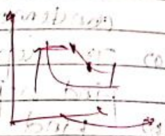
$R.E = m(h_1 - h_4)$

$= 0.025 \times (164 - 80)$

$= 2.1\ KW$

$COP = \frac{2.1}{(200 - 164) \times 0.025}$

$= 2.33$



$(COP)_{H.P} = 3.33$

Compressor

- Reciprocating
- Centrifugal
- Axial

Parameter	Reciprocating	Centrifugal	Axial
	Max.		
r_p/stage	5-8	2-4	1.2 to 2
mass flow rate	min.	Intermediate	max.

Reciprocating Compressor

L = stroke length

D = Diameter (Bore) of cylinder or piston

$V_{swept} = V_{stroke} = V_s$ = stroke volume.

$P_s = P_1$ = Suction pressure (Inlet or entry press).

$P_d = P_2 = P_3$ = Discharge (delivery pressure) (exit press)

V_c = clearence volume $= V_3$

$\frac{V_c}{V_s}$ = Clearence factor or clearence ratio.

Volumetric eff of comp $= \frac{V_{act}}{V_{stroke}} = \frac{V_{act}}{V_{theo.}}$

$\eta_v = \frac{V_1 - V_4}{V_1 - V_3}$

$\eta_v = 1 + C - C\left(\frac{P_d}{P_s}\right)^{1/n}$

N = rpm of crank or crank shaft

$\left[\eta_v \times \frac{\pi}{4} D^2 \frac{LN}{60}\right] \rho_1$ = mass flow rate

$W = \frac{n}{n-1} \times mRT_1 \left[\left(\frac{P_d}{P_s}\right)^{\frac{n-1}{n}} - 1\right]$

$= \frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{n-1/n} - 1\right]$

$V_1 - V_4 = V_{act}$ m^3/sec

$V_{act} = \eta_v \times V_{st}$

Volume $= \eta_v \left[\frac{\pi}{4} \times \frac{D^2 \times L \times N}{60}\right]$

$P_1 = P_4 = P_{suc}$

$T_1 = T_4 = T_{suc}$

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

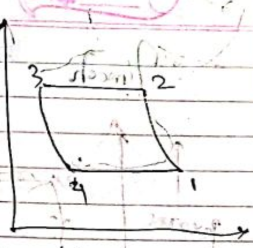
$m = \rho \times V_{act}$

$$m = \frac{P_1}{RT_1} \times V_{act}$$

Que A single acting single cylinder reciprocating air compressor running at 7.5 rev/sec, takes in air at 100 kPa, 27 °C. The compressor delivers air at 600 kPa at a flow rate of 0.12 m^3/sec measured at suction condition. The % clearence is 4 and index of comp & index of expansion is 1.2. Find (a) volumetric η, (b) Bore + stroke of cyl. if B/stroke = 0.8

$P_4 = P_1 = 100$ kPa, $T_1 = 300$ kelvin

$P_3 = P_2 = 600$ kPa, $V_1 - V_4 = 0.12 m^3/sec$



$\frac{V_c}{V_s} = \frac{4}{100}$

$V_1 = \frac{\pi}{4} \times D^2 \times D/0.8$

$V_3 = 0.04 \times \frac{\pi}{4} \times \frac{D^3}{0.8}$

$$P_1(V_1 - V_4) = mRT_1$$

$100 \times 0.12 = m \times 0.287 \times 300$

$$m = 0.14 kg/sec$$

$\eta_{vol} = 1 + 0.04 - 0.04\left(\frac{600}{100}\right)^{1/1.2}$

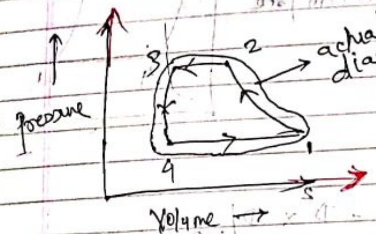
$= 86.2\%$

~~$V_{th} = \frac{V_{act}}{0.8619} = 0.14 m^3/sec$~~

$0.14 = \frac{\pi}{4} \times D^2 \times \frac{D}{0.8} \times 7.5$

$D = 0.5214\ 390$ m, $l = 0.65\ 48$ m

η_{mech} =



Ideal Indicator Diagram

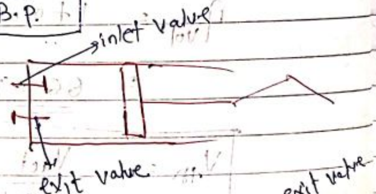
Actual diagram > Ideal diagram

$$(W_{compressor})_{actual} > (W_{compressor})_{ideal}$$

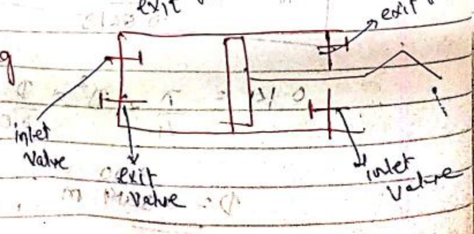
In compressor B.P > I.P.

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

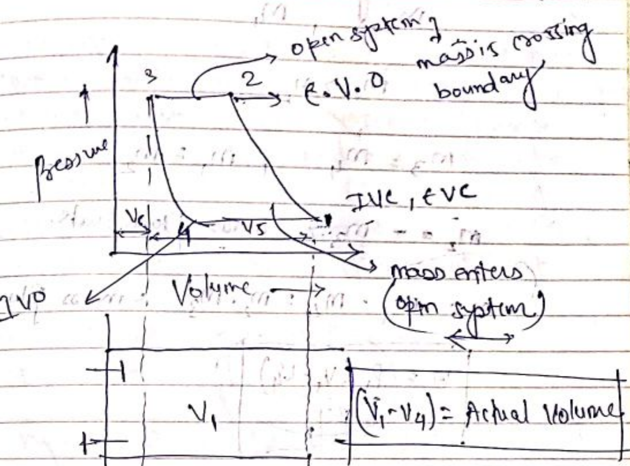
i) Single acting compressor →



ii) Double acting compressor



$(\dot{m})_{Double\ acting} \sim 2(\dot{m})_{single\ acting}$



In compressor valves are opened due to pressure difference while in I.C. engine valves are opened & closed with the help of CAM.

I.V.O. $P_{atm} > P_{cyl}$

I.V.C. $P_{cyl} \geq P_{atm}$

E.V.O. $P_{cyl} > P_{discharge}$

E.V.C. $P_{cyl} < P_{discharge}$

Scanned with CamScanner

$\frac{n}{n-1} \times P_1 (V_1 - V_4) [(r^{n-1/n})]$

from 4 to 1, mass in Tsing continuously

m_4 , m_1

$m_1 > m_4$

$m_3 = m_4$, $m_1 = m_2$

$m_2 - m_3 \Rightarrow$ mass flow rate

$m = m_1 - m_4 = m_2 - m_3$ = mass flow rate

$$m = \frac{P_1 (V_1 - V_4)}{R T_1}$$

Que Find the power reqd by comp. if $\eta_{mech} = 80\%$.

(ii) If comp. is double acting the power.

Soln I.P = $\frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$

$= \frac{1.2}{0.2} \times 100 \times 0.12 \left[(8)^{0.2/1.2} - 1 \right]$

$= 25.05$ kW

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

B.P. = $\frac{25.05}{0.8}$ = 31.32 kW

(B.P.) Double acting = 2×31.32

= 62.64 kW

Que A refrigeration compressor of 50mm diameter 40mm stroke operated at speed of 1460rpm between condenser and evaporator pressure limits of 1.219 MPa and 0.151 MPa. The clearance ratio is 5%. Ratio of specific heat of refrigerant is 1.18. Specific volume of refrigerant at suction is 0.11 m^3/kg. Enthalpy change in evaporator is 93.1. Find the rf. load.

Soln $W = \frac{n}{n-1} \times P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{n-1/n} - 1 \right]$

$C_p - C_v = R$

$C_v [\gamma - 1] = R$

$C_v = R / 1.18$

$C_p / C_v = 8.314 / 1.18$

$\frac{C_p}{C_v} = 1.8$

$C_v = \frac{C_p}{1.8}$

C_v ...

$P_1 (V_1 - V_4) = R T_1$

$151 (V_1 - V_4) = 0.8 (C_p - C_v)$

$151 (V_1 - V_4) = 0.8 (C_p - \frac{C_p}{1.8}) T$

~~$151(V_1 - V_4) = \frac{0.18}{1.18} \times 93.7$~~

~~$V_1 - V_4 = 0.0946$~~

~~$\eta = \frac{0.274}{\frac{\pi}{4} \times 0.05^2 \times 0.04 \times \frac{1460}{60}}$~~

$W_{comp} = \frac{n}{n-1} \times P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$

Refrigeration load $= R = m \times (\Delta h)$

$m = \rho V_{act}$

$= \rho \eta_v \times V_{stroke}$

$= \rho \eta_v \left[\frac{\pi}{4} d^2 l \times \frac{N}{60} \right]$

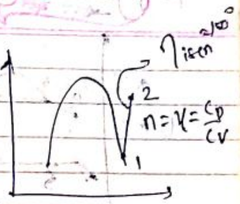
$= \frac{1}{v} \eta_v \left[\frac{\pi}{4} d^2 l \times \frac{N}{60} \right]$

$= \frac{1}{0.11} \times \eta_v \left[\frac{\pi/4 \times 0.05^2 \times 0.04 \times 1460}{60} \right]$

$\eta_v = 1 + c - c \left(\frac{P_d}{P_s} \right)^{1/n_e}$

$= 1 + 0.05 - 0.05 \left(\frac{1.219}{0.151} \right)^{1/n_e}$

$\eta_v = 0.756$



$m = \frac{0.756}{0.11} \left[\pi/4 \times 0.05^2 \times 0.04 \times \frac{1460}{60} \right]$

$m = 13.13 \text{ kg/sec} \times 10^{-3}$

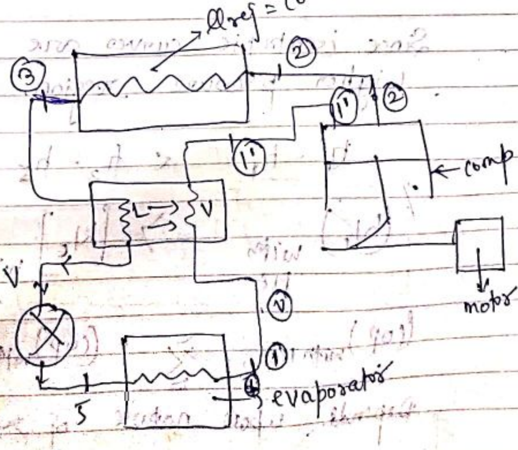
$R.E. = 13.13 \times (\Delta h) \times 10^{-3}$

$= 1230.72$ Watt

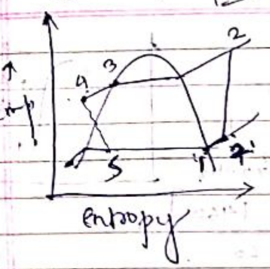
$= 1.230$ kW

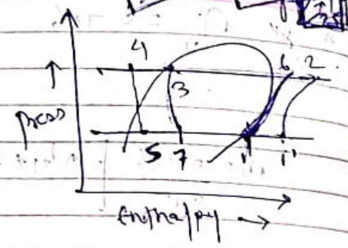
V/ Comp cycle with liquid

Vapour compression cycle with liquid

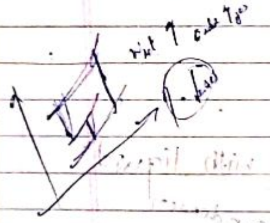


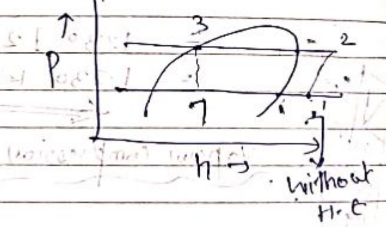
Temp-entropy Diagram





V/C without H.E.	V/C with H.E.
$(R.E)_{\text{with HE}} = h_1 - h_7$	$(R.E)_{\text{with HE}} = h_1' - h_5$





Since isentropic curves are diverging higher pressure region.

$h_1' - h_1 < h_2 - h_6$

$(W_c)_{\text{with HE}} > [W_c]_{\text{without HE}}$

$(COP)_{\text{with HE}} \gtrless (COP)_{\text{without HE}}$

Depends upon nature of refrigerant.

3 → 4

$(Q_{ref})_{\text{with HE}} > (Q_{ref})_{\text{with HE}}$ out

$h_2 - h_3 > h_6 - h_5$

Heat lost by liquid = Heat gained by vapour

$m\, c_{pL}(T_3 - T_4) = m\, c_{pV}(T_1' - T_1)$

$T_3 - T_4$ = Degree of subcooled

$T_1' - T_1$ = Degree of superheat

$c_{pL} > c_{pV}$

$T_3 - T_4 < T_1' - T_1$

(1) $h_1' = h_1 + c_{pV}(t_1' - t_1)$

(2) $S_1' = S_1 + c_{pV} \ln \frac{T_1'}{T_1}$

(3) $h_4 = h_3 - c_{pL}(T_3 - T_4)$

$(\eta_{isen})_{comp} = \frac{W_{rev}}{W_{irrev}}$

Que A 15 TON freon refrigerator plant has twin cyl. single acting recip. compressor of bore 1.5 times the stroke. It runs at 960 rpm. The temp of refr. leaving condenser is 298 kelvin, while temp of refrigerant leaving evaporator is 268 Kelvin. The condenser and evaporator pressure are 7.45 & 2.19 bar resp.

(i) Mass flow rate ✓

(ii) Power reqd to operate plant.

(iii) COP

(iv) Comp. bore & stroke, if volumetric η is 90%.

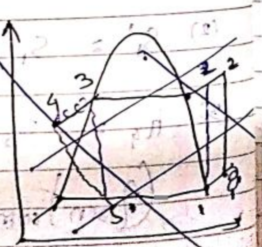
Sol^n →

Sat Temp	Press/bar	V_g	h_f	h_g	s_f	s_g
263	2.19	0.0767	26.9	183.2	0.106	0.702
303	7.45	0.0235	64.6	199.6	0.239	0.685

c_{pv}	c_{pL}
0.615	0.963
0.615	0.963

R.E = 15 × 3.5167

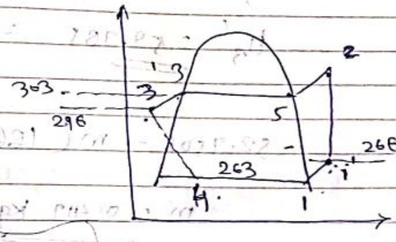
= 52.7505



$h_1 = 183.2$

$s_1 = 0.702 \pm 0.965 \ln T$

In heat exchanger superheating is done outside the evaporator & subcooling is done outside the condenser.



$$RE = m(h_1' - h_4)$$

$$h_1' = h_1 + c_{pv}(t_1' - t_1)$$

$$= 183.2 + 0.615(268 - 263)$$

$$h_1' = 186.275$$

$$s_1' = 0.702 + 0.615 \ln\left(\frac{268}{263}\right)$$

$$s_1' = 0.713 = s_2$$

$$0.713 = 0.685 + 0.615 \ln\left(\frac{x}{303}\right)$$

$$T_2 = 317.414 \text{ kelvin}$$

$$h_2 = 199.6 + 0.615(317.414 - 303)$$

$$= 208.46 \text{ kJ/kg}$$

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$\therefore h_3 = 64.6$

$h_{3'} = 64.6 - 0.963\,(268 - 263)$

$h_3' = 59.785 = h_4$

$52.7505 = \dot m\,(186.275 - 59.785)$

$\dot m = 0.417$ kg/sec ✓

$W_{net} = 0.417 \times (208.46 - 186.275)$

$= 9.25$ kW

$COP = \frac{52.75}{9.25} = 5.70$ ✓

$\dot m = \eta \times \frac{\pi}{4} D^2 \times l \times \frac{N}{60} \times \frac{1}{v_1'} \times 2$

$[P(V_1 - V_4) = mRT]$

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

$\frac{0.0767}{263} = \frac{V_1'}{268}$

$V_1' = 0.0781$

$0.417 = \frac{\pi}{4} \times (1.5 \times l)^2 \times l \times \frac{960}{60} \times \frac{1}{0.0781} \times 2$ ($\times 0.9$)

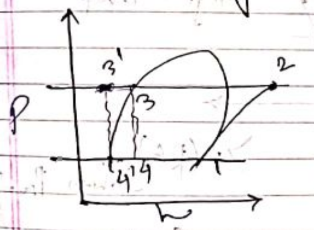
$l = 86$ mm

$D = 129$ mm

Do some ques → P.K. Nag / R.S. Khurmi } 8 to 10.

V/C : Analysis Theory. (Important).

Effects of subcooling the refrigerant before throttling

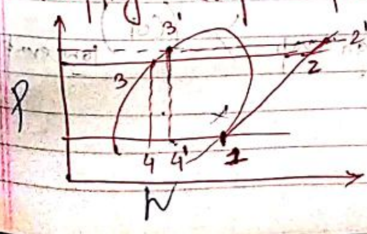


$RE \uparrow = \dot m(h_1 - h_4')$

W_c = same

COP ↑ses

Effect of increasing condensor pressure keeping evap. press. constant



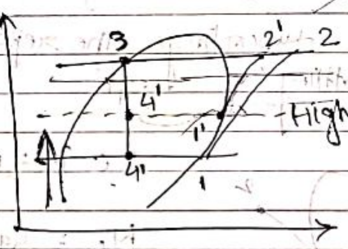
$(RE) = \dot{m}(h_1 - h_4') \downarrow$ ses.

$hl_c = \dot{m}(h_2' - h_1) \uparrow$ ses.

COP $\downarrow$ ses

$(COP)_{winter} > (COP)_{summer}$.

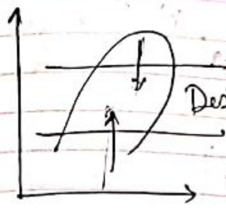
Case (ii) Effecting of increasing evaporator pressure keeping throttling at condenser press. constant

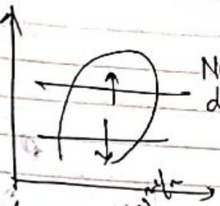


$(RE)_{high\ Evap\ press.} > (RE)_{lower\ P_{evap}}$

$(hl_c)_{high\ evap\ press} < (hl_c)_{low\ evap\ press.}$

$(COP)_{high\ evap} > (COP)_{low\ evap}$.



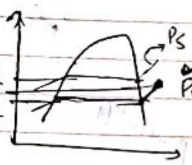


$RH = \frac{P_v}{P_s} = \frac{0.02062}{0.0421} = 46.6\%$

$\omega = \frac{0.622[0.02062]}{1.013 - 0.0206}$

DPT = 18°C
DBT = 30°C

$\omega = \frac{0.622 \times 0.0421}{1.013 - 0.0421}$



$\ln 2.053 = 19.013 - \frac{5325}{T}$

$h_{dry\ air} = 1.005(30 - 0)$

$\omega_B = \frac{0.622 \times 4.21}{(96 - 4.21)}$

$P_v = 0.02062$ ✓
$P_s = 0.04241$ ✓

$P = P_{Air} + P_{vap}$

$P_{atm} = 960$ mBar
DBT = 30°C

$P_s = 4.21$ RH = 0.60
$\omega = ?$

$R.H = \frac{P_V}{P_S}$

$P_A V_A = M_a R_A T_A$

$\frac{P_a}{R_A T_A} = \frac{M_a}{V_A}$

$\ln P_s = 19.013 - \frac{5325}{303}$

$\frac{93.471}{0.287 \times 303} = \frac{2.59}{0.283 \times 303}$

$\rho_1 = 1.07$

$0.6 = \frac{P_v}{P_s}$, $P_v = 2.53$

$\omega = \frac{0.622 \times 2.53}{960 \times 10^{-3} \times 100 - 2.53}$