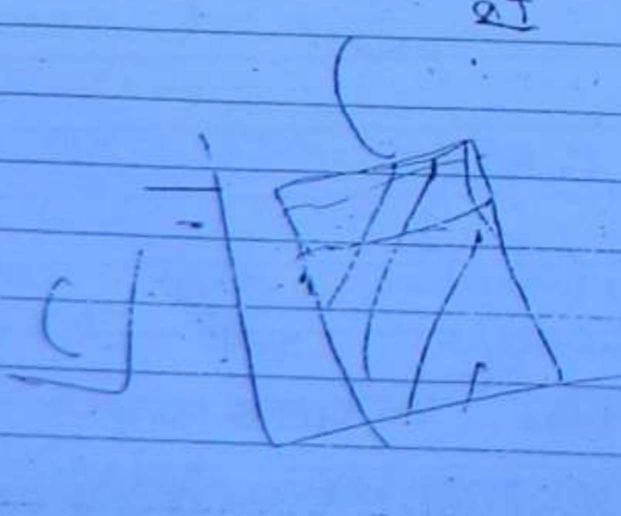
$$\frac{V_1}{V_2} = 16 \Rightarrow V_1 - V_2 = 16V_2 - V_2 = 15V_2$$

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

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

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

$$= 6.966 \text{ bar}$$



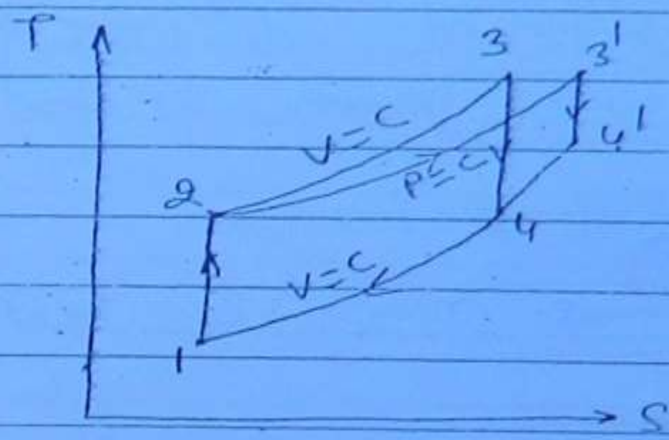
$$\eta = \frac{W}{c_p(T_3 - T_2)}$$

Comparision of otto & diesel cycle:-

Case-I :-

Same compression ratio and heat additio

33



1-2-3-4 — otto cycle

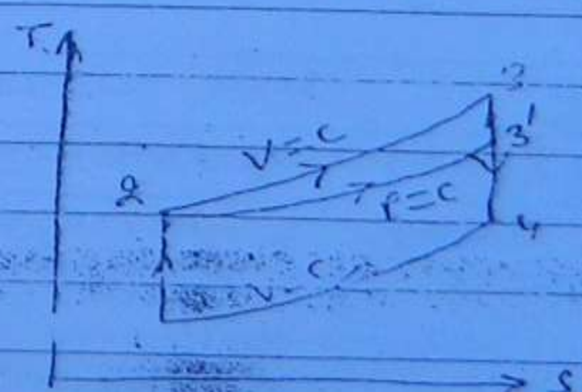
1-2-3'-4' — Diesel cycle

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

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

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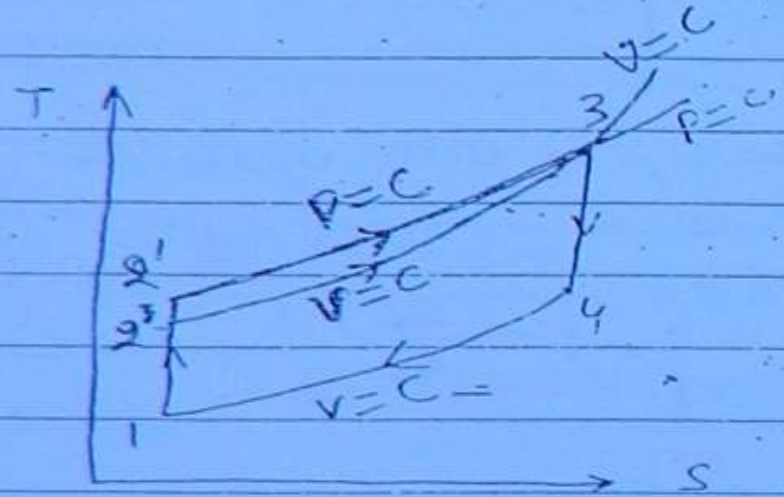
For same C.R. & heat rejection heat supplied is more in otto cycle and hence it is efficient.

Case III
Same max. Temp. & heat rejection



1-2-3-4 — Otto cycle
1-2'-3-4 — Diesel cycle

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

Dual Cycle

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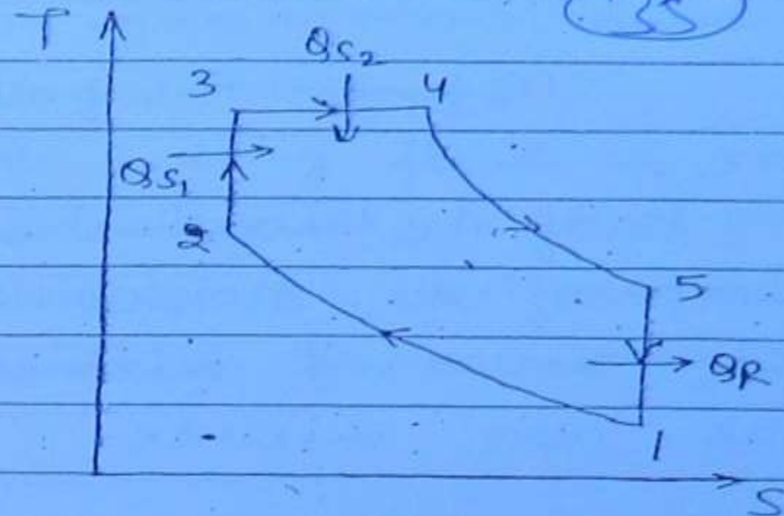

$$\eta = 1 - \frac{Q_R}{Q_{S_1} + Q_{S_2}}$$

$Q_{S_1} = mC_v(T_3 - T_2)$

$Q_{S_2} = mC_p(T_4 - T_3)$

Fuel - Air Cycle

chemically correct Air-fuel - ratio
(or) (Stoichiometric ratio) .

Fuel + oxy. $\longrightarrow$ Heat + Combustion produ

$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O + Heat$

$16 + 2(32) \longrightarrow 52$

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

1 Kg of fuel $\longrightarrow$ 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

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.

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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. Disociation of burnt gases.

Variation of specific heat with Temp.

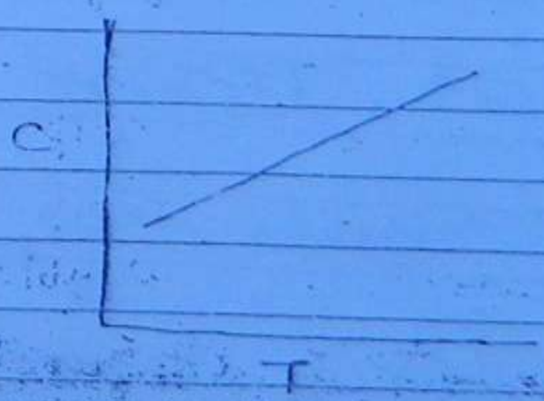
with rise in temp. specific heat increases because larger fraction of heat input is required to produce motion of atoms within the molecules since temp. is the result of motion of molecules.

upto 1500K $\begin{cases} C_p = a + KT \\ C_v = b + KT \\ C_p - C_v = a - b \end{cases}$

when $T > 1500K$

$C_p = a + K_1 T + K_2 T^2$

$C_v = b + K_1 T + K_2 T^2$



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

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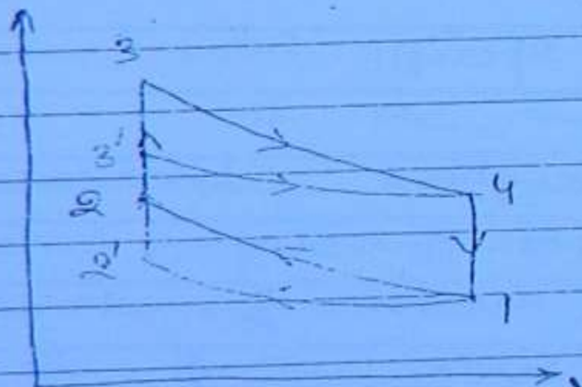
$T\uparrow,\ C_p\uparrow,\ C_v\uparrow,\ C_p - C_v = R = \text{const.}$



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

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

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

$Q = mC_v(T_3 - T_2)$

$1000 = mC_v(T_3 - T_2)$

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

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

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 eff^n. and powerout put decreases.

Dissociation of Combusion product:-

$$HC + O_2 \longrightarrow CO_2 + H_2O + Heat$$

Fuel + Air $\longrightarrow$ Comb. products + heat

dissociation (Endothermic)

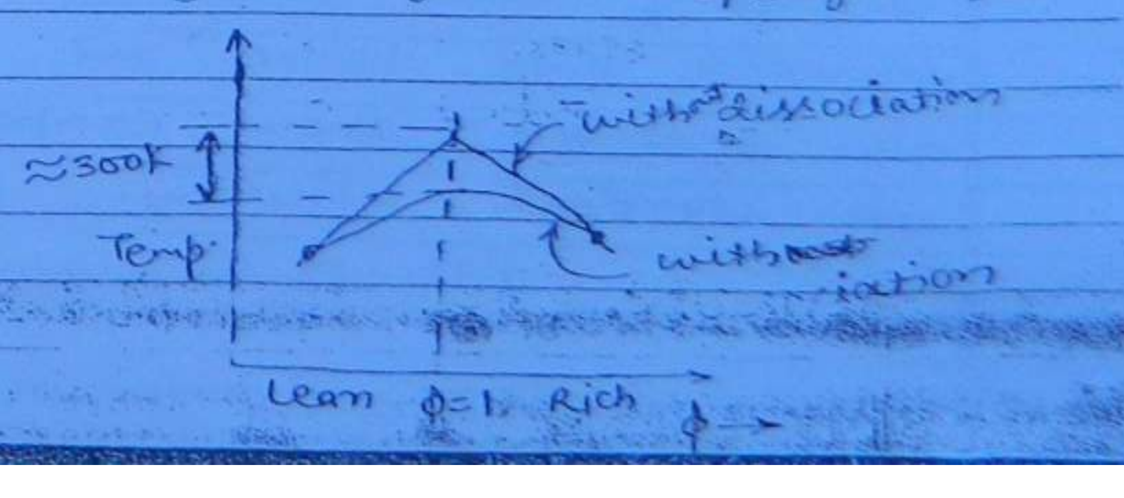
$$CO_2 + heat \longrightarrow CO + O_2$$

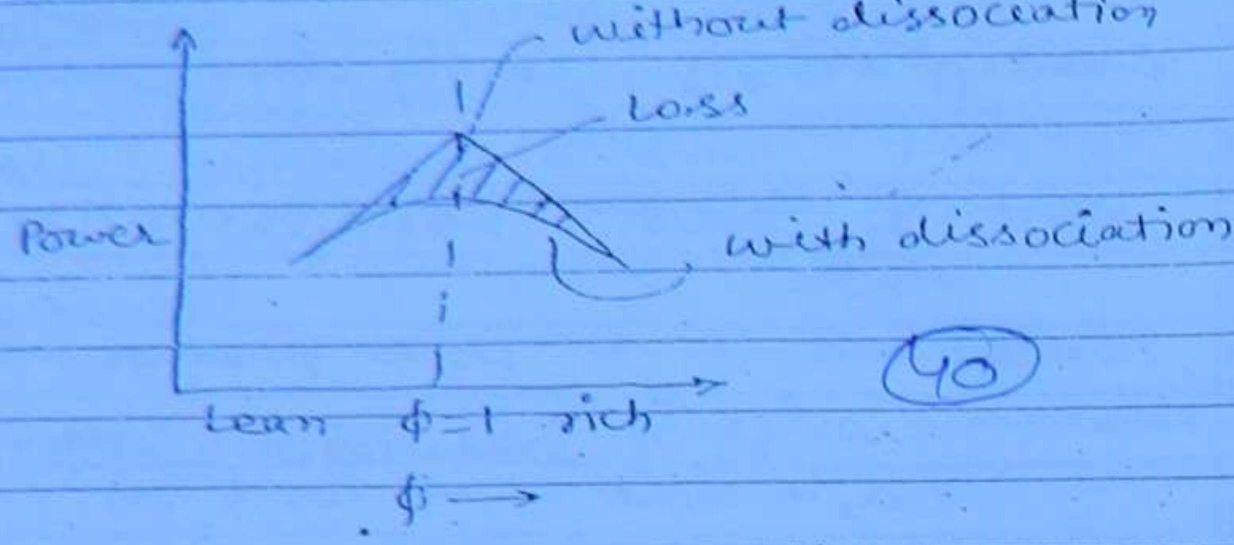
$\cancel{CO + O_2}$ $CO + O_2 \longrightarrow CO_2 + heat$

Association (Exothermic)

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

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





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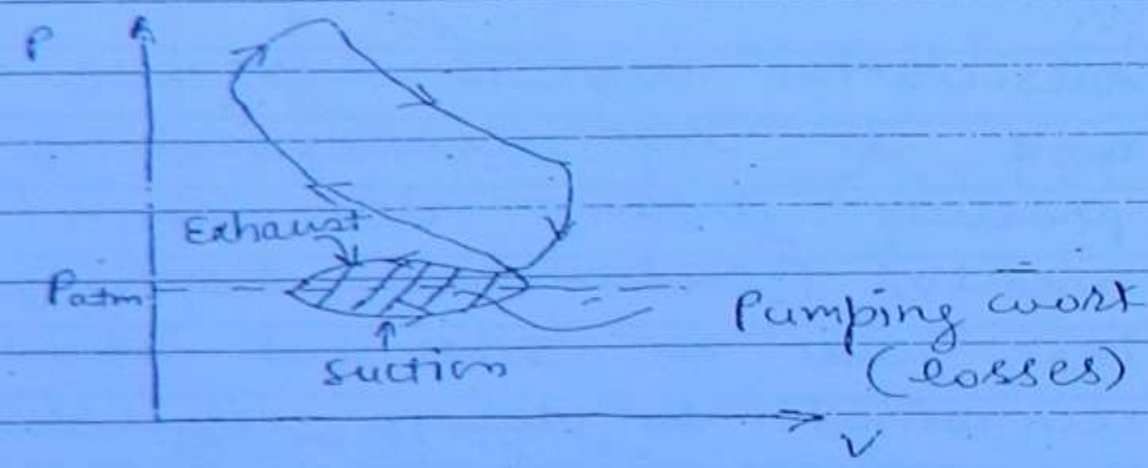
Pumping loss :-



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 then compare to air standard efficiency.

Blow by loss :-

During compression a portion of combustion products leakes ~~through~~ space b/w piston & cylinder and comes to crankcase since these products do not take part in combusion 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 $\gamma = 1.4$)

Solⁿ :-

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

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

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

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

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

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

$$\frac{1}{(1-\eta)}[0 - d\eta] = -R\left(-\frac{1}{C_v^2}\right) dC_v \ln r$$

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

(42)

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

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

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

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

$$= 0.5408$$

$$\frac{d\eta}{\eta} = \frac{-(1-0.54)}{0.54} (1.4-1)\, 0.01 \times \ln 7$$

$$= -6.63 \times 10^{-3}$$

$$= -0.663\%$$

IInd method :-

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

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

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

$$C_v = 0.717$$ (Q3)

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

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

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

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

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

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

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

$$= -0.65\ \%$$

(2) A diesel cycle has a $r = 20$ the cut off volume 6 % of stroke volume initial C_v = ~~0.071~~ 0.71 KJ/Kg-K and R = 0.287 KJ/Kg-K find the % change

effn. if c_v increases by 3%

Solⁿ:-

(44)

$$\cancel{\eta_{i_a} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(20)^{1.4-1}}}$$

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

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

$$v_3 - v_2 = 0.06\,(v_1 - v_2)$$

$$\frac{v_3 - v_2}{v_2} = 0.06\,\frac{v_1 - v_2}{v_2}$$

$$\frac{v_3}{v_2} - 1 = 0.06\left(\frac{v_1}{v_2} - 1\right)$$

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

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

$$r_c = 2.14$$

$\therefore$

$$\eta_i = 0.642$$

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

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

(45)

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

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

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

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

$$= 0.6322$$

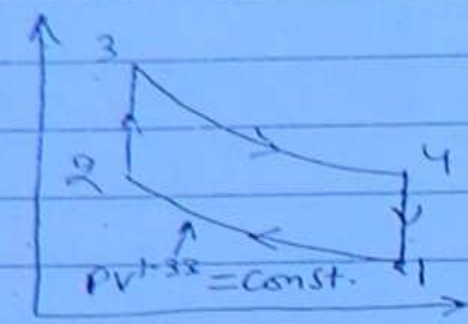
$$= \frac{0.682 - 0.6922}{0.642} \times 100$$

$$= -1.58$$

(3) An otto cycle has a $r = 10$ 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 in index of compression is 1.33 and $C_v = (0.717 + 2 \times 10^{-4} T)$ KJ/KgK

Sol^n: $n = 1.33$

(46)



2-3 ($v = c$)

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

1-2

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

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

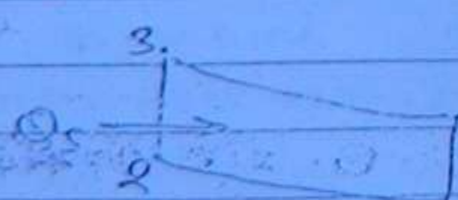
$$P_2 = 1 (10)^{1.33} = 21.38 \text{ bar}$$

$$T_1 V_1^{n-1} = T_2 V_2^{n-1}$$

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

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

$$T_2 = 748.28 K$$



$$Q_s = m_f \times CV$$

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

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

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

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

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

$$T_3 = 3125.4\,K$$

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

$$= 89.29 \text{ bar}$$

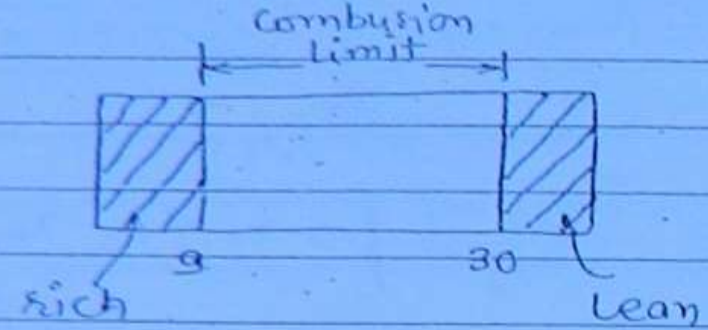
SIT

Ignition of
flame speed - 15-30 m/s

Date 17 Oct.
Page

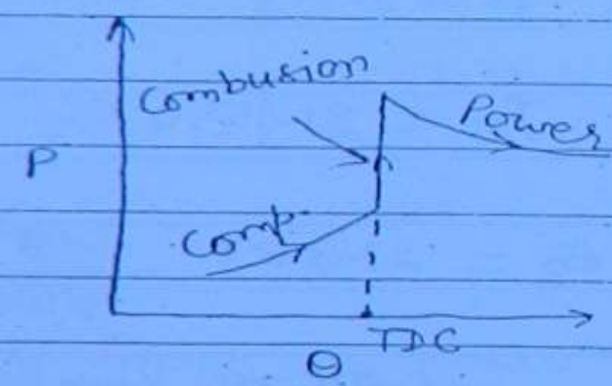
Combusion in S.I. Engine

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

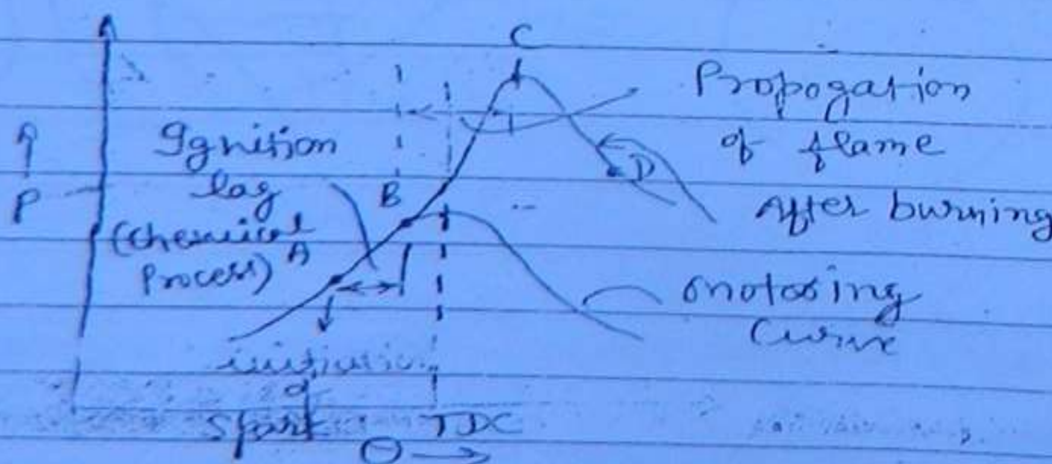


Ignition limits :-

Theoritical P-θ diagram ::



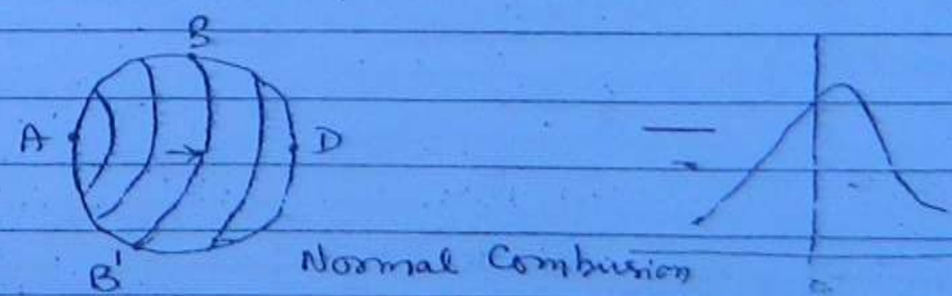
Stages of Combusion in S.I. Engine



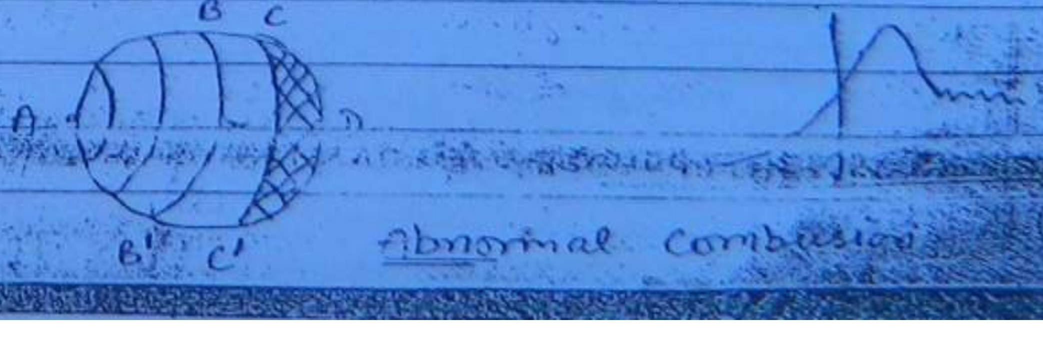
* Point A is passage of spark (initiation of spark) it is generally 20° before T.
* Point B it is the point at which the beggning of pressure rise can be detect generally 8° before TDC.
* Point C is attenment of peak pressure
* AB is first stage of combustion, BC is second stage of combustion and CD is third stage of combustion.
* AB is called ignition lag or preperation phase in which the growth and develepement of flame takes place this is a chemical process.
* BC It is consernt with spraiding of flame throughout the combustion chamb the starting point of second face 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. I shows a flame front traveling across combustion chamber from A to D as the flame propogates the unburnt mixture called as (SO) end gas is compressed causes its temp to increase. The temp. of the end gas increases further when it riceaves heat by 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. [illegible]

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

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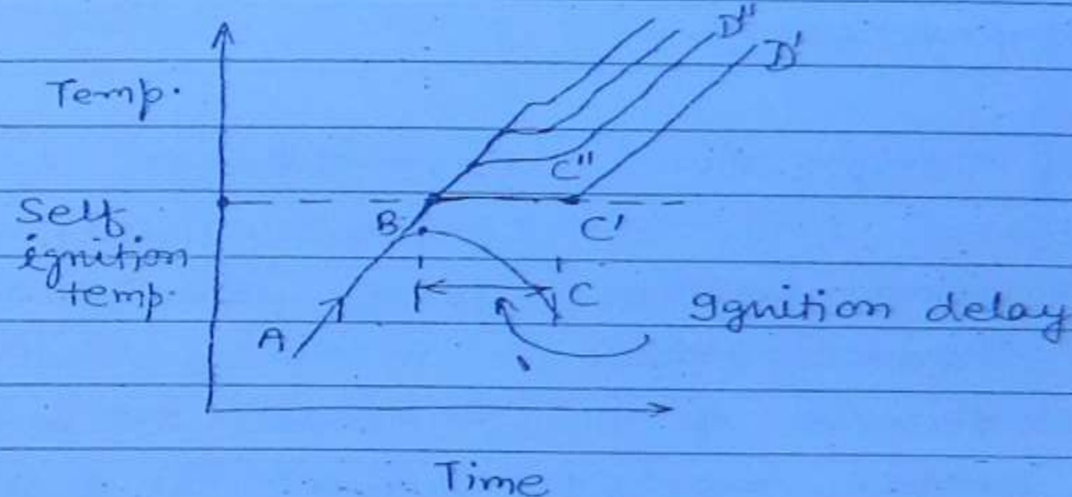
Factors controlling / Effecting detonation:-

Auto - Ignition Theory :-



Auto ignition does not occur immideate 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 rapidily Compressed and held at that temp. and if this temp is less then self ignition ter (Point B) the mixture will not autoignite but will slowly cool down If the self ignition temp is Reached [illegible]

Certain ignition delay. The delay periode is shortened iff 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 periode and hence knocking Tendency increases. Because of this reason the compression ratios in S.I engines is resticted to about 10. (6 – 10)

2. Super charging :- supercharging is the process of alowing 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 result 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 centraly has min Tendency for knocking because the flame travel distance decreases.

8. Location of exhaust valve :- The exhaust valve should be located close to spark plug so that the exh it is not in the end gas region otherwise detonation will occur.

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

10. Turbulence :- Increase in turbulence increases flame speed and hence knocking tandency 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. [illegible] knocking

Tandency. A higher octane no. means lower knocking tandency.

13. Fuel / Air ratio:-

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

T↑ ⟶ ignition lag ↓ ⟶ detonation ↑

T↑ ⟶ flame speed ↑ ⟶ detonation ↓

velocity is max. But the effect of former is more and hence knocking tandency 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 over heating of combustible mixture.

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

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Minimum Air required for Combustion

$$\underset{\text{fuel}}{HC} + \underset{\text{Air}}{O_2} \longrightarrow \text{Combustion + heat Prod.}$$

$$C + O_2 \longrightarrow CO_2$$
$$12 + 32 \longrightarrow 44$$

12 Kg of C requires $\longrightarrow$ 32 Kg of O_2

1 Kg of C ,, $\longrightarrow$ $\frac{8}{3}$ Kg of O_2

For C Kg of carbon oxygen required is $8/3 \times C$

Hydrogen Combustion Equation:-

$$2H_2 + O_2 \longrightarrow 2H_2O$$
$$2(2) + 32 \longrightarrow 2(2+16)$$
$$4 + 32 \longrightarrow 36$$

4 Kg H_2 requires 32 Kg of O_2

1 kg of $H_2 \longrightarrow 1 \times \frac{32}{4} = 8$ kg of O_2

for H_2 kg of $H_2 \longrightarrow 8 H_2$ kg of O_2

If sulphur is present in fuel

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$$S + O_2 \longrightarrow SO_2$$
$$32 + 32 \longrightarrow 64$$

32 kg of S requires $\longrightarrow$ 32 kg of O_2

1 kg of S $\longrightarrow \frac{32}{32} = 1$ kg of O_2

for S kg of sulphur $\longrightarrow$ S kg of O_2 is required

Total O_2 required for the combustion of fuel

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

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

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

100 kg of air — 23 kg of O_2

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

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

This is known as stoichiometric Air/fuel ratio.

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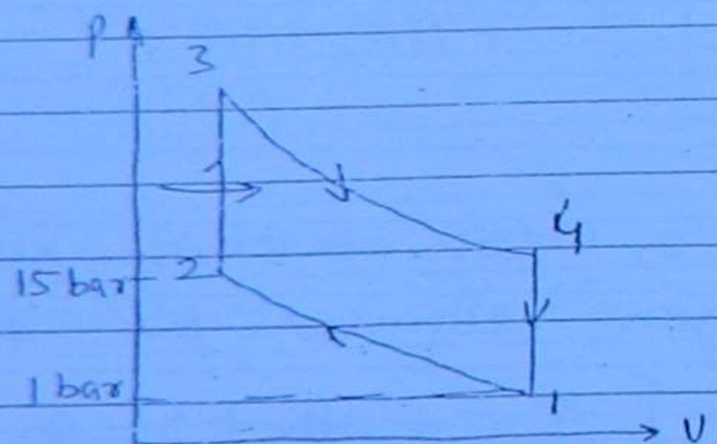
Prob:- (1) In an otto cycle air at 17°C and 1 bar is compressed adiabatically untill the pressure is 15 bar heat is added at const. volume untill the pressure rises to 40 bar Calculate

i. effn.

ii. compn. ratio

iii. mep ($C_v = 0.717$ KJ/Kg-K , $R = 0.287$)

Soln:-



$P_1 = 1$ bar $\quad P_2 = 15$ bar $\quad P_3 = 40$ bar

$T_1 = 17 + 273 = 290$ K

(1-2)

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

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

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

$$r = 6.919$$

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

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

$$= 53.87\%$$

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

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,V

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

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

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

$$= 628.65\,K$$

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

(or) $w_{net} = \eta \times Q_s$

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

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

$$w_{net} = 404.69$$

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

$$= 751.25\ kJ/kg$$

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

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

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

$$V_1 = 0.823$$

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

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

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

$$= \frac{(404.69)}{0.823 - 0.1204}$$

$$= 575.9$$

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

II^nd methode

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

$$= \frac{0.5387 \times (40-15)}{(6.91-1)(1.4-1)}$$

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

Performance parameters :-

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

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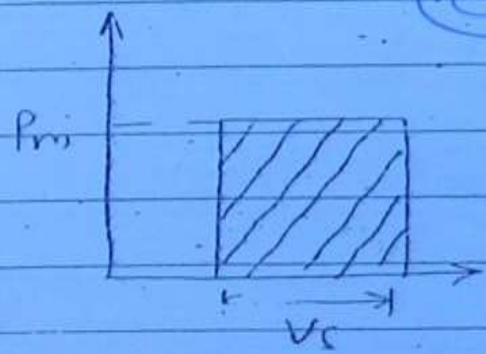

$$W_{net} = \frac{P_m \times V_s}{Cycle}$$

$$I.P = \frac{P_m V_s}{Cycle} \times \frac{N\,cycle}{60\ sec}$$

$$I.P = \frac{P_m \times ALN}{60} \quad \text{(For two stroke engine)}$$

For four stroke engine

$$I.P. = \frac{P_m LAN}{2 \times 60}$$

If there are K no. of cylinders then

$$I.P. = \frac{P_m LANK}{2 \times 60} \quad \text{(for 4-stroke)}$$

$$I.P = \frac{P_m LANK}{60} \quad \text{(for 2-stroke)}$$

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

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

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
CV = calorific value.

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

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

(63)

7. Indicated specific fuel consumption (ISFC) :-

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

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

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

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

* $$\eta_{th} = \frac{ip}{m_f\, CV} = \frac{KW}{\frac{Kg}{sec} \times \frac{KJ}{Kg}} = \frac{KW}{KW}$$

Breake specific fuel consumption (BSFC): It is the fuel consume to develope brake power

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

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

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Combustion in C.I. Engines
(or) Compression ignition engines
(or) 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 begging of injection and attenment of chemical reaction condition is known as physical delay. During this periode fuel is atomised

(or) 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 chambe it will vapourised and mixes with air. The time spend in this process is known as physical delay. The time 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 beggining 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 chamb 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.

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ote:- ① The parometers which controll 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 then diesel. So that the delay periode will be more resulting in civier 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 comes out from the engine and thereby large no. of carbon atoms deposite at the spark plug and hence the spark plug will be blocked.

Factors controlling detonation

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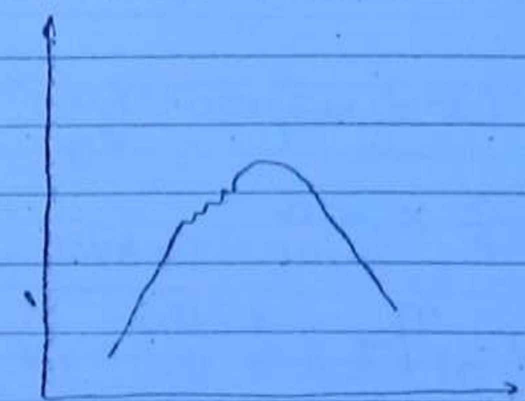
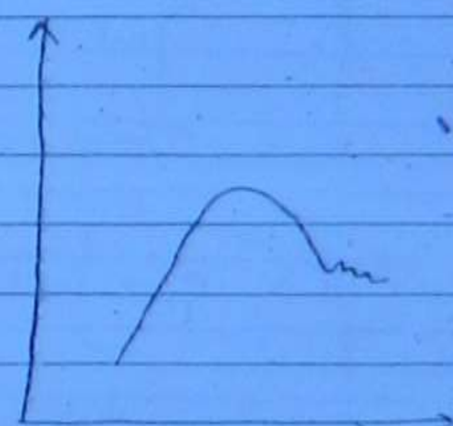
Factors	S.I.	C.I.
S.I.T	high	low
delay period	high	low
CR	Low	high
Inlet Temp.	Low	high
Inlet pressure	Low	high
Combustion wall temp.	Low	high



C.I. Engines



S.I. Engines

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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 C.V. of fuel is 42000 KJ/kg

Density of air is 1.12 Kg/m3 and mep in the cylinder is 10 bar and mechanical effn. is 80%. Determine

i. Indicated Thermal effn. &

ii. Brake power.

Soln:-

$K = 4$ four stroke engine

$r = 8$

$D = L = 100\,mm$

$\eta_{vol.} = 0.75$ $N = 4800\,rpm$

$A/F = 15$ $CV = 42000\ \frac{KJ}{Kg}$

$\rho_a = 1.12\,Kg/m^3$ $mep = 10\,bar$

$\eta_m = 0.80$

$$I.P = \frac{P_m LANK}{60 \times 2}$$

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

$$= 125.66\ KW$$

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

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

$$= 0.8 \times 125.66$$

$$= 100.528\ KW$$

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

(69)

4800 rpm = 2400 Cycles/min.

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

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

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

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

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

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

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

As there are 4-cylinders to

Total volume taken $= 4 \times 0.02356$

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

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

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$$m_a = \rho_a \times V_a$$

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

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

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

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

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

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

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

$$= 0.425$$

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

IES ***

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

Soln:-

C_2H_6O (7)

$12 \times 2 \longrightarrow C = 24$

$1 \times 6 \longrightarrow H = 6$

$16 \longrightarrow O = 16$

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

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

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

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

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

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

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

C_2H_6O + Air $\longrightarrow$ Products + heat

$C_2H_6O + O_2 + N_2 \longrightarrow CO_2 + H_2O + N_2$

$$C + O_2 \longrightarrow CO_2$$
$$12 + 2(16) \longrightarrow 44$$

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12 kg C produces 44 kg of CO_2

1 kg C —— ?

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

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

$= 1.906$ kg of CO_2

$$H_2 + \frac{1}{2} O_2 \longrightarrow H_2O$$

$$2 + 16 \longrightarrow 18$$

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

$= 1.17$ kg of H_2O vapour

$$\underbrace{O_2 + N_2}_{\text{Air}} \longrightarrow 9.06$$

100 kg air 77 kg, N_2

9.06 Kg air —— $\frac{77 \times 9.06}{100}$

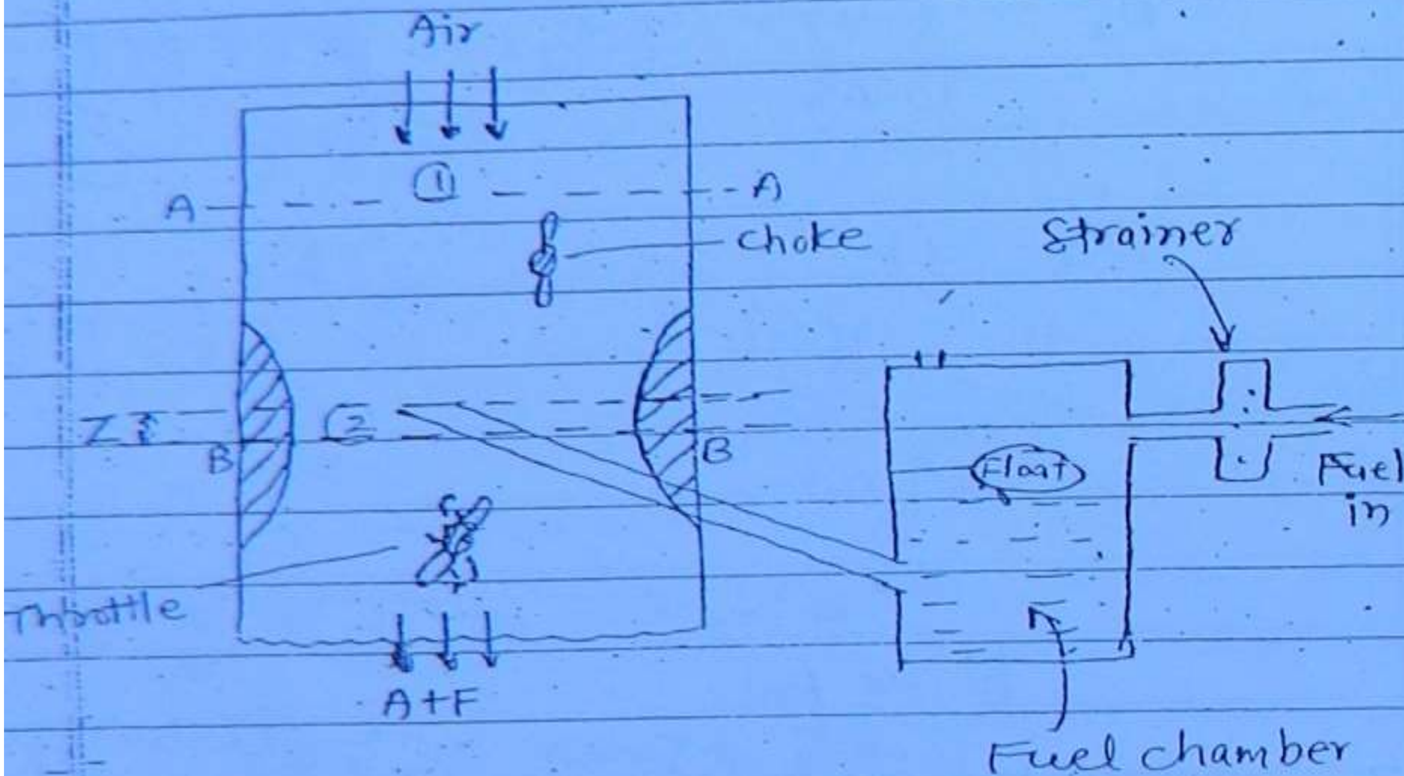
73 $= 6.97$ Kg of N_2

CO_2	1.906	$\frac{1.906}{10.05} \times 100 = 18.96\%$
H_2O	1.17	$\frac{1.17}{10.05} \times 100 = 11.64\%$
N_2	6.97	$\frac{6.97}{10.05} \times 100 = 69.35\%$
	10.05	

B 4	2
4-S [illegible]	2-S [illegible]
[illegible]	[illegible]
TM rotation [illegible]	[illegible]
heavy [illegible]	[illegible]
Power for same bore stroke	[illegible]
Low rate of wear and tear, lubrication	[illegible]
	More lubr parts
Volumetric ↓ Light [illegible] weight	[illegible]
η_{vol} more	η_{vol} less
η_{th} ↑	η_{th} ↓
Bulky [illegible]	

Carburetion & Fuel Injection

Simple Carburetor :-



Carburetor is a device which supplies combustible mixture to the cylinder

Exact analysis to find Air/Fuel ratio:-

$$h_1 + \frac{c_1^2}{2} = h_2 + \frac{c_2^2}{2}$$

$$c_1 <<< c_2$$

$$h_1 = h_2 + \frac{c_2^2}{2}$$

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

C_2 = velocity of air at throat

$$C_2 = \sqrt{2(C_pT_1 - C_pT_2)} \quad 75$$

$$C_2 = \sqrt{2C_pT_1\left(1 - \frac{T_2}{T_1}\right)}$$

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

$$\boxed{C_2 = \sqrt{2C_pT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}}$$

$$m_a = \rho_2 A_2 C_2 \qquad [\because m = \rho AC]$$

$$m_a = \rho_2 Q$$

m_a = Theoritical mass $= \rho_2 A_2 C_2$

Actual mass

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

Theoritical mass flow rate ⇒

$$m_a = \rho_2 A_2 C_2$$

where A_2 = Area of venturi at the float

ρ_2 = density of air at float

(76)

Actual mass flow rate of Air

$$m_a = C_d \rho_2 A_2 C_2$$

C_d = Coefficient of discharge for venturi

$$pV = mRT \qquad \rho = \frac{p}{RT}$$

$$p = \left(\frac{m}{V}\right) RT \qquad \therefore \rho_1 = \frac{p_1}{RT_1}$$

$$p = \rho RT$$

P_1, T_1 — atm. pressure & Temp.

Isentropic

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

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

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

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

$\therefore$

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$$m_a = C_d \, \rho_2 A_2 C_2$$

$$m_a = C_{da} \cdot \rho_1 \left(\frac{p_2}{p_1} \right)^{1/\gamma} \cdot A_2 \, C_2$$

$$\boxed{m_a = C_{da} \cdot \rho_1 \left(\frac{p_2}{p_1} \right)^{1/\gamma} A_2 \cdot \sqrt{2 C_p T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} \right]}}$$

Mass of Fuel :-

Assuming fuel to be incompressible (ρ_f = const.)

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

$$\frac{p_1}{\rho_f g} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{\rho_f g} + \frac{v_2^2}{2g} + z_2$$

$$\frac{p_1}{\rho_f g} + \frac{0^2}{2g} + 0 = \frac{p_2}{\rho_f g} + \frac{C_f^2}{2g} + z$$

$$\frac{p_1}{\rho_f g} - \frac{p_2}{\rho_f g} - z = \frac{C_f^2}{2g}$$

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

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

Theoritical

$$m_f = \rho_f A_f C_f$$

Actual mass flow rate of fuel

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

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

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

*** Approximate analysis to find A/F ratio :-

Air is asumed to be incompressible.

$$m_a = \rho A C$$

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

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

$$C_1 <<< C_2$$

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

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

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

**

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

Note:- If nozzle tip height z is neglected

$$\frac{A}{F} = \frac{C_{da} A_2 \sqrt{\rho_a}}{C_{df} A_f \sqrt{\rho_f}}$$ 83

***

$$\boxed{\frac{A}{F} = \frac{C_{da} A_2}{C_{df} A_f} \sqrt{\frac{\rho_a}{\rho_f}}}$$

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.

Ins 9

(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 diamet of venturi if the air flow velocity at the ~~float~~ (throat) is 90 m/sec & velocity coefficient is 0.8. Assume isentropic flow and treat air as compressible flow. (81)

Soln:-

$m_a = 5$ kg/min

$P_1 = 1.013$ bar $\qquad T_1 = 27 + 273$

or $P_1 = 101.3$ KPa $\qquad = 300$ K

$C_2 = V_2 = 90$ m/sec

$C_v = 0.8$

$$\rho_2 = \rho_1 \left[\frac{P_2}{P_1}\right]^{1/\gamma} \qquad \rho_1 = \frac{P_1}{RT_1}$$

$$= \frac{1.013 \times 10^2}{0.287 \times 300}$$

$$\rho_2 = 0.01176 \left[\frac{P_2}{P_1}\right]^{1/\gamma}$$

$= 0.01176$

$\rho_1 = 1.176$ KPa

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

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

$C_p = 1005$ J/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 = 0.941 bar

$$\rho_2 = \rho_1 \left[\frac{P_2}{P_1}\right]^{1/\gamma}$$

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

$$\rho_2 = 1.1054 \quad kg/m^3$$

assume $C_d = 1$ (82)

$$m_a = \rho_2 A_2 C_2$$

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

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

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

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

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

$$d_2 = 0.0325 \, m.$$

$$d_2 = 3.25 \, cm.$$

or

$$d_2 = 32.5 \, mm.$$

IES(IAS)

(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

$$t = t_s - 0.0065h$$

where h = height in meter

and t_s = sea level temp in °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

Sol^n:-

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

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

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

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

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

$$\frac{A}{F} = \frac{m_a}{m_f} = \frac{C_{da} A_2}{C_{df} A_f} \sqrt{\frac{\rho_a}{\rho_f}}$$

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

SL - Sea level

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

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

$t = t_s - 0.0065\,h$

at $h = 6000$

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

$t = -12^\circ C$

$T = 273 - 12 = 261\,K$

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

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

$P = 0.742$ bar

(or)

$P = 74.2$ 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 (85)

a. Idling

b. Cruising

c. Maximum power

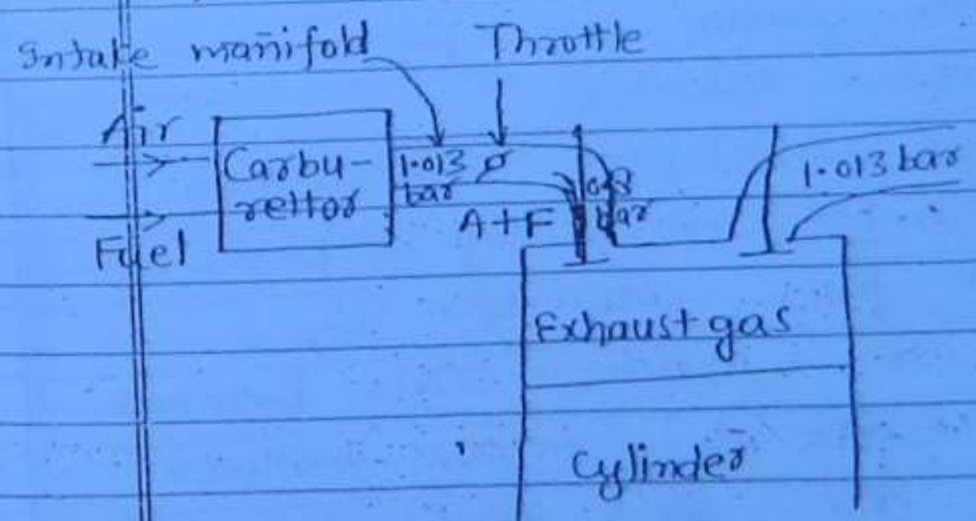
(2) Transient operation

a. Starting

b. Acceleration

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 brought in during idling is much less then 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 then 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 therefor necessary to provide more flow fuel particles by richening 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 $12 - 12.5$. $(\phi > 1)$

Cruising :- (or) (Normal Range)

(97) The exhaust gas dilution problem is insignificant in cruising and the primary aim is better fuel ecconomy.

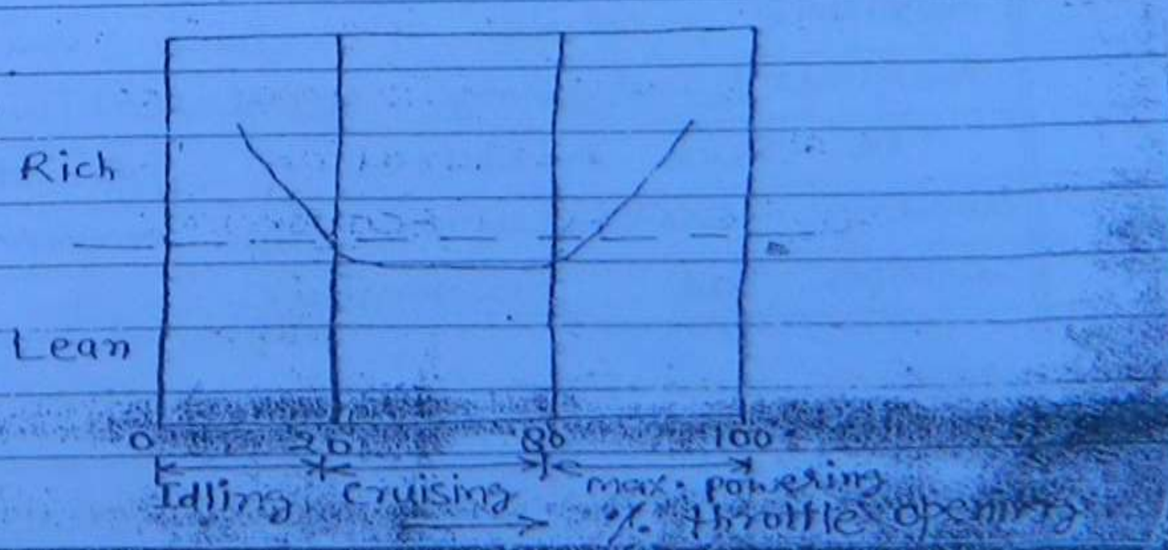
20 – 80 % of throttle opening operation is known as cruising.

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 - peake 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

Starting :- During starting the temp. are low and the fuel remains in liquid form also fuel may condance on coming in contact with cold cylinder walls eventhough air/fuel ratio is within normal combustion range the ratio of evapored fuel to air is less and hence during staring 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 recieves lean mixture. while the rich mixture is required during acceleration. The rich mixture is provided during acceleration by means of a device as Economiser.

IASO

(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

(2) 6 mm below the throat. The coefficient of discharge (d (C_{da}) for venturi, and coeff. of fuel orifice are 0.85 & 0.78 resp. specific gravity of petrol is 0.75.

(99)

Calculate

i. A/F ratio for a pressure drop of 0.08 bar

ii. Petrol consumption in kg/hr.

** iii. Critical air velocity

The intake conditions are

$P_1 = 1$ bar & $T_1 = 290$ K

For air -- $C_p = 1.005$ KJ/kg-K

$C_v = 0.718$ KJ/kg-K

neglect compressibility of air

Soln:-

$D_2 = 20$ mm , $D_f = 1.12$ mm

$z = 6$ mm , $C_{da} = 0.85$, $C_{df} = 0.78$

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

$P_1 = 1$ bar $= 100$ KPa

$T_1 = 290$ K

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

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

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

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

100-

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

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

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

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

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

$$m_f = 0.78 \times \frac{\pi}{4} (1.12 \times 10^{-3})^2 \times \sqrt{2 \times 750 \left[(0.08 \times 10^5) - 750 \times 9.81 \times 6 \times 10^{-3}\right]}$$

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

$m_f = 2.5$ [illegible]

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

$$(P_1 - P_2) > \rho_f g z \qquad (101)$$

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

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

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

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

(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	$5.75\ cm^2$
Area of negative loop of indicator diagram	$0.25\ cm^2$

Length of the diagram (l_d) 5.5 Cm

spring constant (k_s) 3.5 bar/cm

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Find

Indicated power in kW.

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

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

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

or

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

$$IP = \frac{P_m L A N K}{60}$$

$$= \frac{350 \times 0.15 \times \frac{\pi}{4}(0.1)^2 \times 1600 \times 4}{60}$$

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

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:-

$$N = 1800 \text{ rpm}$$ (103)

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

Frictional Power = I.P. − B.P

$$B.P. = \frac{2\pi NT}{60}$$

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

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

(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 effn. is 75%. Calculate indicated specific fuel consumption in Kg/kW-h.

Solⁿ:-

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

$$\eta_m = 0.75$$ 104

$$\eta_m = \frac{BP}{I.P.}$$ η_m

$$I.P. = \frac{B.P.}{\eta_m} = \frac{80}{0.75} = 106.6 \text{ kW}$$

$$BSFC = \frac{m_f}{B.P.}$$

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

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

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

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

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

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

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

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

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

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 eff^n. (CV = 43000 KJ/kg)

iii. Relative eff^n. based on brake power bases

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

Sol^n:-

$D = 80$ mm , $L = 100$mm

$= 0.8$m $L = 1$m

$V_c = 70$ CC

$N = 4000$ rpm

$T = 150$ N-m , $CV = 43000$ KJ/Kg

Fuel 1

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$$BP = \frac{2\pi N T}{60}$$

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

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

or

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

$$\eta_{bth} = \frac{BP}{m_f \times C.V.}$$

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

$$\eta_{bth} = 0.2628$$

or

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

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

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

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

$$r = 1 + \frac{\frac{\pi}{4} D^2 L}{V_c}$$ (107)

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

$$r = 8.18$$

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

$$\eta_{air\,sd} = 0.5685$$

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

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

(or)

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

Fuel Injection Equations

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

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

rad/sec

rad.

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

$180° — \pi$

$\theta° —$

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

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

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

$$t = \frac{\pi \theta \times 60}{180 \times 2\pi N}$$ 109

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

here N in ~~rad~~ 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 periode 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

$N = 1500$ rpm

$m_f = 2.5$ kg/hr.

$\rho_f = 880$ kg/m^3

$\theta = 25°$

$P_{inj} = 150$ bar

$P_{cyl} = 30$ bar

$C_d = 0.88$

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

(110)

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

$$V_f = \frac{m_f}{\rho_f} = \frac{2.5\ kg}{3600\ sec} \times \frac{1}{880\ \frac{kg}{m^3}}$$

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

for 4-stroke engine

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

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

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

$$\text{Velocity} = \sqrt{\frac{2(P_{inj} - P_{cyl.})}{\rho_f}}$$

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

(14)

$$V = 165.14 \ m/sec.$$

Time of injection

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

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

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

$\therefore$

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

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

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

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

(or)

$$d = 0.44 \ mm$$

(2) 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 periode. and take density of fuel as 875 Kg/m^3.

Soln:-

$BP = 25\,KW$

$N = 3000\,rpm$

$BSFC = \cancel{}\ 0.3\ Kg/KW\text{-}hr.$

$P_{inj.} = 160\ bar$

$P_{cyl.} = 30\ bar$

$C_v = 0.88$

$C_d = 0.65$

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

$\rho_f = 875\ Kg/m^3$

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

$$= \frac{0.3\ Kg/KW\text{-}hr}{875\ Kg/m^3}$$

$$v_f = 9.52 \times 10^{-8} \ m^3/sec$$

(113)

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

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

$$V = 151.69 \ m/sec$$

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

$$m_f = BSFC \times BP$$

$$m_f = \underset{(BP)}{25 \ KW} \times \underset{(BSFC)}{0.3 \ \frac{Kg}{Kwhr}}$$

$$m_f = 7.5 \ Kg/hr$$

(or)

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

$$\rho_f = \frac{m_f}{Vol. f}$$

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

$rpm = 3000$

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

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

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

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

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

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

Volume of fuel injected/cycle $= C_d \times A \times Vel. \times t$

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

$$t = 0.00192 \ sec$$

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

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

*** (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.:-

$$P_{inj} = 160 \text{ bar}$$ (115)

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

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

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

$$D = Vt \qquad \because V = \sqrt{\frac{2(\Delta P)}{\rho_f}}$$

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

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

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

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

$$\Delta P_1 = 130 \text{ bar}$$

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

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

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

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

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

A 6- cylinder, 4-stroke diesel engine develops 100KW at 3000 rpm when Brake thermal effn. 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 travell during injection periode is 25°.

Take C_d for fuel orifice 0.65

$d_f = $ ...

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

4 - stroke diesel engine

$K = 6$

$BP = 100\ kW$

$N = 3000\ rpm$

$\eta_{bth} = 25\%$

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$P_{inj} = 160\ bar$

$P_{cyl} = 50\ bar$

$\theta = 25$

$CV = 42000\ kJ/kg$

$\rho_f = 850\ kg/m^3$

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

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

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

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

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

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

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

and

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

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

$$V = 160.87 \quad m/s.$$

rpm = 3000 rpm

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

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

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

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

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

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

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

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

$D =$ [illegible]

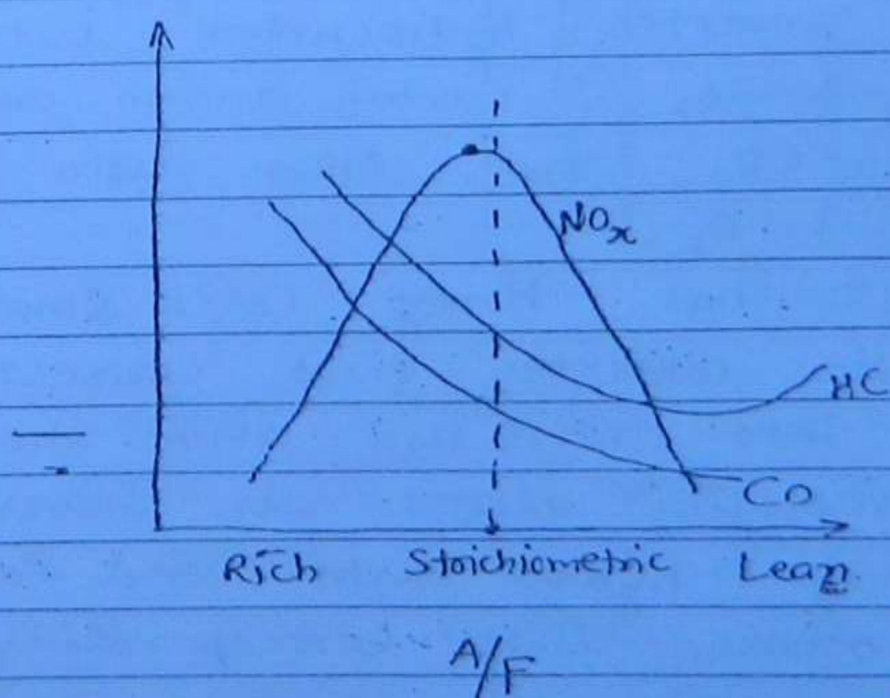
Engine Emission & Controls

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 pallutants.

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.

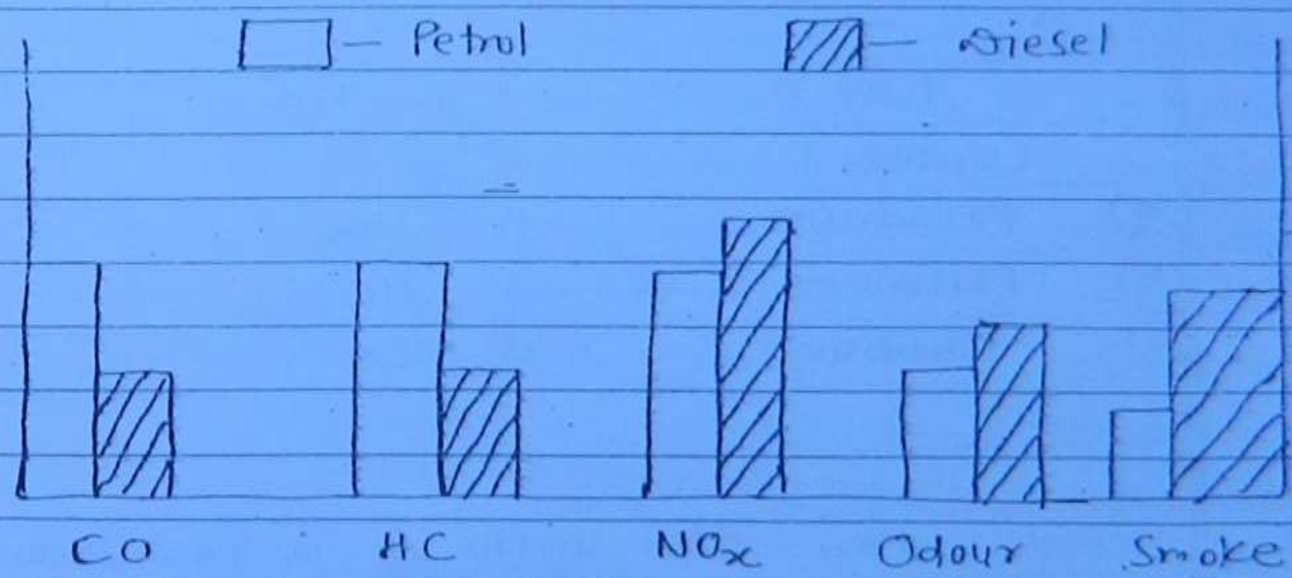
Catalytic Converter consists of Platinum, Paladium & Rhodium and it converts hydrocarbon into H_2O & CO_2, Carbon mono oxide into CO_2 and NO_x into N_2 & O_2.

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 monooxide into corresponding oxides takes place.

Drawbacks of Catalytic Converter :-

(127)

* 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 compare to petrol engine and polluts 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.

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match the following

List-I (Catalyst)		List-II
A) Palladium	1)	NO_x
B) Platinum	2)	HC
C) Rhodium	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

$$\rho_a = m_a / V_a$$

$$m_a = \rho_a V_a$$

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

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

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

$$m_f = \rho_f \times V_f$$

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

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

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

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

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

Page

(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. During a 10 min. test the dynamometer scale beam 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 (124)

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

$K = 8$

$D = 9\text{ cm}$

$L = 8\text{ cm}$

$r = 7$

$N = 4500\text{ rpm}$

dyna. radius = 54 cm

$t = 10\text{ min}$

$m = 42\ kg$

$m_f = 4.4\ kg$ $\qquad m_f = \frac{4.4}{10}\ kg/sec$

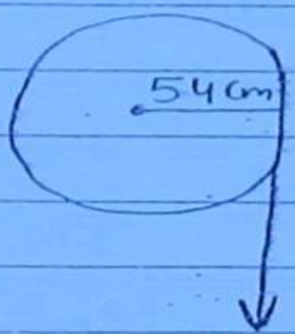
$CV = 44000\ kJ/kg$

$T_1 = 300K$

$P_1 = 1\ bar$

$m_a = 6\ kg/min.$

(2)



42 Kg

$T = F \times r$

$= 42 \times 9.81 \times 0.54$

$= 222.49$

$F = 42 \times 9.81$

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

$$= \frac{2\pi \times 4500 \times 222.49}{60}$$

$$= 104.8 \times 10^3\ W$$

or

$BP = 104.8\ kW$

BSFC

$$BSFC = \frac{m_f}{B.P.}$$

m_f in kg/hr.

$m_f = \frac{4.4 \times 60}{10}$

$$= \frac{4.4 \times 60}{104.8 \times 10} = 0.041$$

$$= 0.251\ kg/kWhr$$

(iii)

$$BSAC = \frac{m_a}{B.P.}$$

$$= \frac{6 \times 60}{104.8} \quad 126$$

$$= 0.0572 \times 60$$

$$BSAC = 3.435 \ Kg/Kwhr.$$

(iv) Brake Thermal eff^n.

$$\eta_{bth} = \frac{B.P.}{m_f \times C.V.} \qquad \frac{KW}{\frac{Kg}{sec.} \times \frac{KJ}{Kg}}$$

$$= \frac{104.8 \ KW}{\frac{4.4 \ Kg}{10 \times 60 \ sec.} \times 44000 \ \frac{KJ}{Kg}}$$

$$\eta_{bth} = 0.3248$$

or

$$\eta_{bth} = 32.48 \ \%$$

(v)

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

$$= \frac{6 \ Kg/min.}{\frac{4.4}{10} \ Kg/min.}$$

$$\frac{A}{F} = 13.63$$

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(vi) Volumetric effn.

$$\eta_v = \frac{\text{Actual volume of Air/cycle}}{\text{Swept volume}} = \frac{V_a}{V_s}$$

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

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

$$P = \rho RT$$

$$\rho_a = \frac{P_a}{R_a T_a} = \frac{100}{0.287 \times 300}$$

$$\rho_a = 1.1614 \ \text{kg/m}^3$$

$$V_a = \frac{6 \ \text{kg/min}}{1.1614 \ \text{kg/m}^3}$$

$$V_a = 5.166 \ \text{m}^3/\text{min}$$

For 4-S engine

$$4500 \ \text{rpm} = \frac{4500}{2} = 2250 \ \text{cycle/min}$$

$$V_a/_{cycle} = \frac{5.166 \ m^3/min}{2250 \ cycle/min}$$

$$= 2.296 \times 10^{-3} \ m^3/cycle$$

$$V_s = \frac{\pi}{4} D^2 L K$$ 128

$$= \frac{\pi}{4} (0.09)^2 (0.08) \ 8 \ m^3$$

$$= 4.07 \times 10^{-3} \ m^3$$

$$\eta_{vol} = \frac{V_a}{V_s}$$

$$= \frac{2.296 \times 10^{-3}}{4.07 \times 10^{-3}}$$

$$\eta_{vol} = 0.5639$$

or

$$\eta_{vol} = 56.39\ \%$$

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 %

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Gate (4) The following details where noted in a test on a 4- cylinder, 4-S 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

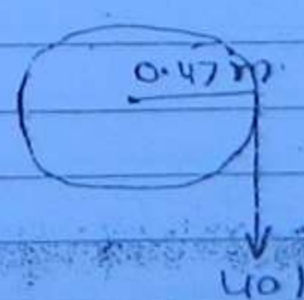
(i) Brake thermal effⁿ

(ii) Indicated thermal effⁿ

(iii) Indicated mean effective pressure &

(iv) Brake specific Fuel consumption

Solⁿ:-



$F = 9.81 \times 40$

$= 392.4$

circumference $= 2\pi r$

$3 = 2\pi r$

$r = 0.477$ m

$$T = F \times r$$

$$= 392.4 \times 0.477$$

$$= 187.35$$

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

$$= \frac{2\pi \times 1600 \times 187.35}{60}$$

$$BP = 31392 \text{ W}$$

(or)

$$BP = 31.39 \text{ KW}$$

i) Brake thermal effn.

$$\eta_{bth} = \frac{BP}{m_f \times C.V}$$

$$= \frac{31.39 \text{ KW}}{\frac{0.2}{60} \frac{Kg}{sec} \times 44000 \frac{KJ}{Kg}}$$

$$\eta_{bth} = 0.2156$$

(or)

$$\eta_{bth} \quad 21.56\ \%$$

(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

$$IP = \frac{P_m LANK}{2 \times 60}$$

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

$$0.8 = \frac{31.36}{I.P}$$

$$I.P = 39.24\ kW$$

$$39.24 = \frac{P_m \times (0.12) \times \frac{\pi}{4}(0.1)^2 \times 1600 \times 4}{60 \times 2}$$

$$P_m = 780.65\ kPa$$

(or)

$$P_m = 780.65 \times 10^3\ N/m^2$$ (or) $$P_m = 7.8065\ bar$$

(iv) Brake specific Fuel consumption

$$BSFC = \frac{m_f}{B.P}$$

(132)

$$= \frac{0.2 \times 60}{31.39} \frac{kg}{kW\,hr}$$

$$BSFC = 0.3826 \text{ kg/kW hr}$$

(5) The power output of a 6-cylinder, 4-stroke engine by measure means of a dynamometer for which the law is

$$bp = \frac{WN}{20000} \text{ kW}$$

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 $\qquad \frac{83}{17}$

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Coeffⁿ for discharge for venturi	0.6
atmp. pressure	1 bar
pressure drop across venturi	14.5 cm of [illegible]
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 effiⁿ.

** (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

$$bP = \frac{2\pi NT}{60} \text{ KW}$$

$$67.2 \times 10^3 = \frac{2\pi \times 2400 \times T}{60}$$

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$$T = \frac{67.2 \times 10^3 \times 60}{2\pi \times 2400}$$

$$T = 267.3 \text{ N-m}$$

(iii) ~~B STRE~~ % axcess of 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

$$V_f = 100 \text{ cm}^3 / 20 \text{ sec}$$

$$\rho_f = 831 \text{ Kg/m}^3$$

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

$$m_f = \rho_f \times V_f$$

$$= 831 \frac{kg}{m^3} \times \frac{100 \times 10^{-6}\ m^3}{20\ sec}$$

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$$m_f = 4.155 \times 10^{-3}\ kg/sec.$$

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

$$(P_1 - P_2) = 14.5\ cm\ of\ Hg$$

$$P = \rho R T$$

$$\rho_a = \frac{P}{RT}$$

$$P = \rho g h$$

$$\rho_a = \frac{100}{0.287 \times 300}$$

$$(P_1 - P_2) = 13.6 \times 10^3 \times 9.81 \times \frac{14.5}{100}$$

$$\rho_a = 1.16$$

$$P_1 - P_2 = 19345.3\ N/m^2$$

$\therefore$

$$m_a = 0.6 \times \frac{\pi}{4} (0.03)^2 \times \sqrt{2 \times 1.16 (19345.3)}$$

$$m_a = 0.089\ kg/sec.$$

$$\left(\frac{A}{F}\right)_{act.} = \frac{0.089}{4.155 \times 10^{-3}}$$

$$= 21.62$$

$$\% \text{ Excess} = \frac{\left(\frac{A}{F}\right)_{act.} - \left(\frac{A}{F}\right)_{st.}}{\left(\frac{A}{F}\right)_{st.}}$$

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$$\% \text{ excess} = \frac{21.62 - 15.536}{15.536}$$

$$\% \text{ excess} = 0.391$$

(or)

$$\% \text{ excess} = 39.1\ \%$$

(iv) Brake specific fuel consumption (BSFC)

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

$$BSFC = \frac{4.155 \times 10^{-3} \times 3600}{67.2}$$

$$BSFC = 0.222 \ \text{Kg/Kw hr.}$$

(v) Volumetric effn.

$$\eta_v = \frac{V_a/\text{cycle}}{V_s}$$

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

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

$$V_a = \frac{0.089 \text{ kg/sec}}{1.16 \text{ kg/m}^3}$$

$$V_a = 0.0766 \text{ m}^3/\text{sec}$$

For 4-s engine 2400 rot./min

$$= \frac{2400}{2} = 1200 \text{ cycle/min}$$

$$\frac{V_a}{\text{cycle}} = \frac{0.0766}{\frac{1200}{60}} \quad \frac{\text{m}^3/\text{sec}}{\text{cycle/sec}}$$

$$\frac{V_a}{\text{cycle}} = \frac{60 \times 0.0766}{1200} \text{ m}^3$$

$$V_a/\text{cycle} = 3.8315 \times 10^{-3} \text{ m}^3$$

$$V_s = \frac{\pi}{4} D^2 L K = \frac{\pi}{4} (0.1)^2 (0.12) \times 6$$

$$V_s = 5.652 \times 10^{-3} \text{ m}^3$$

$$\eta_v = \frac{V_a}{V_s}$$

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$$= \frac{3.8315 \times 10^{-3}}{5.652 \times 10^{-3}}$$

$$\eta_v = 0.6877$$

or

$$\eta_v = 68.77\%$$

***

(6) A 6-cylinder, 4-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 axcess with resp. to stoichiometic conditions.

(i) Estimate volumetric composition of dry exhaust gases.

(ii) determine bore & stroke dr

the engine. Taking stroke to Bore ratio $L/D = 1.5$.
Assume density of air as $1.298 \frac{kg}{m^3}$
and air contains 23 kg of oxygen in 100 kg.

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Soln:

4-Stroke

$K = 4$

$N = 1600$ rpm

$bp = 120$ kW

$CV = 43000$ kJ/kg

$\%C = 0.86$, $\% H_2 = 0.13$

$\eta_{vol.} = 0.8$

$\eta_{ith.} = 0.4$

$\eta_m = 0.8$

$\left(\frac{A}{F}\right) =$ 110 % Excess

$\rho_a = 1.298$ kg/m³

$\frac{L}{D} = 1.5$

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

$$= 30.45 - 14.5$$
$$= 15.95$$

Fuel + Air $\longrightarrow$ CO_2, H_2O

[CH] [14.5 + 15.95] O_2, N_2

$N_2 + O_2$ (St.) → Consumed; $N_2 + O_2$ → Excess

O_2 → Excess; N_2 → Total $(N_2^{(St.)} + N_2)$

te :- As axcess air is used stoichiometric oxygen will react with fuel resulting in CO_2 & H_2O whereas excess O_2 will come out in exhaust gases. as N_2 is innert it does not ~~g~~ undergo any reaction and hence stoichiometric N_2 & ~~CO_2~~ and excess N_2 will come out in exhaust gases.

$$\eta_m = \frac{bp}{ip}$$

$$ip = \frac{bp}{\eta_m}$$ 141

$$= \frac{120}{0.8} \text{ kW}$$

$$ip = 150 \text{ kW}$$

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

$$0.4 = \frac{150 \text{ kW}}{m_f \times 43000 \frac{kJ}{kg}}$$

$$m_f = 8.72 \times 10^{-3} \text{ kg/sec.}$$

For actual

$$\frac{m_a}{m_f} = 30.45$$

$$m_a = 30.45 \times m_f$$

$$m_a = 30.45 \times 8.72 \times 10^{-3}$$

$$m_a = 0.265531 \text{ kg/sec}$$

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

(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

$$1600 \text{ rpm} \Rightarrow 800 \text{ cycle/min}$$

$$= \frac{800}{60} \frac{\text{cycle}}{\text{sec}}$$

$$\frac{V_a}{\text{cycle}} = \frac{0.2045 \text{ m}^3/\text{sec}}{\frac{800}{60} \text{ cycle/sec}}$$

$$\frac{V_a}{\text{cycle}} = 0.0153 \text{ m}^3$$

$\therefore$

$$0.8 = \frac{0.0153 \text{ m}^3}{\frac{\pi}{4} D^2 L K \text{ m}^3}$$

$$\frac{\pi}{4} D^2 L K = \frac{0.0153}{0.8}$$

$$\frac{\pi}{4} D^2 (1.5D) \times 6 = \frac{0.0153}{0.8}$$

$D = 0.139\ m.$ 143

(or)

$D = 13.9\ Cm. \simeq 14\ Cm.$

$L = 1.5D$

$= 1.5 \times 14$

$L = 21\ cm.$

Exhaust Gases,

$CO_2,\ N_2,\ O_2$

$C + O_2 \longrightarrow CO_2$

$12\ Kg\ C \longrightarrow 44\ Kg\ of\ CO_2$

$1\ Kg\ C \longrightarrow \frac{44}{12}\ ,,\ CO_2$

$1\ Kg\ C$ produces $\longrightarrow \frac{44}{12}\ Kg\ of\ CO_2$

$0.86\ ,, \longrightarrow \frac{44}{12} \times 0.86$

$0.86\ Kg\ C\ ,, = 3.153\ Kg\ of\ CO_2$

F + Air

St. — Excess air

14.5 — 15.95

O_2 N_2 O_2 N_2

100 Kg of air — 23 kg of O_2

15.95 Kg " — $\frac{15.95 \times 23}{100}$

15.95 Kg Air = 3.6685 Kg of O_2

100 Kg air — 77 Kg of N_2

30.45 — " $\longrightarrow$ $\frac{77 \times 30.45}{100}$

30.45 Kg Air = 23.446 Kg of N_2

The above analysis is known as Gravimetric (or) Mass analysis

CO_2 — 3.153

O_2 — 3.668

N_2 — 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

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$$\frac{0.07165}{1.023} \times 100 = 7\%$$

$$\frac{0.1145}{1.023} \times 100 = 11.2\%$$

$$\frac{0.8373}{1.023} \times 100 = 81.8\%$$

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

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The main constituents are of fuel

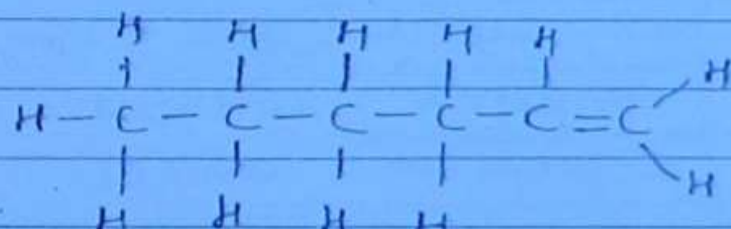
1. Paraffins (C_nH_{2n+2}), they are straight chain compounds

2. Olefins (C_nH_{2n}) , They are straight chain unsaturated Compounds

3. Naphthens (C_nH_{2n}), They are cyclic saturated Compounds.

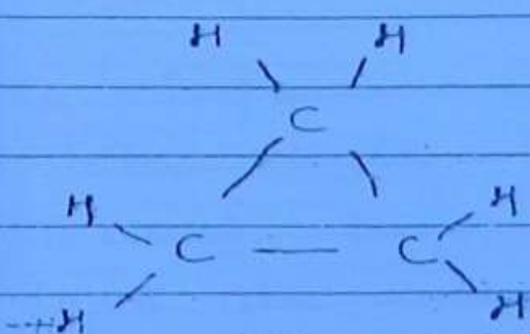
4. Aromatics (C_nH_{2n-6}), They are cyclic Unsaturated compounds.

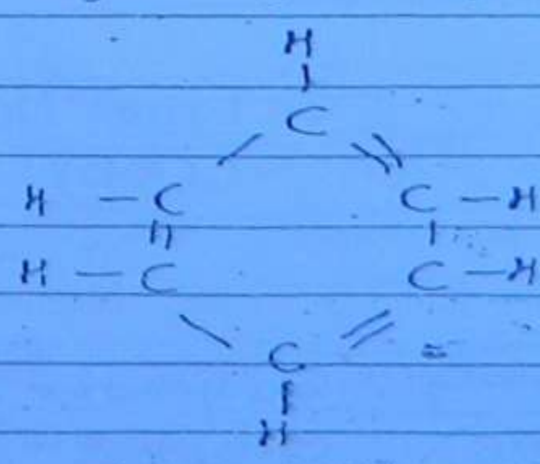
C − C − C − C − C



Paraffins (C_nH_{2n+2})

Olefins (C_nH_{2n})





Naphthens (C_nH_{2n}) Aromatics (C_nH_{2n-6})

* Parafins have high knocking Tandency when use in S.I. engine whereas aromatics have least knocking Tandency in S.I. engine fuel.
* Olifins 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)

* Initialy Cooprative Fuel Research Engine (CFR) operated on test fuel and the compression ratio is adjusted such that detination

occurs at say Top dead centre.
This compression ratio is fixed and now the engine operates on mixture of Iso-octane & n-heptane.

The percentage by volume of Iso 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. Iso-octane has very little knocking tendency hence its octane number is arbitraly assigned as 100. whereas n-heptane detonates easily And hence its octane number is assigned as 0 (zero).

Higher Octane no. means better S.I engine fuel.

Dopes in S.I. engine fuels

To improves anti-knocking properties
Tetra ethyle lead, Tetra methyle lead & Tersary 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 sometim.

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Motor Octane number
(MON)

It is measures anti-knocking performance under severe condition like large loads and high speeds

Research Octane Number
(RON)

It measures anti-knocking performance under mind conditions like less load and low speed.

*** Fuel sensitivity = RON − MON

*** Antiknock Index = $\frac{RON + MON}{2}$

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Cetane Number

or

Knock Rating of C.I. engine fuels

The refference fuels used in C.I. engine

* Cetane (n-hexa decane), ($C_{16}H_{34}$) (CN=100)
 Straight chain compount --

* α-methyle 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

* Alcohal
* Hydrogen
* CNG
* Biogas
* LPG etc.

Alcohal

Advantages:-

1. High octane No. (high compression ratio can be used)
2. It can be obtain from no. of sources.
3. Less emissions compared with petrol.
4. Low sulphur containt.

Disadvantages:-

1. Low calorific value
2. Ignition characterstics are poor
3. As alcohol is less volatile starting characterstics are poor. when refueling automobile headacks have been experienced.

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Hydrogen

Advantages:-

1. Low emissions.
2. Easily available
3. Hydrogen burns faster
4. Fuel leakage to environment is not a pollutent.

Disadvantages:-

1. Handeling 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 knocks 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~~ cheap.
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 %.

Sol^n:-

$$I.P. = \frac{P_m\, L A N x K}{60}$$

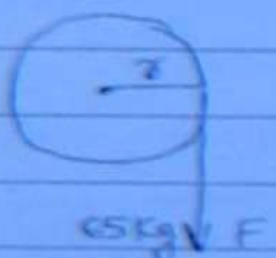
(2-stroke)

$$= \frac{300 \times 0.28 \times \pi \times (0.22)^2 \times 350}{4 \times 60}$$

$$I.P. = 18.63 \text{ KW}$$

$d = 1\,m.$

$r = 0.5\,m.$



$T = F \times r$

$= 65 \times 9.81 \times 0.5$

$F = 65 \times 9.81$ N

$$bp = \frac{2\pi N T}{60}$$

$$= \frac{2\pi \times 350 \times 65 \times 9.81 \times 0.5}{60}$$

$$bp = 11.68 \text{ KW}$$

The input energy = chemical energy of fuel

$= m_f \times C.V.$

mass flow rate of fuel

$$m_f = 4 \ kg/hr = \frac{4}{60} \ \frac{kg}{min}$$

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$$m_f \times C.V. = \frac{4}{60} \ \frac{kg}{min} \times 43000 \ \frac{KJ}{kg}$$

$$= 2866.67 \ KJ/min$$

Brake power in KJ/min $= 11.68 \times 60$

$$= 700.8 \ KJ/min$$

Heat absorbed by cooling water

heat carried $= m C_{P_w} \Delta t$

$$= \frac{500}{60} \ \frac{kg}{min} \times 4.187 \ \frac{KJ}{kg\text{-}K} \times (40-20) K$$

$$= 697.83 \ KJ/min$$

Heat carried by Dry exhaust Gases

$$= m_{de} \ C_{P_{de}} (\Delta t)$$

m_{de} = mass of dry exhaust gases.

Exhaust = dry exh. gases + H_2O

$m_f + m_a \longrightarrow$ Exhaust gases

$4 + 128 \longrightarrow 132$ Kg/hr.

total exh. gases

$\frac{A}{F} = 32$ $\quad m_a = 32 \times 4 = 128$

$m_f = 4$ Kg/hr.

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$$H_2 + \frac{1}{2} O_2 \longrightarrow H_2O$$

2 Kg H_2 $\longrightarrow$ 18 Kg of H_2O

1 Kg H_2 $\longrightarrow$ 9 Kg of H_2O

0.15 Kg of H_2 $\longrightarrow$ $\frac{0.15 \text{ Kg } H_2}{\text{Kg fuel}}$

1 Kg $\longrightarrow$ 0.15×9 $\frac{\text{Kg of } H_2O}{\text{Kg fuel}}$

$\longrightarrow$ $\frac{1.35 \text{ Kg } H_2O}{\text{Kg fuel}}$

As 4 Kg/hr. of fuel is used total water vapour produce

$= 4 \times 1.35$ Kg H_2O/hr.

$= 5.4$ Kg H_2O vapour/hr.

* Total dry exhaust gas $= 132 - 5.4$
$= 126.6 \ Kg/hr.$

159 $\quad m_{de} = 126.6 \ Kg/hr.$

$\therefore$

* $$Q_{de} = m_{de} \cdot C_{pde} (\Delta t)$$

$$= \frac{126.6}{60} \frac{Kg}{min} \times 1 \times (40 - 20)$$

$$Q_{de} = 801.8 \ KJ/min.$$

Heat carried by steam

$$= m C_{pw} (100 - 20) + mL + m C_{ps} (400 - 100)$$

$$\underbrace{20 - 100}_{SH} \rightarrow \underbrace{100}_{LH} - \underbrace{400}_{Steam}$$

$$= m [C_{pw} \times 80 + L + C_{ps} \times 300]$$

$$= \frac{5.4 \ Kg}{60 \ min} \left\{ 4.187 \times 80 + 2250 + 2.1 \times 300 \right\}$$

$$= 289.4 \ KJ/min.$$

Heat balance sheet

Parameters	Heat (KJ/min)	
1. Fuel	2866.67 (Total)	
2. bp	700.8	2489.83
3. Cooling H_2O	697.83	
4. Dry exhaust gases	801.8	
5. Steam	289.4	

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* Heat unaccounted $= 2866.67 - 2489.83$
$= 376.84$ 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 (ip.) of a multicylinder engine.
* In this test speed is kept constant so that frictional power remains constant.

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All cycle firing → $ip_1 + ip_2 + ip_3 + ip_4 = \sum_{i=1}^{4} bp_i + fp$

First cut-off → $ip_2 + ip_3 + ip_4 = \sum_{i=2}^{4} bp_i + fp$

$\therefore$

$$ip_1 = \sum_{i=1}^{4} bp_i - \sum_{i=2}^{4} bp_i$$

$$ip = ip_1 + ip_2 + ip_3 + ip_4$$

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.

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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 effn.

Soln:-

$$ip_1 = 3037 - 2102$$
$$= 935 \text{ KW}$$

$$ip_2 = 935 \text{ KW}$$

$$ip_3 = 937 \text{ KW}$$

$$ip_4 = 939 \text{ KW}$$

$$\dot{i}p = \dot{i}p_1 + \dot{i}p_2 + \dot{i}p_3 + \dot{i}p_4$$

$$= 935 + 935 + 937 + 939$$

$$\dot{i}p = 3746 \text{ KW}$$

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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 enter 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 thermodymic cycles is higher then Actual efficiency of engines.

Reason:- Air is not a perfect gas.

Soln:-

Assertion (✓) . Reason (✗)

(C)

* (4) which of the following factors increase detonation in S.I. engines

(1) Increase spark advance

(2) Increase speed

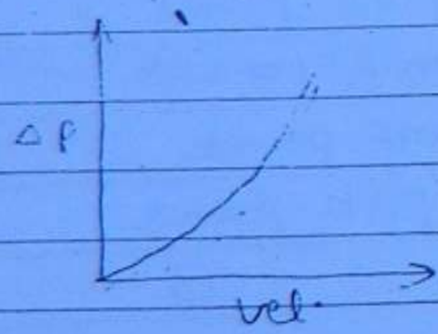
(3) Increase A/F ratio beyond stoichiometric strength

(4) Increased compression ratio. 164

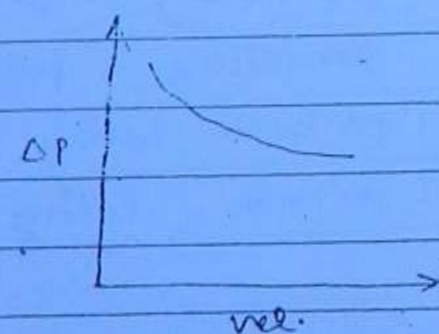
Sol^n:- (1) & (4)

Prob. (5) which of the following curve is proper representation of pressure differential (ΔP) Vs velocity of Air at the throat of Carburetor.

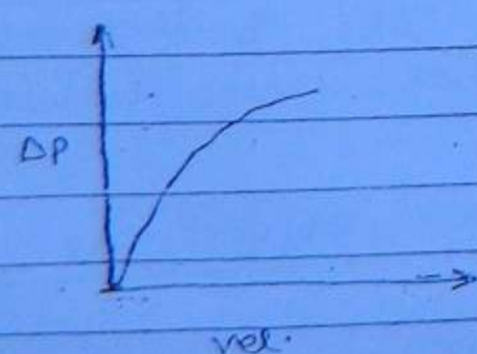
(a)



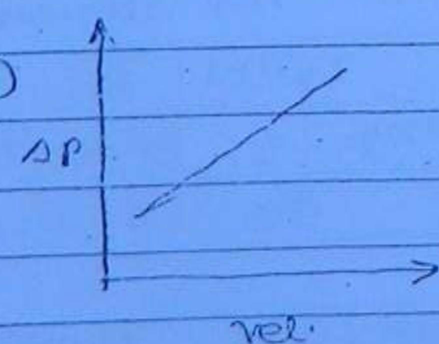
(b)



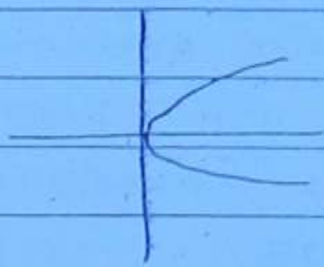
(c)



(d)



Sol^n:-



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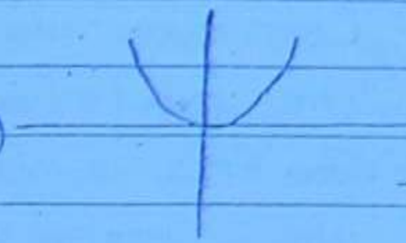

$y^2 = 4ax$ $\qquad\qquad$ $x^2 = 4ay$

$$v = \sqrt{\frac{2\Delta P}{\rho_a}}$$

$$v^2 = \frac{2\Delta P}{\rho_a}$$

$$x^2 = 4a(y)$$

** (6) Match the following

List I (Elements of Carburetor)	List II (Rich mixture requirement)
(A) Idling system	(1) To Compensate for dilution of charge
(B) Economiser	(2) For cold starting
(C) Acceleration pump	(3) For meeting power range of operation
(D) Chock	(4) For meeting Rapid opening of throttle

Sol^n:- 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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Sol^n:- 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.

Sol^n:- Assertion (✓) , Reason (X)

3 times in IES (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 25cc its volumetric effn. is 0.88 and mech. effn. is 0.9. what is the volume of mixture taken in per cycle.

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Soln. —

$$\eta_{vol} = \frac{Va/cycle}{V_s}$$

$$Va/cycle = \eta_v \times V_s$$

$$= 0.88 \times 300$$

$$Va/cycle = 264$$

(11) The correct sequense 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 then 2-S engine (168)

* Volumetric effⁿ. of 4-S engine is greater then 2-S engine because of more time for induction.

* The power of 2-S engines is more then the 4-S engines for same size.

15. 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
In 2-S → Ports
& 4-S → Valves

(13) If methane undergoes combustion with stoichiometric quantity of air then A/F ratio on mass basis is.

Soln:-

CH_4 169

$$C - \frac{12}{16} = 0.75$$

$$H - \frac{4}{16} = 0.25$$

$$\left(\frac{A}{F}\right)_{st.} = \frac{100}{23}\left[\frac{8}{3}C + 8H\right]$$

$$= \frac{100}{23}\left[\frac{8}{3}(0.75) + 8(0.25)\right]$$

$$\left(\frac{A}{F}\right)_{st.} = 17.4$$

(14) When a 4-S 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:-

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

$$\theta = \frac{1.5 \times 10^{-3} \times 360 \times 2000}{60}$$

$$\theta = 18$$

(15) A hypothetic pressure diagram for a C.I. engine is shown in the fig. The diesel knock is generated during the periode

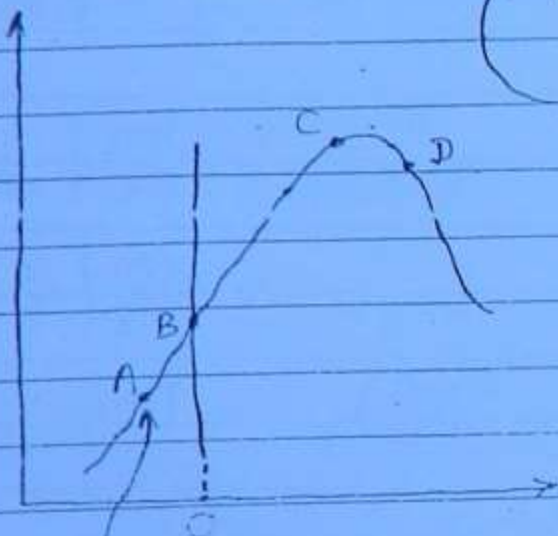
(a) AB

(b) BC

(c) CD

(d) After D

770



start of ignition

760mm

2400 rpm

HMT

TD

RAC

Supercharging

We know that

$$IP = \frac{P_m L A N}{60}$$

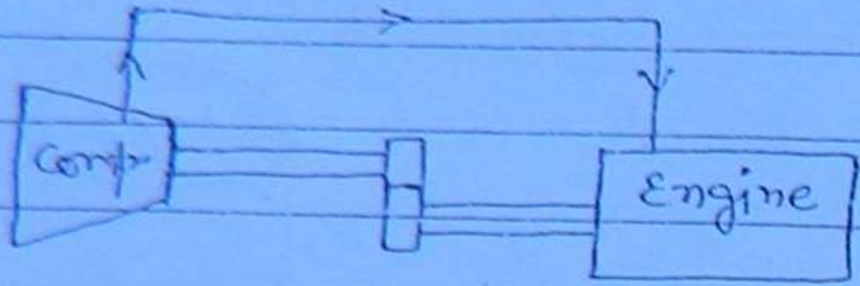
Indicated power can be increased by increasing all the terms in Numerator but increase in L & A results in larger engine and increase in N results in increasing frictional losses and hence to best obtion to increase I.P. is to increase mean effective pressure. This can be done by increasing the pressure of charge at the inlet to the engine cylinder.

Methods of Supercharging :-

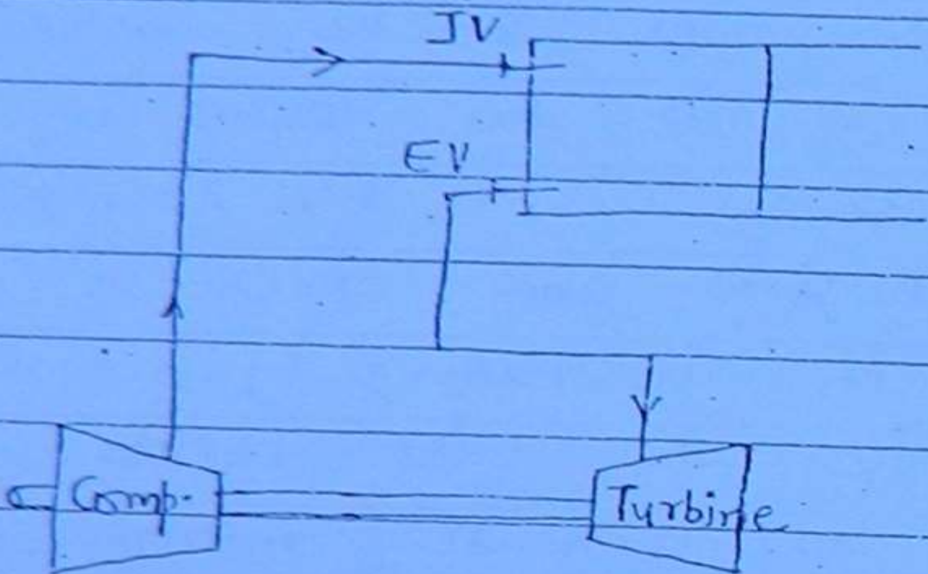
1. Mechanical Supercharging
2. Turbo charging.

In mechanical supercharging, some engine power is utilised to run the compressor. Whereas in turbocharging exhaust gas energy is utilised in running compressor.

(172)



mechanical Supercharging



Turbo charging

AS

Assignment (173)

(1) An experimental 4-S petrol engine of 1710 cm^3 (V_s) Capacity. is to develope max. power at 5400 rpm. the volumetric effn. is assumed to be 70% and A/F ratio is 13. Two carborators 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 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 develope a power at 4000 rpm. The Vol. effn. is 0.75 and A/F = 14 The venturi throat dia. is 30mm. 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 6-cylinder, 4-S 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.

Soln:-

$C = 0.82$

$H_2 = 0.18$

$$\left(\frac{A}{F}\right)_{st.} = \frac{100}{23}\left[\frac{8}{3}C + 8H_2\right]$$

$$= \frac{100}{23}\left[\frac{8}{3}(0.82) + 8(0.18)\right]$$

$$\left(\frac{A}{F}\right)_{st.} = 15.76 \quad 178$$

Actual A/F ratio

Ex. gas	%	Kg exh.
CO_2	16.45	0.1645
O_2	3.85	0.0385
N_2	79.7	0.797

$$C + O_2 \longrightarrow CO_2$$

$$12\ \text{Kg of C} \longrightarrow 44\ \text{Kg } CO_2$$

$$0.1645\ \text{Kg } CO_2 / \text{Kg exhaust gas}$$

$$\frac{12 \times 0.1645}{44} \quad \frac{\text{Kg C}}{\text{Kg Exh. gas}}$$

$$0.04486 \quad \frac{\text{Kg C}}{\text{Kg exhaust gas}}$$

We know that
0.82 kg of C is present in per kg fuel.

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$$\frac{\text{kg C}}{\text{kg fuel}} = 0.82$$

$$\frac{\text{kg C}}{\text{kg Ex. gas}} = 0.04486$$

$$\frac{\text{mass of exh. gas}}{\text{kg fuel}} = 18.28$$

We know that
1 kg exhaust gas contains 0.797 kg of N_2 therefor 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 — 77
?. — 14.57