$h_1 = 344$ KJ/kg

$h_7 = 338$ KJ/kg

$h_2 = 377$ KJ/kg

$h_3 = 373$ KJ/kg

$h_5 = h_6 = 228.5$ KJ/kg

$h_4 = 223.5$ KJ/kg

(i) Ref. Capacity $= m(h_7 - h_6)$

$= 8.958(338 - 228.5)$

$= 980.90$ KJ/s

$= 280.26$ TR

(ii) Compressor power

$\delta q - \delta w_s = dh$

${}_1q_2 - {}_1w_2 = h_2 - h_1$

$\Rightarrow Q_2 - {}_1W_2 = H_2 - H_1$

$-25 - {}_1W_2 = 8.958(377 - 344)$

${}_1W_2 = -320.61$ KJ/s

(iii) C.O.P $= \dfrac{980.90}{320.61} = 3.06$

(iv) Mass of Condensing Cooling water

Heat rejected in the Condenser

$= m(h_3 - h_4) = 8.958 \times (373 - 223.5)$

$= 1338.33$ KJ/s $= m_w \times C_{pw} \times \Delta t$

$= m_w \times 4.1868 \times 10$

$\therefore \dot{m}_w = 31.77$ kg/s

Energy entering	Energy outgoing
Compressor for 1 kg $\frac{320.61}{8.958} = 35.79$ KJ/kg	$\frac{25}{8.958} = 2.79$ KJ/kg
Discharge line —	$h_2 - h_3 = 4.0$ KJ/kg
Condenser —	$h_3 - h_4 = 149.4$ KJ/kg
Liquid line; $h_5 - h_4 = 4.9$ KJ/kg	—
Evaporator: $h_7 - h_6 = 109.5$ KJ/kg	—
Suction line: $h_1 - h_7 = 6.0$ KJ/kg	
Total $= 156.19$ KJ/kg	Total $= 156.19$ KJ/kg

P.T.O

***

Q → A refrigeration system of 10.5 tons capacity at an evaporator temperature of $-12°C$ and condenser temp. of $27°C$ is needed in a food storage locker. The refrigerant ammonia is subcooled by $6°C$ before entering the expansion valve. The vapour is 0.95 dry as it leaves the evaporator coil. The compression in the compressor is adiabatic.

Find

(i) Condition of vapour at outlet of the compressor

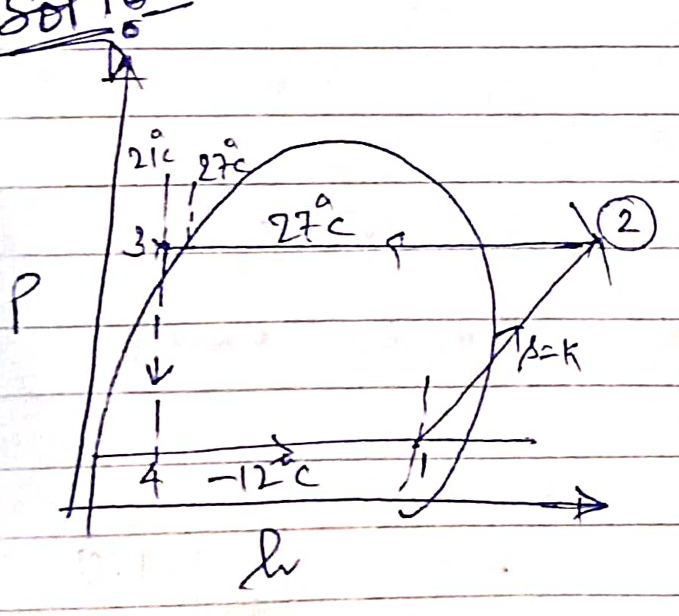
(ii) Condition of vapour at entrance to evaporator

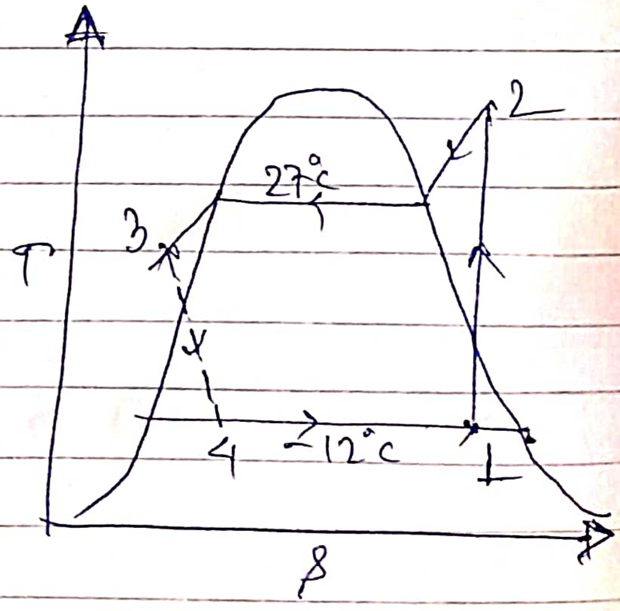
(iii) COP

(iv) Power reqd. in KW.

Neglect valve throttling and clearance effect.

Sol^n :-





$$s_1 = s_2$$

$$(s_f + x s_{fg})_1 = \left[s_g + c_p \ln \frac{T_2}{T_s}\right]_2$$

$$\Rightarrow 0.79501 + 0.95(5.7853 - 0.79501)$$

$$= 5.2953 + 2.7918 \ln \frac{T_2}{300}$$

$$\therefore T_2 = 53.98°C.$$

		50°C Superheat (c_p)
25°C	——	2.765
27°C	——	?
30°C	——	2.832
5°C		0.067
2	——	$\frac{0.067}{5} \times 2 = 0.0268$

27°C ⟶ $= 2.765 + 0.0268$
$= \cancel{0.07}\ 2.7918$

$$\begin{array}{r} 53.8 \\ -27 \\ \hline 26.8°C \end{array}$$ Superheat

	10°C	26.8°C	50°C	(Degree of Superheat)
25°C	2.994		2.765	
27°C	(?)=3.030	2.8704	2.7918	$2.765 + 2 \times \frac{(2.832 - 2.765)}{5}$
30°C	3.084		2.832	

5°C ⟶ $(3.084 - 2.994) = 0.090$

2 ⟶ $\frac{0.090}{5} \times 2 = 0.036$; (?) $= 2.994 + 0.036$

Now when diff. is 40 (in degree of superheat)

$40 \text{ — } 0.2382 \ (= 3.03 - 2.7918)$

$1^\circ \text{ — } \frac{0.2382}{40}$

$26.8 \text{ — } \frac{0.2382}{40} \times 26.8 = 0.159594$
$\simeq 0.1596$

At 3.03
-0.1596
2.8704

$\therefore 0.79501 + 0.95(5.7835 - 0.79501)$
$= 5.2963 + 2.8704 \ln \frac{T_2}{300}$

$\therefore T_2 = 53.02^\circ C$

(i) Vapours are superheated & tempr is $53.02^\circ C$.

(ii) $h_3 - h_4 = (h_f + x h_{fg})_4$

$298.527 = 144.929 + x(1447.74 - 144.929)$

$\therefore x = 0.1179$

$h_1 = (h_f + x h_{fg})_1$
$= 144.929 + 0.95(1447.74 - 144.929)$
$= 1382.60 \ kJ/kg$

$$h_2 = \left[h_g + C_p(T_2 - T_s)\right]_2$$
$$= [1484.42 + 2.8704(53.02 - 27)]$$
$$= 1559.107 \text{ KJ/kg}$$

$$N = h_1 - h_4$$
$$= 1382.60 - 298.527$$
$$= 1084.073 \text{ KJ/kg}$$

$$W = h_2 - h_1$$
$$= 1559.107 - 1382.60$$
$$= 176.507 \text{ KJ/kg}$$

$$C.O.P = \frac{N}{W} = \frac{1084.073}{176.507} = 6.1418$$

$$m(h_1 - h_4) = 10.5 \times 3.5$$

$$m = \frac{10.5 \times 3.5}{1382.60 - 298.527} = 0.0339 \text{ kg/s}$$

$$\text{Power reqd.} = m(h_2 - h_1)$$
$$= 0.0339(1559.107 - 1382.60)$$
$$= 5.983 \text{ KW}$$

(7) The effect of pressure losses etc. :

During the flow of the refrigerant through the piping, evaporator, condenser receiver and the valves, there is a pressure drop due to internal fluid friction.

It may be noted that refrigerating effect for simple saturation cycle is approximately same (slightly decreased).

But the increase in specific volume for cycle with pr. drop is appreciable.

Thus the volume of vapour handled per minute is greater for cycle with pressure drop. And added to this, is the increased range of pressure through which compressor must ~~compressor must~~ compress the refrigerant. This decreases also the volumetric efficiency. The latter results in greater power per ton.

The following assumptions were made in explaining the cycle with pressure drops:

(i) There is no superheating in the suction line.

(ii) There is no heating of the cool suction vapour as it enters the hot cylinder.

(iii) There is no sub cooling of the liquid in the sub cooler and the liquid line.

(iv) The compression process was assumed reversible adiabatic.

However, if a polytropic compression process with superheating in suction line, cylinder heating, sub cooling in liquid line and the various pressure drops are considered, the p-h diagram gets modified as shown in fig