$h_3 = h_4 = (h_f)_{40°C} = 255.73$ kJ/kg

Now, $m(h_1 - h_4) = 10 \times 3.5$

$$m = \frac{1.0 \times 3.5}{398.78 - 255.73} = 0.2447 \text{ kg/s}$$

Work reqd. $= m(h_2 - h_1)$

$= 0.2447(423.82 - 398.78)$

$= 6.1273$ kW

Heat rejected in the condenser

$= m(h_2 - h_3)$

$= 0.2447(423.82 - 255.73)$

$= 41.1316$ kW.

$$C.O.P = \frac{N}{W} = \frac{35}{6.1273} = 5.7121$$

R-12 $h_1 = 187.5$ kJ/kg

$s_1 = s_2$

$s_{g1} = (s_{sup})_2$

$$0.6966 = \left[s_g + c_p \ln \frac{T_2}{T_s}\right]_2$$

$$0.6966 = 0.6825 + 0.776 \ln \frac{T_2}{313}$$

$T_2 = 318.74$ K $(45.74°C)$

$h_2 = h_{g8} + C_p(T_2 - T_s)$
$= 203.2 + 0.776(45.74 - 40)$
$= 207.65$ KJ/kg

$h_3 = h_4 = (h_g)_{40} = 74.6$ KJ/kg

$m = \dfrac{10 \times 3.5}{(h_1 - h_4)}$

$= \dfrac{35}{187.5 - 74.6} = 0.31$ kg/s

Work reqd $= m(h_2 - h_1)$
$= 6.2465$ kW

Heat rejected in the Condenser
$= m(h_2 - h_3) = 41.2455$ kW.

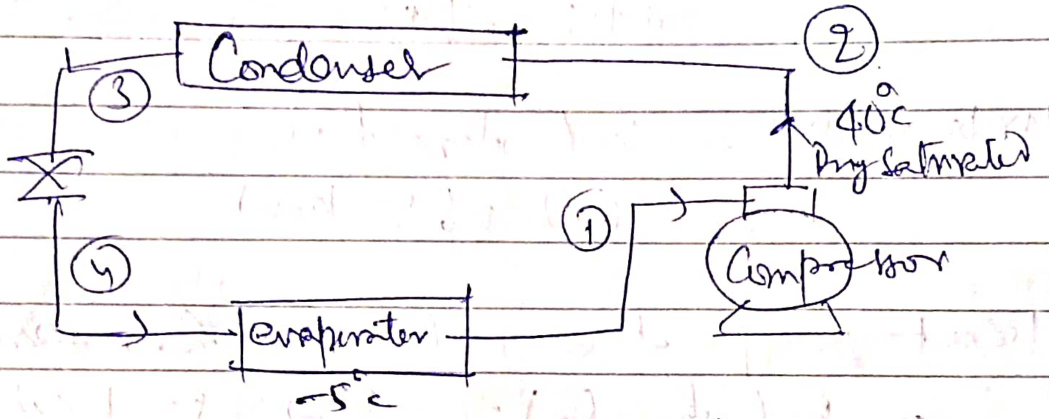
$C.O.P = \dfrac{N}{W} = \dfrac{35}{6.2465}$

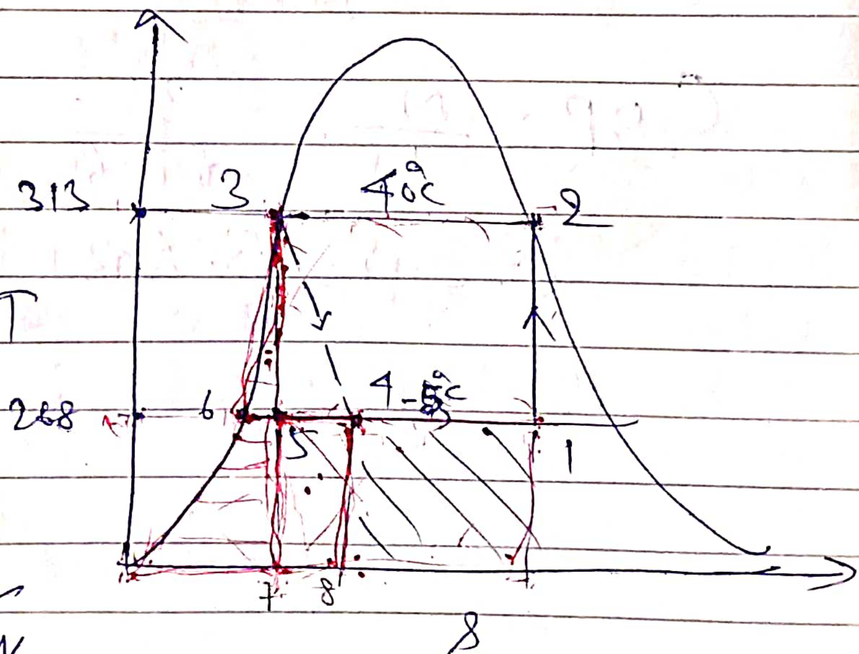
$C.O.P = 5.6031$

(3)

Q → A ref. m/c uses R12 as a working fluid. The tempr of R12 in the evaporator coil is $-5^\circ C$ and the gas leaves the compressor as dry saturated at a tempr of $40^\circ C$. The mean specific heat of liquid R12 b/w the above tempr is 0.963 KJ/kgK. Enthalpy of evaporation at $40^\circ c$ is 203.2 KJ/kg. Find C.O.P.

Soln:-





$h_2 - h_3 = 203.2$ KJ/kg

= Area under 2-3.

and

$(s_2 - s_3) \times T_2$ = Area under 2-3

$(s_2 - s_3) = \frac{203.2}{313}$

$= 0.6492$ KJ/KgK

$\Delta\, 3-5-6 = \frac{1}{2} \times \text{base} \times \text{ht.}$

$\text{Base} = s_5 - s_6$

$= s_3 - s_6 \quad [\because s_3 = s_5]$

$= C_{p_e} \ln \frac{T_3}{T_6} = C_p \ln \frac{T_3}{T_6} \quad [T_5 = T_6]$

$\text{height} = (T_3 - T_5)$

Now

$h_3 = h_4$

$\Delta\, 3-5-6 = \square\, 4-5-7-8$

$\frac{1}{2} \times C_{p_e} \ln \frac{T_3}{T_6} \times 45 = 268 \times (s_4 - s_5)$

$(s_4 - s_5) = \frac{1}{2} \times 0.963 \ln \left(\frac{313}{268}\right) \times 45$

$= 3.3631$

Work done = area 4-1-2-3-6-4

= area 3-6-5 + area 1-2-3-5

$W = 0.6492 \times 45 + 3.3631$

$= 32.58$ kW

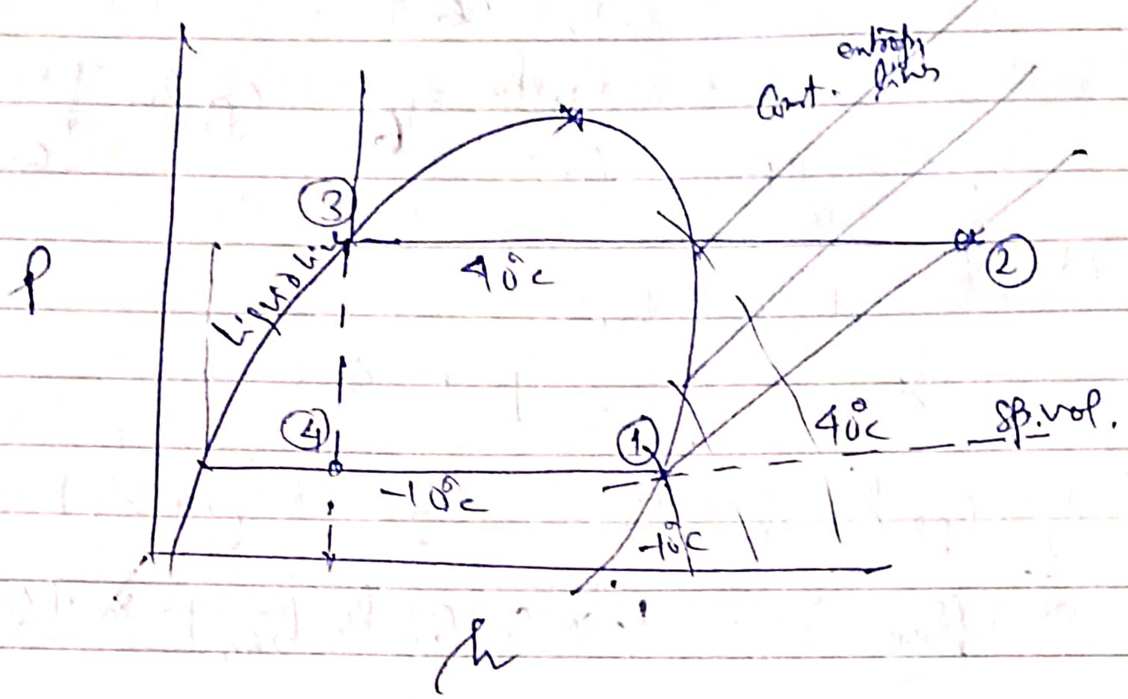
$N = 0.6492 \times 268 - 3.3631$

$= 170.623$

$C.O.P = \frac{N}{W} = \frac{170.623}{32.58}$

$\boxed{C.O.P = 5.237}$ Ans

For easy calculation we use P-h diagram.



*** Throttling is an expansion through a narrow restricted passage without doing any external work under adiabatic conditions.

In case of expansion, pressure drops. Due to loss in pressure there is drop in enthalpy ($h = U + Pv$) which is equivalent to amount of frictional heat generated. i.e. there is no change in enthalpy.

(4)

Q-D A Simple Refrigerant 134a (Tetrafluoro ethane) heat pump for space heating operates between temp limits of 15 and 50°c. The heat required to be pumped is 100 MJ/hr. Determine:

(a) The dryness fraction of refrigerant entering the evaporator.

(b) The discharge temp assuming the sp. heat of vapours as 0.996 KJ/kgK.

(c) the theoritical piston displacement of the Compressor.

(d) The theoritical horse power of the Compressor.

(e) The C.O.P.

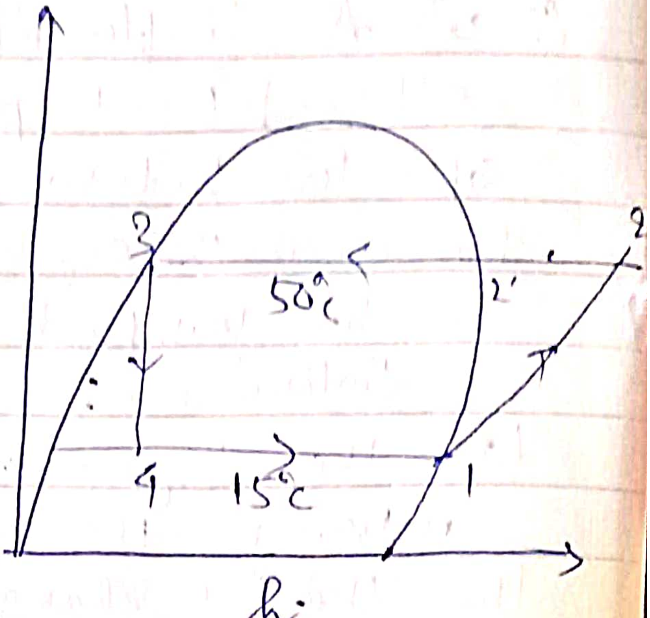
The sp. volume of refrigerant 134a Sat. vapour at 15°c is 0.04185 m³/kg. The other relevant properties of R134a are given below.

Sat. temp	Pressure	Sp. enthalpy		Sp entropy ~~enthalpy~~	
		h_f	h_g	S_f	S_g
°C	MN/m^2	KJ/kg		KJ/kgK	
15	0.4887	220.26	413.6	1.0729	1.7439
50		271.97	430.4	1.241	1.7312

$h_3 = h_4 = (h_{4f}) + x\, h_{fg}$

$x = \cancel{h_3 = h_{4f} + x\, h_{fg}}$

Sol^n:



$h_1 = (h_g)_{15} = 413.6 \ \mathrm{KJ/kg}$

$h_3 = h_4 = (h_f)_{50^\circ C} = 271.97 \ \mathrm{KJ/kg}$

$s_1 = s_2 = (s_g)_{15^\circ C} = 1.7439 \ \mathrm{KJ/kg}$

(a) Dryness fraction of refrigerant entering the evaporator

$$x = \frac{h_3 - h_{4f}}{h_1 - h_{4f}} = \frac{271.97 - 220.26}{413.6 - 220.26} = 0.2675$$

(b) $s_2 = s_{2'} + c_p \ln \frac{T_2}{T_{2'}}$

$1.7439 = 1.7312 \times 0.996 \ln \frac{T_2}{323}$ $\qquad s_{g'} = (s_g)_{50^\circ C} = 1.7312$

$\therefore T_2 = 327.145 \ \mathrm{K} = 54.145^\circ c$

$h_2 = h_{2'} + c_p (T_2 - T_{2'})$

$= 430.4 + 0.996 (54.15 - 50)$

$= 434.5 \ \mathrm{KJ/Kg}$

(c) Mass flow rate of refrigerant

$$m = \frac{100 \times 10^3}{3600 (h_2 - h_3)}$$

57

$$= \frac{1000}{36(434.5 - 271.97)}$$

$= 0.171$ kg/s

Theoritical piston displacement of Compressor

$V = m \cdot v_1 = 0.171 \times 0.04185$

$= 7.156 \times 10^{-3}$ m³/s

(d) Power Consumption

$W = \dot{m}(h_2 - h_1) = 0.171 \times (434.5 - 413.6)$

$= 3.57$ kW.

theoritical H.P. of Compressor

British 746 watt

MKS 735.6 watt

$$HP = \frac{3.57 \times 10^3}{746} = 4.79 \text{ hp}$$

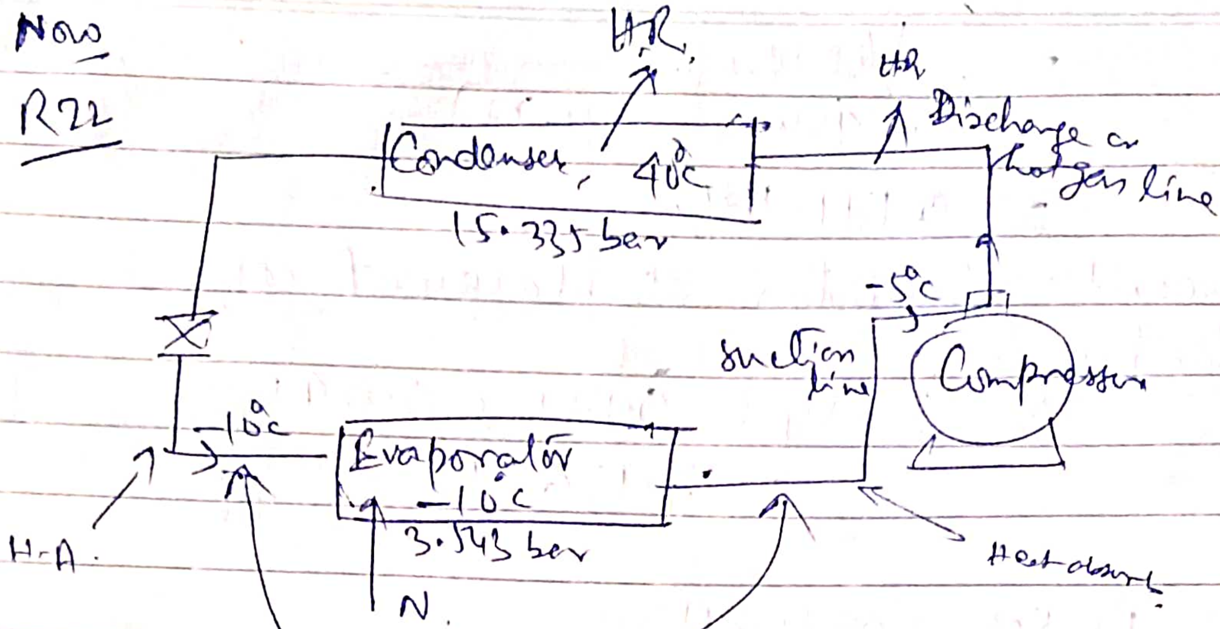
(e)

$$C.O.P. = \frac{h_2 - h_3}{h_2 - h_1} = \frac{434.5 - 271.97}{434.5 - 413.6}$$

$= 7.78$

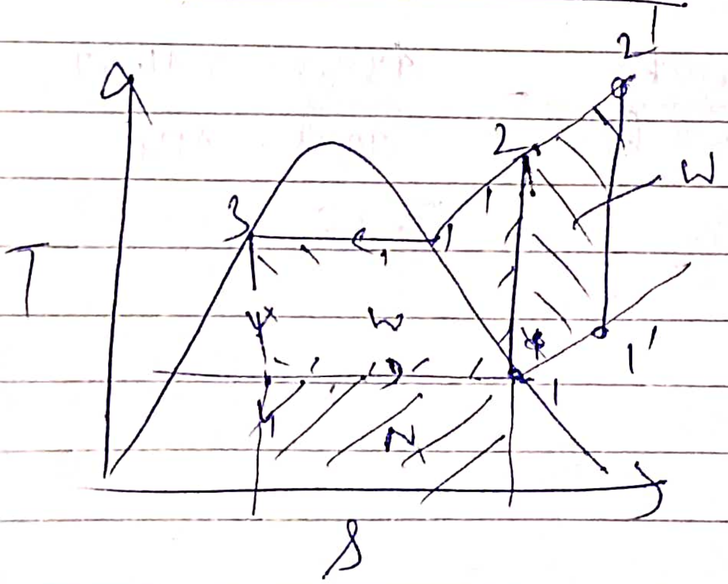
Now

R22



line is kept minimum as possible and both thermally insulated. But insulation is not 100%.

(i) Effect of Superheating at suction to the Compressor,



for 1-2-3-4

$$COP = \frac{N}{W}$$

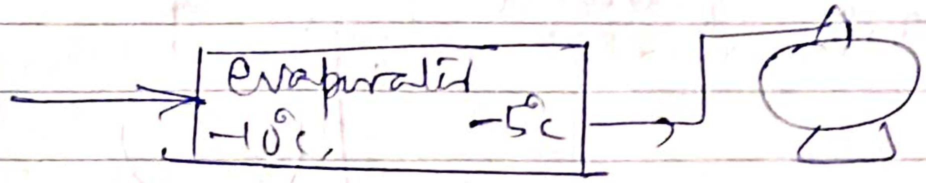
for 1-1'-2'-2-3-4

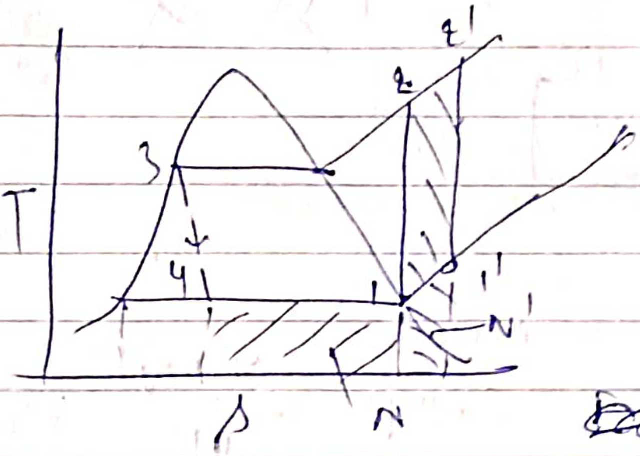
$$COP = \frac{N}{W+W'}$$

i.e. C.O.P decreases.

*** Always N is greater than W because C.O.P is always greater than 1.

(ii) Superheating within the evaporator itself :

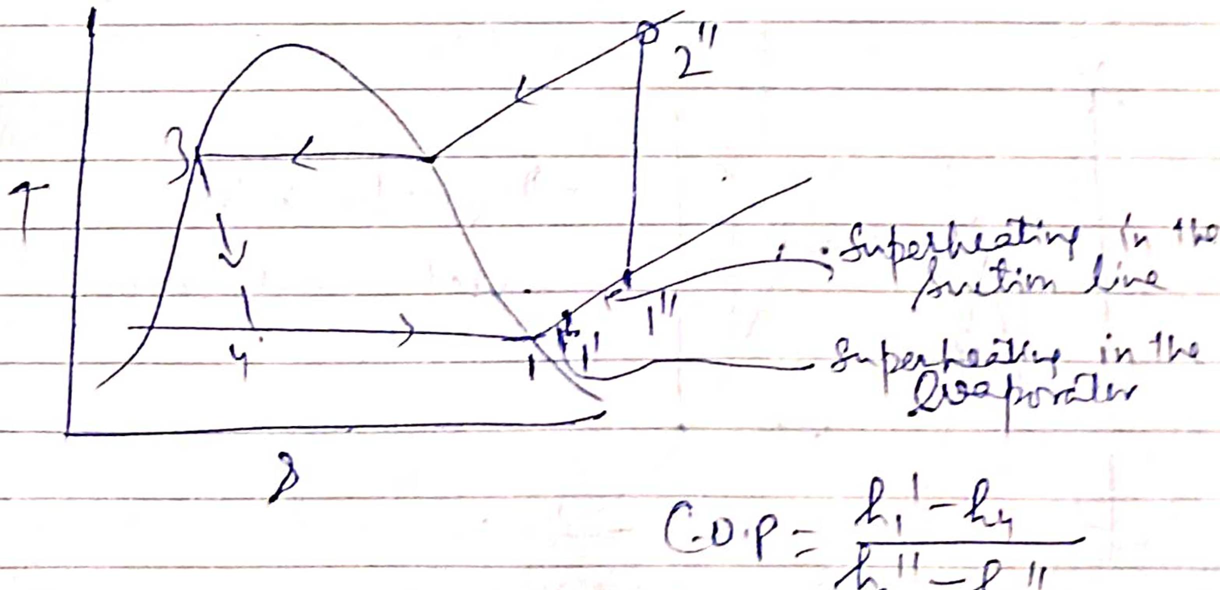




New C.O.P = $\frac{N+N'}{W+W'}$

C.O.P. ~~coefficient~~ increases.
when N' is greater than W'

(iii) Superheating in the evaporator as well as suction line :



$$C.O.P = \frac{h_1' - h_4}{h_2'' - h_1''}$$