REFRIGERATION

TONNE OF REFRIGERATION (TR)

1 ton of ice @ 0°C → 1 ton of water @ 0°C →

1 TONNE OF REFRIGERATION.

It is the refrigeration effect produced by the uniform melting of 1 ton of ice from and at 0°C in 24 hours.

1000 kg ice

Latent heat of ice - 335 KJ/kg.

For 1 ton of ice 335000 KJ. heat is required

$$1\ \text{TR} = \frac{1000 \times 335}{24 \times 60} = \underline{\underline{232.6\ \text{KJ/min}}}$$

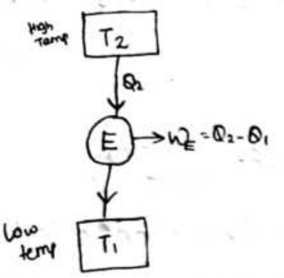
For practical application,

$$\boxed{1\ \text{TR} = 210\ \text{KJ/min}} \qquad \boxed{1\ \text{TR} = 3.5\ \text{KW}}$$

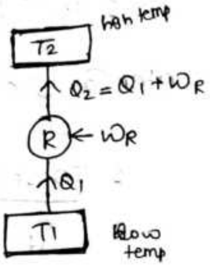
3.5 kW heat is absorbe
by AC of 1 TR

Comparison of Heat Engine, Refrigerator and Heat Pump.

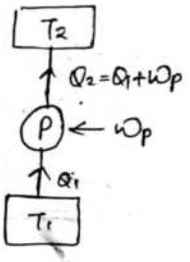
Heat Engine



REFRIGERATOR



Heat Pump.



Coefficient of Performance (COP)

COP of refrigerator, $(COP)_R = \frac{\text{Ref. effect}}{\text{Work Done}}$.

$$(COP)_R = \frac{Q_1}{Q_2 - Q_1}$$

COP of heat pump., $(COP)_{HP} = \frac{\text{Heating effect}}{\text{work done}}$

$$(COP)_{HP} = \frac{Q_2}{Q_2 - Q_1}$$

$$\cancel{1 + (COP)_{HP} = 1 + \frac{Q_2}{Q_2 - Q_1} = Q_2 - Q_1 + Q_1}$$

$$(COP)_{HP} - (COP)_R = \frac{Q_2}{Q_2 - Q_1} - \frac{Q_1}{Q_2 - Q_1} = 1$$

$$(COP)_{HP} = 1 + (COP)_R .$$

for same work more heating effect is obtained.
(since work is also added to output)

Efficiency of heat engine

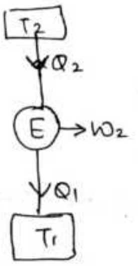
$$\eta_{HE} = \frac{\text{Work done}}{\text{heat supplied}}$$

$$= \frac{Q_2 - Q_1}{Q_2} = 1 - \frac{Q_1}{Q_2}$$

$$\boxed{(COP)_{HP} = 1/\eta_{HE}}$$

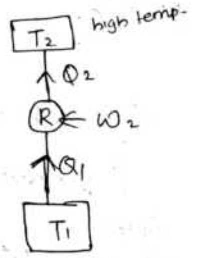
Scanned by CamScanner

Heat engine



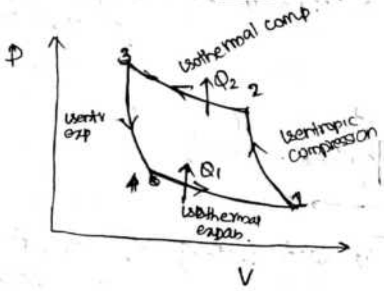
completely reversed.

Refrigerator



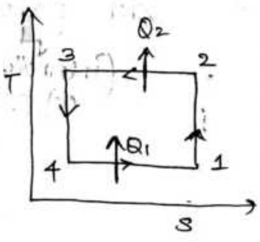
(heat engine uses Carnot cycle) → (So reversed Carnot cycle for refrigerator

Reversed Carnot Cycle.



Q2 to high temp reservoir.

(If Heat goes out, entropy decreases)



1-2 : isentropic compression.
2-3 : Isothermal compression.
3-4 : isentropic expansion.
4-1 : Isothermal expansion.

$$COP = \frac{Q_1}{Q_2 - Q_1}$$

$$Q = T(\boxed{\Delta S})$$

$$COP = \frac{Q_1}{Q_2 - Q_1} = \frac{T_1(S_1 - S_4)}{T_2(\underset{S_2 - S_3}{S_3 - S_2}) - T_1(S_1 - S_4)}$$

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

Q and W are path functions. So inexact different-al. δQ, δW.

$S_1 - S_4 = S_2 - S_3$

$$(COP)_R = \frac{T_1}{T_2 - T_1}$$

$$(COP)_{HP} = \frac{T_2}{T_2 - T_1}$$

Q. A machine working on Carnot Cycle operates between 32°C and -13°C. Determine the cop when it is operated as,

1) a refrigerating machine.
2) a heat pump. and
3) a heat engine.

T in K

Ans.

T_2 (higher temp) = 32°C = 305 K
T_1 (lower temp) = -13°C = 260 K

for ref. desired work eff. is lower temp.

$$(1)\ (COP)_R = \frac{\text{desired eff}}{\text{work}} = \frac{T_1}{T_2 - T_1}$$

$$= \frac{(-13 + 273)}{305 - 260} = \frac{260}{305 - 260} = 5.778$$

② $(COP)_{HP} = \frac{T_2}{T_2-T_1} = \frac{305}{305-260}$

$= \underline{\underline{6.778}}$ or $(COP)_{HP} = 1+(COP)_R$

③ $\eta_{HE} = \frac{T_2-T_1}{T_2} = \frac{305-260}{305}$

$= \underline{\underline{0.1475}}$

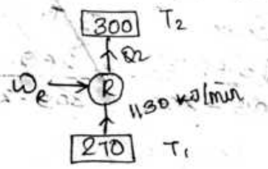
Q. A carnot refrigeration cycle absorbs heat at 270 K and rejects heat at 300 K ① Calculate the COP of this refrigeration cycle.,

② If the cycle is absorbing 1130 kJ/min at 270 K how many kJ of work per second is required.

③ If the carnot heat pump operates btw the same temperatures as the above refrigeration cycle what is the coefficient of performance.

④ How many kJ/min will the heat pump deliver if it absorbes 1130 kJ/min at 270 K.

Ans. ① Given



① $(COP)_R = \frac{T_1}{T_2-T_1} = \frac{270}{300-270} = \underline{\underline{9}}$

② $(COP)_R = \frac{\text{Ref. effect}}{\text{Work Done}}$

$9 = \frac{Q_1}{W_R}$

$Q_1 = 1130$ kJ/min $= 18.83$ kJ/sec.

$9 = \frac{18.83}{WD}$

$\Rightarrow$ Work Done $= \underline{\underline{2.09 \text{ kJ/s}}}$

③ $(COP)_{HP} = \frac{T_2}{T_2-T_1} = \frac{300}{300-270} = \underline{\underline{10}}$

④ $(COP)_{HP} = \frac{Q_2}{Q_2-Q_1}$

$\cancel{\frac{Q_2}{Q_2-}}\ \frac{Q_2}{Q_2-1130} = 10 \Rightarrow Q_2 = 10Q_2 - 11300$

$9Q_2 = 11300 \Rightarrow Q_2 = \underline{\underline{1255.5 \text{ kJ/min}}}$

or

$(COP)_{HP} = \frac{Q_2}{W_R}$ (Work Done)$_{HP}$ = $(WD)_R$.
(Same carnot cycl

$Q_2 = Q_1 + W_R$.

Q. A cold storage plant is required to store 20 tons of fish. The fish is supplied at a temp of 30°C. The specific heat of fish above freezing point is 2.93 $\frac{kJ}{kgK}$. The specific heat of fish below freezing point is 1.26 kJ/kg K. The fish is stored in a cold storage which is maintained at -8°C. The freezing point of fish is -4°C. The latent heat of fish is 235 kJ/kg. If the plant requires 75 kW to drive it. Find

(1) the capacity of plant (find TR)

(2) time taken to achieve cooling.

Assume actual COP of the plant as 0.3 of the carnot COP.

$$(COP)_R = \frac{T_1}{T_2 - T_1}$$

$$T_1 = -8 + 273 = 265\,K$$

$$T_2 = 30 + 273 = 303\,K.$$

$$(COP)_R = \frac{265}{303-265} = \underline{\underline{6.97}}. \quad \cancel{6.97}$$

$$\text{Actual COP} = 0.3 \times 6.97 = \underline{\underline{209}}.$$

$$(COP)_{act} = \frac{\text{Refr. effect}}{\text{work done}} \Rightarrow 2.092 = \frac{\text{Ref. effect}}{75}$$

$\therefore$ Ref. effect $= 156.75$ kW

$$1\,TR = 3.5\,kW \Rightarrow \text{capacity of plant} = \frac{\text{Ref. effect}}{3.5}$$

$$= \underline{\underline{44.78}}\ TR.$$

$$\left.\begin{matrix} 30°C \\ \downarrow \\ -4°C \end{matrix}\right] \text{ sensible cooling } Q_1$$

$Q_1 = mC_p\Delta T$ (Heat removed to cool the fish from 30°C to -4°C)

$$= 20\times10^3 \times 2.93\times10^3 \times (30+4)$$

$$= 1.9\times10^9\ J = \underline{\underline{1992400\ kJ}}.$$

$$\text{Latent heat received} = M \times L_H = 20\times10^3 \times 235$$

$$= \underline{\underline{4.7\times10^6\ kJ}}.$$

Heat removed to cool from -4°C to -8°C

$$= mC_p\Delta T = 20\times10^3 \times 1.26\,(-4+8)$$

$$= \underline{\underline{100800\ kJ}}$$

$$\text{Total heat} = 1992400 + 4700000 + 100800$$

$$= \underline{\underline{6793200\ kJ}}$$

$$\text{Ref. effect} = 156.75\ kW = 156.75\ kJ/s.$$

$\therefore$ in 1 s → 156.75 kJ/s.

$\therefore$ Time taken ~~for~~ to achieve cooling

$$= \frac{6793200}{156.75}$$

$$= 43337.8 \text{ s}$$

$$= 12.04 \text{ hr} = 12 \text{ hr } 2 \text{ min } 17.8 \text{ sec}$$

Q. A cold storage is to be maintained at -5°C while the surrounding are at 35°C. Heat leakage from the surrounding into the cold storage is estimated to be 29 kW. The actual COP of the refrigeration plant is one-third of an ideal plant working between the same temperatures. Find the power required to drive the plant.

$T_1 = -5°C = 268 K$

$T_2 = 35 = 308 K$.

$$COP = \frac{T_1}{T_2 - T_1} = \frac{268}{308 - 268} = \underline{\underline{6.7}}$$

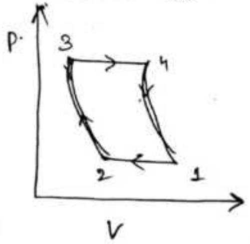
$$\text{Actual COP} = \frac{1}{3} \times 6.7 = 2.233$$

$$2.233 = \frac{\text{Ref. effect}}{\text{work done}} = \frac{29 \text{ kW}}{\text{W.D}}$$

$$W = \frac{29}{2.233} = \underline{\underline{12.98 \text{ kW}}}$$

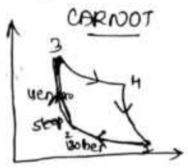
BELL-COLEMAN CYCLE (REVERSED BRAYTON OR JOULE CYCLE)

BRAYTON (gas turbine)



Isothermal processes are not possible. They are very slow. ← So they are replaced by isobaric processes

CARNOT



PV=C - isothermal

$\frac{dp}{dv} = 1$

PV^r = Const → isentr

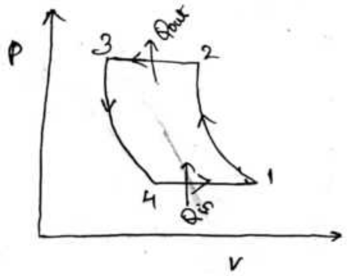
$\frac{dp}{dv} > 1$

Slope is more for isentropic

Isobaric heat addition & heat rejection.

Brayton cycle → Not reversible cycle. as isobaric processes are not reversible

Reversed brayton cycle or Bell Coleman Cycle.



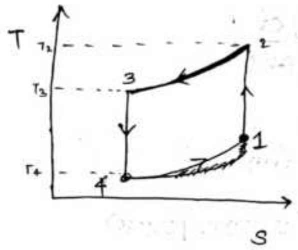
1-2: isentropic compr.

2-3 ~~isentropic~~ isobaric heat rejection

3-4 - isentropic exp

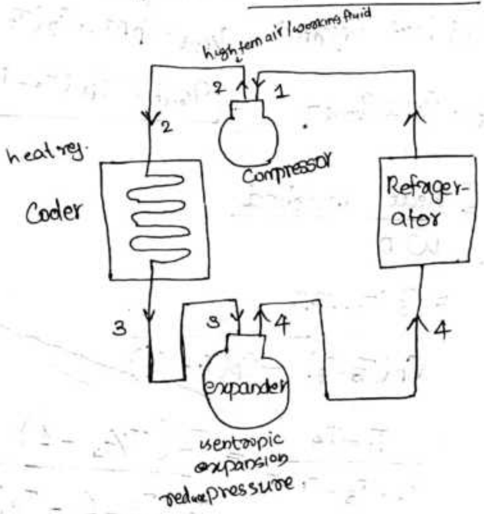
4-1 - isobaric heat addition

T-S diagram



heat rej., temp reduce

Air - refrigeration system working on Bell-Colman Cycle



Use of compressor: To increase temp of working fluid, so as to facilitate ~~heat to~~ reject heat to surroundings.

COP of BELL-COLEMAN AIR REFRIGERATION.

$$COP = \frac{\text{Ref. effect}}{\text{work done}}$$

$$= \frac{Q_{in}}{Q_{out} - Q_{in}}$$

Amount of heat absorbed, Q_{in} in 1-4 (isobaric process)

$$Q_{in} = C_p \Delta T = C_p \times (T_1 - T_4)$$

Amount of heat rejected, $q_{out} = C_p \Delta T = C_p(T_2 - T_3)$

~~$q_{net} = q_{in} + q_{out} =$~~

$$\boxed{q_{out} = C_p(T_3 - T_2) \text{ (-ve)}}$$

$$COP = \frac{\text{Heat absorbed}}{WD}$$

$$= \frac{C_p(T_1 - T_4)}{C_p(T_2 - T_3) - C_p(T_1 - T_4)}$$

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

1-2 is isentropic

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

3-4 is also isentropic

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$P_2 = P_3$, $P_4 = P_1$

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \boxed{\frac{T_2}{T_1} = \frac{T_3}{T_4}}$$

$$T_1/T_4 = T_2/T_3$$

$$COP = \frac{T_4 \cancel{(T_1/T_4 - 1)}}{T_3 \cancel{(T_1/T_4 - 1)} - T_4 \cancel{(T_1/T_4 - 1)}}$$ ~~$= (T_1 - 1)$~~

3-4 is ents

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

~~(T_1/T_4)~~ $=$

$$= \frac{T_4}{T_3 - T_4}$$ //

$$COP = \frac{T_4}{T_4\left(\frac{T_3}{T_4} - 1\right)}$$

$$= \frac{1}{\left(T_3/T_4 - 1\right)} = \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

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

γ_p = Pressure ratio or compression ratio $= \frac{P_2}{P_1}$

$$\boxed{COP = \frac{1}{\gamma_p^{\frac{\gamma-1}{\gamma}} - 1}}$$

(~~ref~~ Bell-Coleman air refrigeration)

* If the process is Polytropic the equation of COP can be derived as, (not ideal, any actual)

$$COP = \frac{\text{Ref. effect}}{\text{W.D}}$$

* Instead of isentropic exp. & compre Polytropic exp. & compression

ref. page 51

Network $= W_c - W_E$

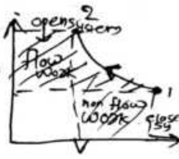
$$W_c = \frac{P_1V_1 - P_2V_2}{n-1}$$ } non-flow work or closed system.

Compressor → There is mass transfer (air come in & exit) → so, open system

So, W_c is not equal to $\frac{P_1V_1 - P_2V_2}{n-1}$

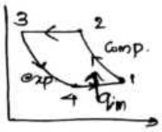
(Pdv is non flow work)

Flow work is Vdp



$$\therefore W_c = \frac{n}{n-1}(P_2V_2 - P_1V_1)$$ ⟶ Open system

Work of expansion.



$$W_E = \frac{n}{n-1}(P_3V_3 - P_4V_4)$$

Net work $= W_c - W_E$ work

$$= \frac{n}{n-1}(P_2V_2 - P_1V_1) - \frac{n}{n-1}(P_3V_3 - P_4V_4)$$

$$= \frac{n}{n-1}(RT_2 - RT_1) - \frac{n}{n-1}(RT_3 - RT_4)$$

$$= \frac{n}{n-1} R\left[(T_2 - T_1) - (T_3 - T_4)\right]$$

Refrigeration effect of (4-1)

Heat absorbed $= C_p(T_1 - T_4)$

$$\boxed{R = C_v(\gamma - 1)}$$

$$\text{Net work} = \frac{n}{n-1} C_v(r-1)\left[(T_2-T_1)-(T_3-T_4)\right]$$

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

$C_p - C_v = R$

$$COP = \frac{Q_{in}}{W.D} = \frac{C_p(T_1-T_4)}{\frac{n}{n-1} C_v(r-1)\left[(T_2-T_1)-(T_3-T_4)\right]}$$

$$= \frac{n-1}{n} r \frac{(T_1-T_4)}{(r-1)((T_2-T_1)-(T_3-T_4))}$$

$$= \frac{T_1-T_4}{\frac{n}{n-1} \times \frac{r-1}{r}((T_2-T_1)-(T_3-T_4))}$$

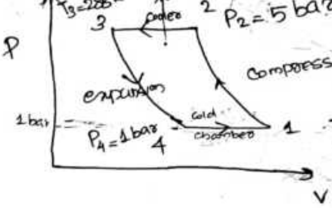
Q. Atmospheric air at 1 bar and -5°C is drawn into the cylinder of the compressor of a bell-Coleman refrigerating machine. It is compressed isentropically to a pressure of 5 bar. In the cooler, the compressed air is cooled to 15°C, pressure remaining the same. It is then expanded to a pressure of 1 bar in an expansion cylinder, from where it is passed to the cold chamber. find

1) Workdone per kg of air.

2) COP of the plant.

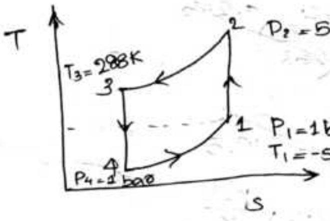
For air assume law for expansion $PV^{1.2} = C$ and for compression $PV^{1.4} = C$ and specific heat of air at constant pressure = 1 kJ/kgK.

ans.



Low press - evaporator

high press → Condenser



$\frac{P_2}{P_1} = \frac{T_2}{}$

a) For expansion, $PV^{1.2} = C$. $P \alpha T_3$ $R T_4$

$$W_E = \frac{n}{n-1}(P_3V_3 - P_4V_4)$$

$$= \frac{1.2}{1.2-1}(5 \times V_3 - 1 \times V_4)$$

$$= \frac{n}{n-1} R(T_3 - T_4)$$

for compression, $PV^{1.4} = C$

$$W_c = \frac{n}{n-1}(P_2V_2 - P_1V_1)$$

$$= \frac{n}{n-1}(RT_2 - RT_1)$$

$$= \frac{n}{n-1}R(T_2 - T_1)$$

R of air = 287

R = 287 $\frac{J}{kg\,K}$

~~PV~~

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

~~$\frac{5}{1} = \left(\frac{T_2}{268}\right)^{\frac{1.4-1}{1.4}}$~~

~~$5 = \left(\frac{T_2}{268}\right)^{0.285}$~~

~~$\frac{T_2}{268} = 279.5$~~

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

$$\frac{T_2}{268} = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 424.46 \text{ K}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{1.2-1}{1.2}}$$

$$\frac{288}{T_4} = 5^{0.166} \Rightarrow T_4 = 220 \text{ k}.$$

$$W_c = \frac{1.4}{1.4-1} \times 287 \times (424.46 - 268)$$

$$= 158 \text{ kJ/kg}.$$

$$W_E = \frac{1.2}{1.2-1} \times 287\,(288 - 220)$$

$$= 117096 = 117 \text{ kJ/kg}$$

Net work done = $W_c - W_E$

$$= 158 - 117$$

$$= 41 \text{ kJ/kg}.$$

b) ~~Eop~~ = $\frac{T_1 - T_4}{\frac{n}{n-1}\cancel{\frac{\gamma-1}{\gamma}}\left((T_2 - T_1) - (T_3 - T_4)\right)}$

b) $COP = \frac{C_p(T_1 - T_4)}{W.D}$

$= \frac{1 \times 10^3 \times (268 - 220)}{41 \times 10^3}$

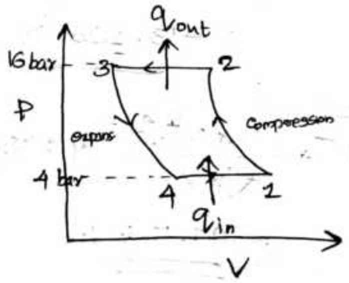
$= \underline{\underline{1.17}}$

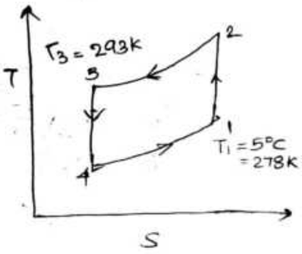
Q. A ~~dense~~ ~~cross~~ closed cycle refrigeration system working b/w 4 bar and 16 bar extracts 126 MJ of heat /hour. The air enters the compressor at 5°C and into the expanders at 20°C. Assuming the uni runs at 300 rpm, findout,

1) Power required to run the unit.
2) ~~Power~~ bore of the compressor
3) refrigerating capacity in tons of ice at 0°C per day.

Take the following, the compressor & expander are double acting & stroke of compressor & expander is 300 mm. The mech. efficiency of compressor is 80%. η_{mech} of expander is 85%. Assume the compression & expansion are isentropic

Ans. Ref. cycle is bell-Coleman Cycle





Pressure at condenser & evaporator

$P_1 = P_4 = 4$ bar

$P_2 = P_3 = 16$ bar

$T_1 = 278$ K

$T_3 = 293$ K.

$Q_E = Q_{in} = 126$ MJ/h $= \frac{126 \times 10^3}{3600}$ kJ/s

$= \underline{35 \text{ kJ/s}}$

$\eta_c = 80\%$.

$\eta_e = 85\%$.

$N = 300$ rpm

For process 1-2

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}} \Rightarrow T_2 = \left(\frac{16}{4}\right)^{\frac{1.4-1}{1.4}} \times 278$$

$$= \underline{413.106 \text{ K}}$$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{r-1}{r}} \Rightarrow T_4 = \left(\frac{4}{16}\right)^{\frac{1.4-1}{1.4}} \times 293$$

$$= \underline{197.174 \text{ K}}$$

$q_{in} = C_p(T_1 - T_4)$ $\quad C_p = 1.005$ kJ/kg K

$= 1.005(278 - 197.174)$

$= \underline{81.23 \text{ kJ/kg}}$.

$\frac{Q_E}{q_{in}} = \frac{126 \times 10^3}{81.23} = 1551.15$ kg/hour

$= 0.4308$ kg/s

$= \underline{4.3 \times 10^{-1} \text{ kg/s}}$

$= \underline{430.8 \times 10^{-3} \text{ kg/s}}$

Work done in compressor,

(isentropic work done) $W_C = \frac{n}{n-1}(P_2V_2 - P_1V_1) = \frac{n}{n-1} R(T_2 - T_1)$

for 100% η

$\rightarrow W_C = \frac{r}{r-1} R(T_2 - T_1)$

$R = 287$ J/kgK

$$= \frac{1.4}{1.4 - 1} 287 \times (413.106 - 278)$$

$$= \underline{135713.977} = \underline{135.7 \text{ kJ/kg}}$$

~~Work done in expander,~~

~~$W_E = \frac{r}{r-1} R(T$~~

$\eta_{com} = 80\%$. So more work has to be done.

Actual work of compressor $= \frac{W_c}{0.8}$ (work consuming) So

$= \frac{135.7}{0.8}$

$= 169.625$ kJ/kg.

$W_e = \frac{r}{r-1} R (T_3 - T_4)$ (isentropic work done by expander)

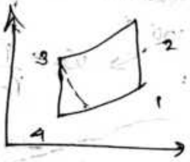
$= \frac{1.4}{1.4-1} \times 287 (293 - 197.174)$

$= 9625$ kJ/kg

Actual work $W_e = 0.85 \times W_e$ (If comp. final temp is high) (In expansion if friction is present work will reduce final temp will decrease)

$= 0.85 \times 96.25$

$= 81.81$ kJ/kg



$W_{net} = W_c - W_e$

$= 169.625 - 81.81 = 87.815$ kJ/kg

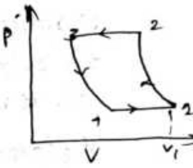
Power = Net work × mass flow rate

$= 87.815 \times 430.8 \times 10^{-3}$

$= 37.76$ kJ/s $= 37.76$ kW

b) bore dia — cylinder diameter.

$N = 300$ rpm $\quad l = 300$ mm



$P_1 V_1 = mRT_1$

$V_1 = \frac{mRT_1}{P_1} = \frac{0.4308 \times 287 \times 278}{4 \times 10^5}$ V_1: Vol. of compressor

P = bar → 10^5 Pa

$= 8592.95$ m³/s $\times 10^{-5} = 0.0859 \frac{m^3}{s}$

$V = \pi/4 \, D^2 \, L \times \underset{\text{double acting}}{2} \times \frac{300}{60}$ → rpm

$0.0859 = \frac{\pi}{4} D^2 \times 0.3 \times 2 \times \frac{300}{60}$

$D = 0.19$ m

c) Refrigerating capacity in tonnes of ice at 0°C per day. (weight of ice)

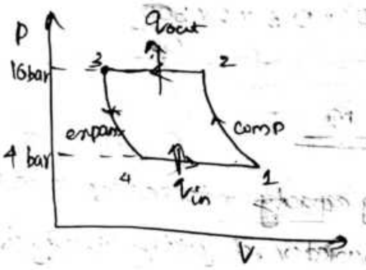
We know that heat extracted or refrigerating capacity of the system per day $= 126 \times 24 = 3024$ MJ $= 3024 \times 10^3$ kJ

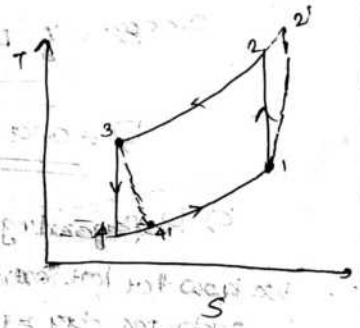
Since the latent heat of ice is 335 kJ/kg, therefore ice formation capacity of the system per day

$= 3024 \times 10^3 / 335 = 9000$ kg $= 9$ tonnes dense air refrigeration

Q. * A system operates btw pressures of 4 bar & 16 bar. The air temperature after heat rejection to the surrounding is 37°C. and air temp at the exit of refrigerator is 7°C. The isentropic efficiencies of turbine and compressor are 0.85 and 0.82 resp. Determine compressor & turbine work per ton of refrigeration, COP and power per ton of refrigeration. Take $r = 1.4$ & $C_p = 1.005$ kJ/kgK.

Here difference is that the processes (comp & expan) are not isentropic.





$T_3 = 37° = \underline{\underline{310\ K}}.$

$T_1 = 7°C = \underline{\underline{280\ K}}.$

$\eta_{isentropic\ comp} = 0.82$

$\eta_{isen.\ turbine} = 0.85$

$r = 1.4 \qquad C_p = 1.005$ kJ/kgK.

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

$$T_2 = 280 \times \left(16/4\right)^{\frac{1.4-1}{1.4}} = 416\ K$$

$$\cancel{\frac{T_4}{T_1} = \left(\frac{P_4}{P_1}\right)^{\frac{r-1}{r}}}$$

$$T_4 \cancel{= 280 \times 4^{\frac{1.4}{1}}}$$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{1.4-1}{1.4}}$$

1 - 2 &
3 - 4 are
~~isentropic~~
polytropic

$$T_4 = 310 \times \left(4/16\right)^{\frac{0.4}{1.4}}$$

$$= \underline{\underline{208.614\ K}}$$

$\eta_c = 0.8$ $\qquad$ $\eta_c =$ isentropic

$$\eta_c = \frac{\text{isentropic temp difference}}{\text{Actual temp difference}}$$

$$W_c = \frac{r}{r-1} R(T_2 - T_1) = C_p(T_2 - T_1)$$

$$R = C_p\left(\frac{r-1}{r}\right)$$

$$0.8 = \frac{T_2 - T_1}{T_2' - T_1} = \frac{416 - 280}{T_2' - 280} = \frac{136}{T_2' - 280}$$

$$T_2' - 280 = \frac{136}{0.8} = 170$$

$$T_2' = \underline{\underline{450\ \text{K}}}$$

$$\eta_E = \frac{\text{Actual temp diff.}}{\text{Isentropic temp difference}}$$

$$= \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.85 = \frac{310 - \cancel{208.614}\ T_4'}{310 - \cancel{T_4}\ 208.614}$$

$$T_4' = \underline{\underline{\cancel{190.72}}}\ \underline{\underline{223.82}}$$

$$\text{Turbine work} = C_p(T_3 - T_4')$$

$$= 1.005 \times 10^3\ (310 - 223.82)$$

$$= \underline{\underline{86.61 \times 10^3\ \text{J/kg}}}$$

$$\text{Heat absorbed/kg of air} = C_p(T_1 - T_4')$$

$$1\ \text{TR} = 3.5\ \text{kW} \qquad \frac{\text{Heat}}{3.5\ \text{kW}} = \ \text{kg/s}.$$

$$\text{Heat absorbed/kg of air} = 1.005 \times 10^3 \times (280 - 223.82)$$

$$= 56460.9\ \text{J/kg} \qquad \frac{\text{J/kg}}{\text{J/s}}$$

Mass ~~of air~~ (per TR) ~~= 56460.9 / 3.5×10^3 = 16.131 kg/s.~~

∴ ~~Turbine work per ton of refrigeration~~ =

$$\dot{m} = \frac{3.5 \times 10^3}{C_p(T_1 - T_4')} \quad \begin{matrix}(1\ \text{TR}) \\ (\text{Heat by 1 kg of air})\end{matrix} \qquad 1\ \text{TR}$$

$$= \frac{3.5 \times 10^3}{56460.9} = 6.19 \times 10^{-2}\ \text{kg/s} \quad \text{(kg of air requ.) for 1 TR}$$

$$\frac{W_T}{\text{TR}} = 86.61 \times 10^3 \times 6.19 \times 10^{-2}$$

$$= \underline{\underline{5361.159}}\ \text{J/s (W)} = \underline{\underline{5361.159\ \text{W}}}$$

$$\text{Compressor work} = C_p(T_2' - T_1)$$

$$= 1.005 \times 10^3 \times (450 - 280)$$

$$= 170850.$$

~~Actual co~~ Work / Ton of refrigeration $= 170850 \times 6.19 \times 10^{-2}$

$$= 10575.615\ \cancel{\text{W/TR}}\ \text{J/s (W)}$$

$$COP = \frac{\text{Ref. effect}}{W.D} = \frac{\text{Ref. effect}}{W_C - W_E}$$

$$= \frac{C_p(T_1 - T_4')}{C_p(T_2' - T_1) - C_p(T_3 - T_4')}$$

$$= \frac{1.005 \times (280 - 223.82)}{1.005 \times (450 - 280) - 1.005 \times (310 - 223.82)}$$

$$= \cancel{0.33}$$

$$= \frac{56.18}{83.82}$$

$$= \underline{\underline{0.6702}}$$

power = net work × mass flow rate

Power per ton of refrigeration
= net work × $\dot{m}$

Net work = 10575.615 − 5361.159 = $\underline{\underline{5214.456\,W}}$

~~Power per ton of ref = 5214.456 × 6.19 × 10^{-2}~~

~~= 322.77 ~~kg~~W/s~~

∴ Power/TR = net work = 5.214 kW

Assig. Q. With a neat sketch explain simple boot-strap system

Q. explain reduced ~~a~~ ambient system and regenerative systems with neat sketch.

Q. explain vortex tube refrigeration system with a neat sketch.

Q. Write a note on very low temp. refrigeration system.
→ (14 th)(4 paper)

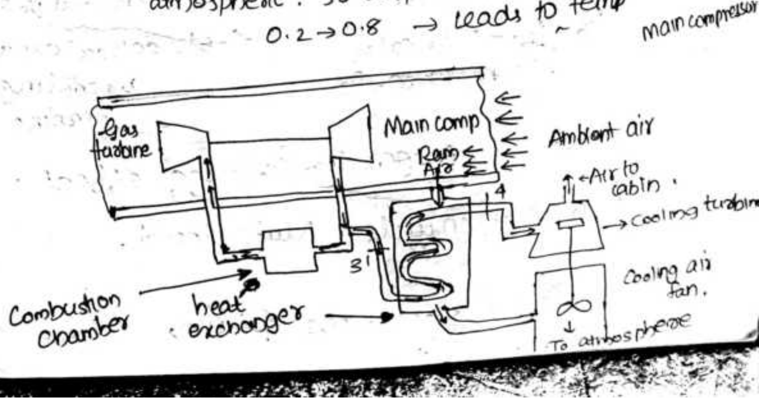
AIRCRAFT REFRIGERATION SYSTEM.

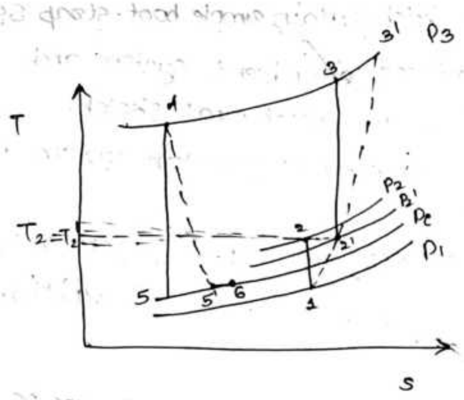
~~100 kg~~ 10 km

Lapse rate − 6.49 kelvin/km

Flight height − @ 10 km

Ram effect : pressure inside is very less eg. 0.2 bar. for comfort of occupants (exit) ~~when~~ P must be atmospheric. So compress. 0.2 → 0.8 → leads to increase in temp





P_1: ambient pressure

P_2: ideal ramming pressure

P_2': actual ramming pressure

$T_2 = T_2'$ (isenthalpic)(so temp is same)

P_3 : main compressor pressure.

$P_{5,5'}$ - Cabin pressure

$P_6 = P_5 = P_{5'} = $ Cabin pressure $= P_c$

1-2 - ramming (ideal isentropic)

1-2' - actual process

2'-3 - ideal compression (isentropic)

2'-3' - actual compression.

3'-4 → heat exchanger at const pressure

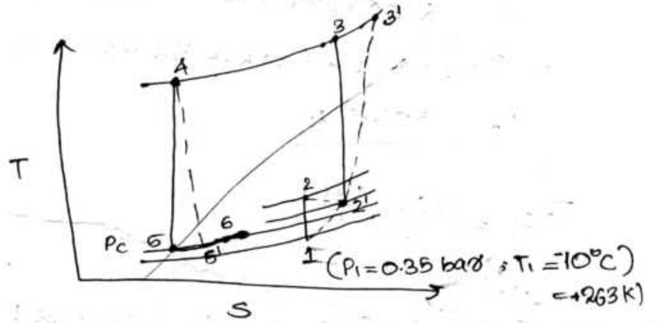
4-5' - actual expansion in cooling turbine

5'-6 - refrigeration / cooling effect.

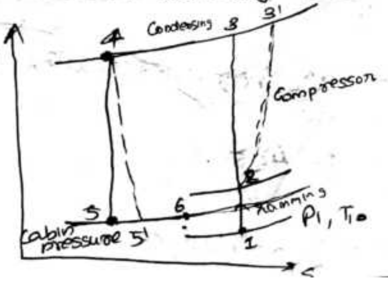
It is an open cycle. Not closed.

5'-6 → heat absorption.

Q. An aircraft moving with a speed of 1000 km/hr uses simple gas refrigeration cycle for air conditioning. The ambient pressure and temperature are 0.35 bar & -10°C respectively. The pressure ratio of the compressor is 4.5. The heat exchanger effectiveness is 0.95. The isentropic efficiencies of compressor & expander are 0.8 each. The cabin pressure & temperature are 1.06 bar and 25°C. Determine temperatures & pressures at all points of the cycle. Also find the volume flow rate through the compressor inlet & expander outlet for 100 TR.



Not said about ramming η. so taken as isentropic



expansion -4-
in cooling

5 - cabin pressure

5'-6' - heat absorb

$V = 1000\ km/hr = \frac{1000 \times 10^3}{3600} = \underline{\underline{277.77}}\ m/s$

Enthalpy change = change in kinetic energy.

$h_2 - h_1 = \Delta KE$

$V_2 = 0$

$h_2 - h_1 = \frac{V_1^2}{2000} - 0$

$C_p T_2 - C_p T_1 = \frac{V_1^2}{2000} \Rightarrow T_2 - T_1 = \frac{V_1}{2000\, C_p}$

$T_2 - T_1 = \frac{277.77^2}{2000 \times 1.005} = \cancel{0.138}\ 38.386.$

$\frac{V_1^2}{2000}$ to get value in KJ since C_p is in KJ 1.005

$T_2 = T_1 + 38.36 = 263 K + 38.36$

$T_2 = \underline{\underline{301.4\ k}}$

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

$\left(\frac{301.4}{263}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{P_2}{0.35 \times 10^5}\right) \Rightarrow P_2 = 56392.8\ Pa$

$= \underline{\underline{0.56\ bar}}$

Pressure ratio of compressor,

Given $\frac{P_3'}{P_2} = 4.5$

$P_3' = 2.53\ bar$ //

$P_3' = P_3$

$\eta_c = \frac{T_3 - T_2}{T_3' - T_2}$

~~0.8 =~~

ventro. 2-3 so)

$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{r-1}{r}}$

$T_3 = 301.4 \times \left(\frac{2.53}{0.56}\right)^{\frac{1.4-1}{1.4}}$

$= \underline{\underline{463.73\ k}}$

$0.8 = \frac{463.73 - 301.4}{T_3' - 301.4}$

$T_3' = \underline{\underline{504.3\ K}}$

3'-4 ~~expan~~ heat exchanger.

Heat exchanger effectiveness = 0.95

$= \frac{T_3' - T_4}{T_3' - T_2}$ (actual temp diff / (req. temp diff

$0.95 = \frac{504.3 - T_4}{504.3 - 301.4} \Rightarrow T_4 = \underline{\underline{311.545\ k}}$

4-5 expansion.

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{r-1}{r}}$$

Cabin temp — temp at 6

$(P_3' = P_4$ same line$)$

temp 5 — is temp of refrigerant / air

$P_4 = 2.53$ bar

$P_5 = 1.06$ bar.

5' is actual temp of air supplied to cabin.
∴ cabin temp is 6.

$$\frac{T_5}{311.545} = \left(\frac{1.06}{2.53}\right)^{\frac{1.4-1}{1.4}}$$

$\Rightarrow T_5 = 242.98$ k.

$\eta_E = 0.8$

$$\eta_E = \cancel{\frac{T_{5'}-T_4}{T_5-T_4}} \quad \frac{T_4 - T_5'}{T_4 - T_5} \quad \frac{\text{actual}}{\text{isentropic}}$$

$$0.8 = \frac{311.545 - T_5'}{311.545 - 242.98}$$

$$\underline{\underline{T_{5'} = 256.693 \text{ k}}}$$

$T_6 = 25°c = \underline{\underline{298 \text{ k}}}$

~~100 TO~~

Mass flow rates same.
Volume flow rate needn't to be same.

100 TR. = 350 kW.

5'-6 — refrigeration process

For 1 kg of air, (we get ref. effect)

$$C_p(T_6 - T_5') = 1.005 \times 10^3 \times (298 - 256.693)$$
$$= 41513.535 \text{ kJ/kg}.$$

$W = $ kJ/s.
$C_p = \frac{\text{kJ}}{\text{kgk}}$
$C_p \Delta T = \frac{\text{kJ}}{\text{kg}}$.

$$\therefore \dot{m}_a = \frac{350 \times 10^3}{41513.535} = \underline{\underline{8.43 \text{ kg/s}}}$$

Volume flow rate,

P_2 ∵ Compressor pressure (inlet of compressor

$P_2 V_2 = mRT_2$ $\qquad R = 287$ J/kgk

$0.56 \times 10^5 \times V_2 = 8.43 \times 287 \times 301.4$

Volume flow rate, (at compr. inlet) $V_2 = \underline{\underline{13.02 \text{ m}^3/\text{s}}}$

Expander outlet $P_5' = 1.06$ bar

$P_5' V_5' = mRT_5'$

$1.06 \times 10^5 \times V_5' = 8.43 \times 287 \times 256.693$

$\therefore V_5' = \underline{5.85 \ m^3/s}$

Volume flow rate at expander outlet $V_5' = 5.85 \ m^3/s$

(Simple bootstrap aircraft study)

Module 2

VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

Refrigerants – R-134

earlier CFC is used. (global warming)

Higher COP can be obtained

In air only sensible cooling, no phase change.

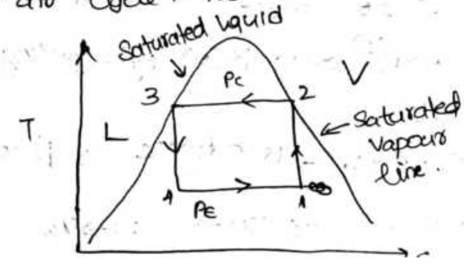
In refrigerant without changing refrigerant temp. room temp can be changed.

1. Compressor
2. Condenser
3. expansion device
4. evaporator

Vapour Cycle

Phase change occurs.

In air cycle - no phase change.

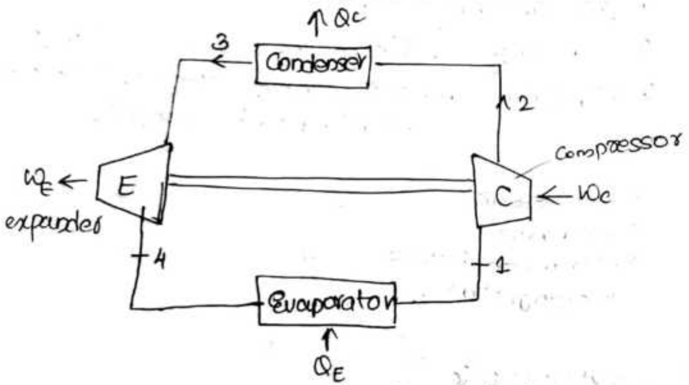


[As pressure change, boiling point change (Iso line)]

⑧ Saturated vapour & liquid state at boiling point.

Dryness fraction$_{SVL}$ = 1

Dryness fraction$_{SLL}$ = 0



Carnot cycle

Condenser ~~Constant~~ : heat rejected at const pressure.

vapour(2) → liquid (3) by rejecting latent heat.

3-4 – expansion – isentropic → pressure ↓

4-1 → Heat absorption → $x_1 \rightarrow x_2$.

Why carnot Cycle is not possible practically ?

1-2 compressor.

At 1, liquid + gaseous (vapour) state. Compressor (reciprocating compressor) may gets damaged due to liquid presence.

3-4

Work done by turbine $w_E = \int_{P_c}^{P_e} v\,dp$

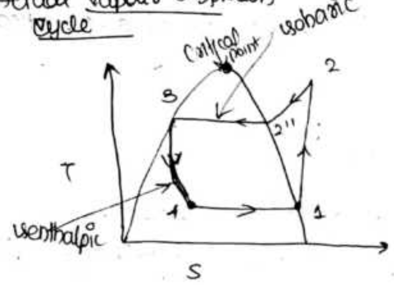
V at point 3 → specific volume of liquid → it is very low.

So only small work is obtained.

So carnot cycle can't be used in refrigeration system.

So using expander is not useful. ~~As~~ (Also expander is not efficient.) So using expander is not efficient.

Actual Vapour Compression Cycle



EVANS – PERKINS CYCLE

(Reverse Rankine Cycle)

2 is superheated vapour (heated above saturation temp)

3 - liquid

2-3 → at const pressure

Isothermal process in carnot cycle (2-3) is replaced by isobaric process.

Expander is replaced by ~~[illegible]~~ expansion valve & capillary tube. Isenthalpic process is used

Isotherm and isobar are same inside the curve.

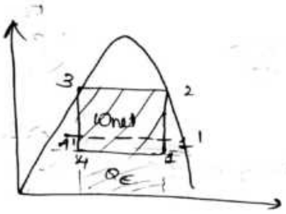
Condensation (no ~~temp~~ change) (phase change, inside curve. heat release),

At critical point vapour to liquid without giving any heat.

Condensation is isobaric (not isothermal, isothermal inside the curve.

For reverse carnot Cycle

$$COP = \frac{T_E}{T_C - T_E}$$



$W_{net} = \underset{(2,3)}{Q_C} - \underset{\substack{\text{(evaporato} \\ 4-1}}{Q_E}$

If evaporator temperature increase $\overset{(4'-1')}{}$ Q_E increase, workdone decrease.

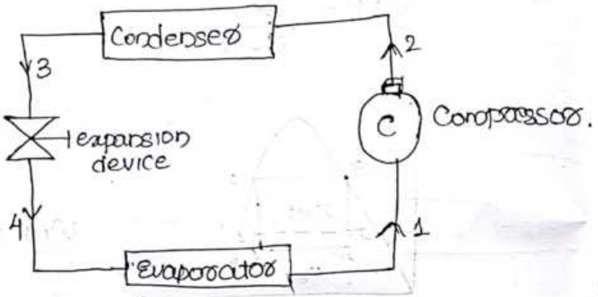
$$COP = \frac{\text{Ref. eff}}{WD}$$

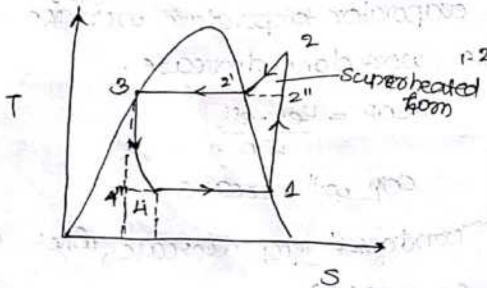
COP will increase.

If condenser temp decrease, Wnet decrease.

COP ↑ (increase).

Standard Vapour Compression Refrigeration System (VCRS)





1-2"-3-4" → carnot

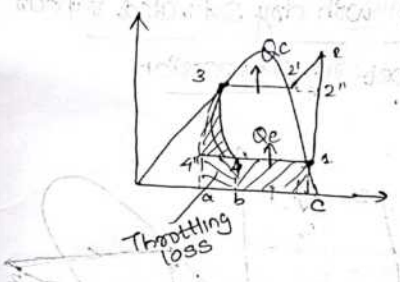
4-1 is not — outside the dome

Process 1-2 : Isentropic compression in the compressor.

2-3 : Isobaric heat rejection in the condenser.

3-4 : Isenthalpic expansion in expansion valve

4-1 : Isobaric heat extraction.



1-2"-3-4" - ideal

1-2-3-4 - actual

ref. effect is less in actual cycle.

4-1 < 4"-1 area.

↓

more area - more heat extraction - more ref. effection

Throttling Loss - Loss due to the isenthalpic process.

less ref. effect - less heat is absorbed

Superheated horn → more work has to be done.

more area under 2, 2', 3.

W = area under 2'-2-3 - area 1-4

Work is more. More work has to be done.

So efficiency reduces

$(Q_e)_{carnot} = a-4'-1-c$

$(Q_c)_{carnot} = c-2''-2'-3-a$

$W_{net} = 1-2''-3-4'$.

$(Q_e)_{VCRS}$ = ~~c-1-4-b~~ b-4-1

$(Q_c)_{VCRS} = a-3-2'-2-c$

$W_{net} = 1-4'-3-2'-2 -$

?

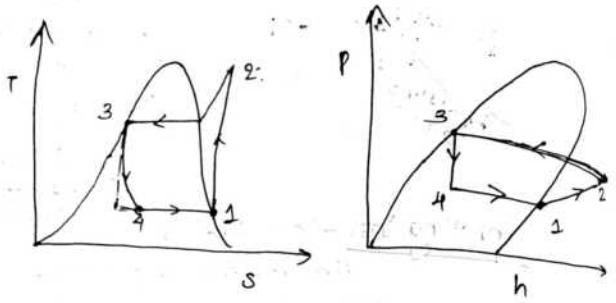
$4-4'-b-a$

W_{net} increase

c.o.p ↓

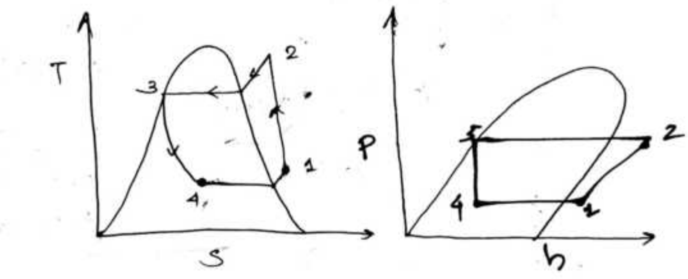
ref. effect decre

Standard VCRS with dry saturated vapour ~~Dry~~ before compression

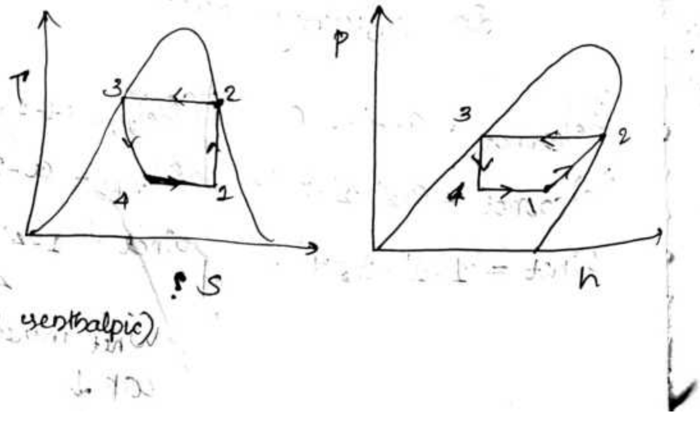


2-3 isobaric
4-1 isobaric

Standard VCRS with superheated vapour before compressor

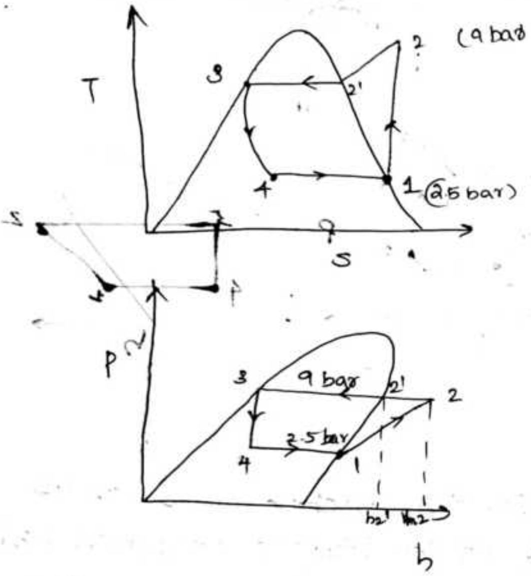


Standard VCRS with dry saturated vapour after compression



(3-4 isenthalpic)

Q : A refrigeration machine using R12 as refrigerant operates btw the pressures 2.5 bar and 9 bar. The compression is isentropic and there is no undercooling in the condenser. The vapour is in dry saturated state at the beginning of the compression. Estimate theoretical coefficient of performance. If the actual coeff. of performance 0.65 of theoretical values calculat the net cooling produced per hour. The refrigerant flow is 5 kg/hour. Take Cp of Superheated vapour at 9 bar as 0.64 kJ/kgK



1-2 isentropic compression

2-3 - isobaric heat rejection

3-4 - isenthalpic expan

4-1 - isobaric

$$COP = \frac{\text{Refrigeration effect}}{\text{Work done}}$$

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

(At constant pressure heat transfer is equal to enthalpy change)

1 is on sat. vapour line.

$h_1 = h_g$ at 2.5 bar.

g: saturated vapor line

f: sat. liquid line

~~2.519~~ ~~$\frac{2.5 - 2.519}{2.519}$~~

$$\frac{x - x_1}{x_2 - x_1} = \frac{y - y_1}{y_2 - y_1}$$

$$\frac{x - 2.3498}{2.5190 - 2.3498} = \frac{y - 349.08}{349.99 - 349.08}$$

Only small change in h_g. (just 0.9)

So take h_g value at 2.519

$h_1 = 349.99$ kJ/kg.

$h_4 = h_3 = h_f$ at 9 bar $= 236.5$ kJ/kg.

h_2 is not available from table.

$S_1 = S_2 = \boxed{S_g}_{2.5\,bar}^{=S_1} = 1.5618$

$S_2 = S_2' + C_{pv} \ln\left(T_2/T_2'\right)$ $\quad S = C_p \ln\left(\frac{T_2}{T_1}\right)$

$S_2' = \boxed{S_g}\,9\,bar = 1.5465$ kJ/kgK. $\quad h = C_p(T_2 - T_1)$

$C_{pv} = C_p$ of superheated vapour

$T_2' = T_{sat}$ at 9 bar $= 38°C = 311$ K.

$1.5618 = 1.5465 + 0.64 \ln\left(T_2/311\right)$

$0.0239 = \ln\left(T_2/311\right)$

$1.024 = T_2/311$

$T_2 = 318.524$ K.

$h_2' = (h_g)$ @ 9 bar

$h_2 = h_2' + C_p (T_2 - T_2')$

$h_2' = 368.1$ kJ/kg.

$h_2 = 368.1 + (0.64 \times (318.524 - 311))$

$= ~~5183.46 kJ/kg~~$

$= 372.915$ kJ/kg.

$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{349.99 - 236.5}{372.915 - 349.99}$

$= 4.95$

$0.65 \times COP = \frac{\text{ref. effect}}{h_2 - h_1}$

$= \frac{\text{ref. eff}}{h_2 - h_1}$

$0.65 \times 4.95 = \frac{\text{Ref eff}}{372.915 - 349.99}$

~~1560.9~~ $= 73.43$ kJ/kg

kJ/kg × kg/hr

Net cooling produced per hour $73.43 \times .5 = 367.197$ kJ/hr

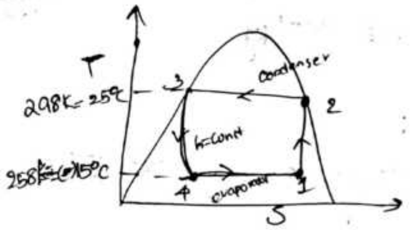
$= 6.119$ kJ/min

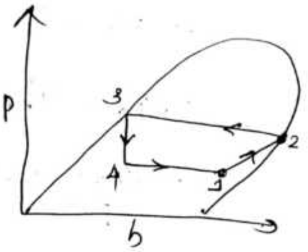
1 TR = 210 kJ/min.

$= \frac{6.119}{210} = 0.029$ TR

Q) 28 tons of ice from and at 0° is produced per day in an ammonia refrigerator. The temperature range in the compressor is from 25°C to -15°C. The vapour is dry and saturated at the end of compression and an expansion valve is used. There is no liquid subcooling. Assume actual COP of 62% of the theoretical. Calculate the power required to drive the compressor.

Total refrigeration - 28 tons of refrigeration.





$$COP = \frac{\text{Ref. effect}}{\text{Work done (compressor)}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$S_1 = S_2 = (S_g)_{25°C}$

$h_2 = (h_g)_{25°C}$

$S_1 = S_{f1} + x(S_{g1} - S_{f1})$ (page 6)

Table 3 page 6. $S_1 = S_2 = (S_g)_{25} = 5.3178$ kJ/kg K.

$h_2 = 1483.105$

$S_{f1} = (S_f)_{-15°C} = 0.74235$ kJ/kg K

$S_{g1} = (S_g)_{-15°C} = 5.8285$ kJ/kg K

$5.3178 = 0.74285 + x(5.8285 - 0.74235)$

$x = \underline{\underline{0.899}}$

$h_1 = h_{f1} + x(h_{g1} - h_{f1})$

~~= 182.67 + 0.899 (399.05)~~

= ~~541.41~~

$h_1 = 131.28 + 0.899(1444.19 - 131.28)$

$= 1311.58$ kJ/kg.

Ref. effect $= h_1 - h_4$

$h_4 = (h_3)_f = 317.92$ @ 25°C.

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1311.58 - 317.92}{1483.105 - 1311.58}$$

$= \underline{\underline{5.79}}$

Actual COP $= 0.62 \times 5.8 = 3.596$

Ref. eff $= 28 \times 335$ kJ/kg $= 9380$ kJ/kg
(TR) (latent heat) $= \underline{\underline{98\text{ kW}}}$

$\frac{28 =}{24 \times 60 \times 60} =$

$$COP = \frac{\text{Ref. eff}}{WD} \Rightarrow WD = \frac{\text{Ref. eff}}{COP}$$

~~WD = $\frac{938098}{3.596}$ = 2608.45 kJ/kg day~~

~~= 0.03 kJ/kg s~~

28 ton → kg

Total heat absorbed per day = 28 × 1000 × 335
= 9380000

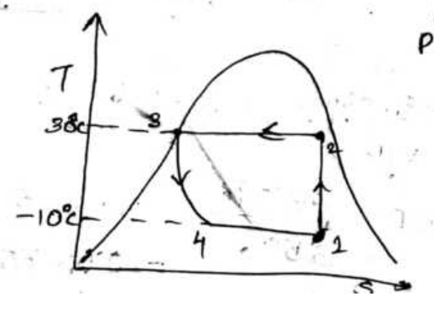
Total Heat absorbed / second = $\frac{9380000}{24 \times 60 \times 60}$ = 108.56 kJ/kg

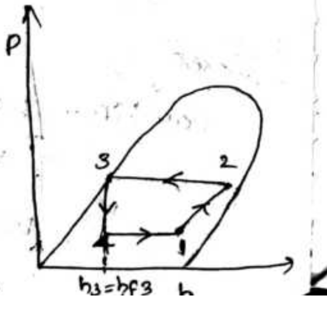
$$COP = \frac{Ref. \ eff}{WD}$$

$$WD = \frac{Ref. \ eff}{COP} = \underline{\underline{80.19}} \ \ W$$

Q. An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of −10°C and 30°C. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the compressor at 30°C. Assuming actual cop as 60% of ~~fuel~~ theoretical cop, calculate the kilograms of ice produced per kW hour at 0°C from water at 10°C.

Ans.





95% after compression

$x_2 = 0.95$.

$$(COP)_{theoretical} = \frac{Ref}{work \ done} = \frac{h_1 - h_4}{h_2 - h_1}$$

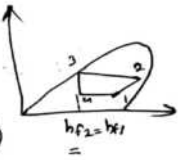
$\underline{h_2 \ (at \ 30° \text{ - ammonia})}$

$h_f = 342.08 \ kJ/kg$ (h at saturated liquid line) $h_g = 1485.93 \ kJ/kg$ (h at satu. vapour line)

$$h_2 = h_{f2} + x_2 \times (h_{g2} - h_{f2})$$

$$= 342.08 + 0.95 (1485.93 - 342.08)$$

$$= \underline{\underline{1428.73}} \ kJ/kg.$$



$\underline{h_1}$ (at −10°C)

find x.

$s_1 = s_2$

$$s_2 = s_{f2} + x_2 (s_{g2} - s_{f2})$$

$$= 1.4892 + 0.95 (5.2623 - 1.4892)$$

$$= \underline{\underline{5.0736}} \ kJ/kg K.$$

$s_1 = s_2 = 5.0136$

$s_1 = s_{f1} + x_1(s_{g1} - s_{f1})$ at -10°C

$= 0.8294 + x_1(5.7559 - 0.8294)$

$x_1 = \underline{\underline{0.86}}$

$h_1 = h_{f1} + x_1(h_{g1} - h_{f1})$

$= 154.03 + 0.86(1450.42 - 154.03)$

$= \underline{\underline{1268.92 \text{ kJ/kg}}}$.

$h_4 = h_3 = h_f @ 30°C$

$= \underline{\underline{342.08 \text{ kJ/kg}}}$.

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1268.92 - 342.08}{1428.13 - 1268.92}$$

$= \underline{\underline{5.8}}$

Actual COP = $0.6 \times 5.8 = \underline{\underline{3.48}}$.

$$(COP)_{actual} = \frac{\text{Ref. effect}}{\text{work}}.$$

per kW hour. So work = 1 kWh

1 kW hour = 3600 kJ.

So W = 3600 kJ.

~~Ref. effect =~~

10°C to 0°C ice →

Amount of Heat to be extracted $= C_p \Delta T + 335$ kJ/kgK (latent heat)

$\underbrace{}_{\text{sensible cooling}}$ ~~= 1.005 × 10 = 10.05 kJ/kg.~~

$= 4.187 \times 10 + \underbrace{335}_{\text{latent}}$

$= \underline{\underline{41.87 \text{ kJ/kgK} + 335}}$

$C_{p\,air} = 1.005$

$C_{p\,water} = 4.187$ kJ/kgK

$= \underline{\underline{376.87}}$ kJ/kgK.

For 1 kg of water = 376.87 kJ/kgK. heat extract

$$(COP)_{actual} = \frac{\dot{m} \times (376.87)}{3600} = 3.48$$

$\dot{m} = \underline{\underline{33.24}}$ kg.

Q. A vapour compression refrigeration machine with freon12 as refrigerant has a capacity of 12 TR operating btw -28°C and 26°C. The refrigerant is subcooled by 4°C before entering the expansion valve and the vapour is superheated by 5°C before leaving the evaporator. The machine has a 6 cylinder single acting compressor with stroke = 1.25 times the bore.

It has a clearence of 3% of the stroke volume.

Determine,

1) Theoretical power required.

2) COP -

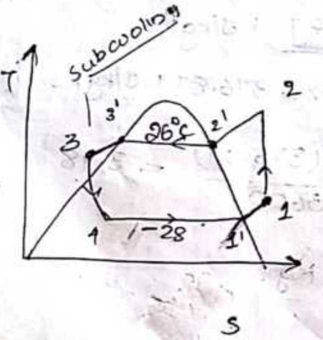
3) Volumetric efficiency

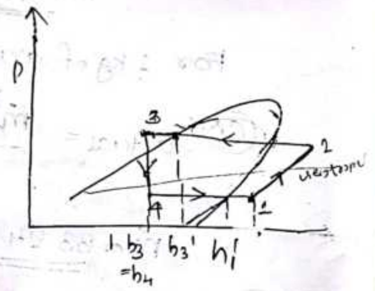
4) Bore and stroke of the cylinder.

Speed of the compressor is 1000 rpm.

Freon 12 - R12

subcooling - before saturation temp





$T_1 = -28 + 5° = \underline{\underline{-23}}\ °C$

$h_1 = h_1' + C_p \Delta T$ @ $-28°C$

$h_1 = 339.76 + (0.578 \times 5)$

$= \underline{\underline{342.62}}$

(b/w temp limit → take saturation temp b/w condenser & evaporator)

(no data about point 2)

$\underline{\underline{h_2 =}}$

Trace point 2

$S_1 = S_2$

$S_1 = S_{g_1} + C_{p_v} \ln (T_1/T_1')$ at

$(S_{g_1}) = (S_g)_{-28°C}$ (-28 sat temp)

$= 1.5762$ KJ/kgK

$C_{pv} = 0.575$

$T_1 = -23 = 250$

$T_1' = -28 = 245$

T in K

$S_1 = 1.5762 + 0.575 \times \ln \left(\frac{250}{245}\right)$

$= \underline{\underline{1.5878}}$ KJ/kgK

$S_1 = S_2 = S_2' + C_{pg_2} \times \ln (T_2/T_2')$

$S_2' = (S_g)_{26} = 1.5498$

$C_{pg_2} = 0.707$

$1.58 = 1.5498 + 0.707 \times \ln \left(\frac{T_2}{299}\right)$

Solve using calculator

$T_2 = 312.048$ K.

$h_2 = h_2' + C_{p\,vapour} \times (312.05 - 299)$

$h_2' = h_g$ at 26 = 363.62 kJ/kg

$C_{pv} = 0.707$

$h_2 = 363.62 + 0.707\,(312.05 - 299)$

$= \underline{\underline{372.846}}$ kJ/kg.

$h_4 = h_f + x(h_g - h_f)$

$h_3 = h_4$

$h_3 = h_3' - C_{p\,liq.} \times (T_s - T)$

$h_3' = h_f$ at 26°C = 224.92 kJ/kg

$C_{pf} = 0.987$.

$\therefore \Delta T = 4°$, (subcooled by 4°)

$h_3 = 224.92 - 0.987\,(4)$

$= 220.972$ kJ/kg

$= h_4$

$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{342.62 - 220}{372.848 - 342}$ ok

$= \underline{\underline{4.024}}$

Ref. $\frac{\text{capacity}}{s}$ = 12 TR.

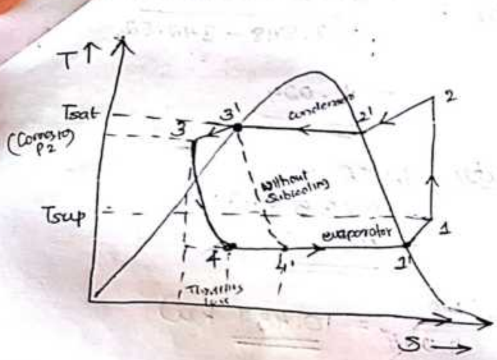
= 12 × 3.5 kW

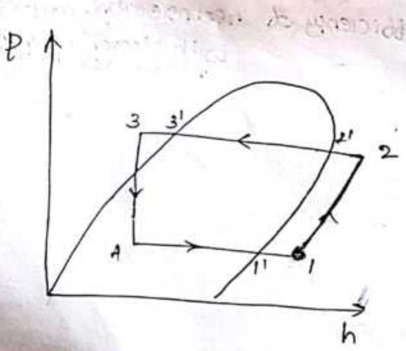
= $\underline{\underline{42 \text{ kW}}}$

$\frac{W.D}{s}$ = Power = $\frac{42}{4.024}$ = $\underline{\underline{10.437}}$ kW

(Volumetric efficiency of reciprocating compressor
with clearence volume).
without clearence volume.

Subcooling





throttling - isenthalpic expansion.

eg. spray

(temp reduce)

For ideal gas no temp decrease for throttling process..

Uses

- * More refrigerating effect 4-1 > 4'-1
- * Throttling. $P_{high} \rightarrow P_{low}$ temp decrease

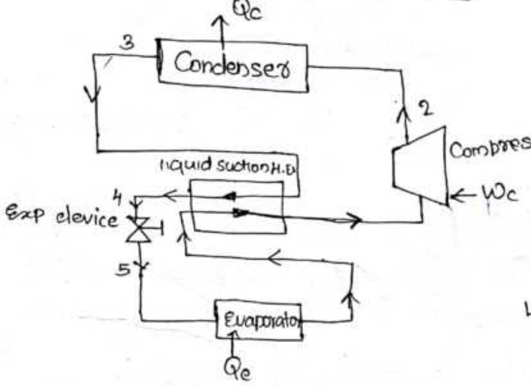
Reduce throttling losses.

Uses of superheating

- * Refrigerating effect increase
- * Ensure that only dry air enter compressor. No problem to compressor.

Liquid Suction heat exchanger

(Khummi problems)



Liquid suction heat exchange - LSHE

To achieve subcooling + superheating.

Evaporator → LSHE → compressor
(superheat)

LSHE is counterflow heat exchanger.

Evaporator draws heat from atmosphere. → less than atmospheric temp.

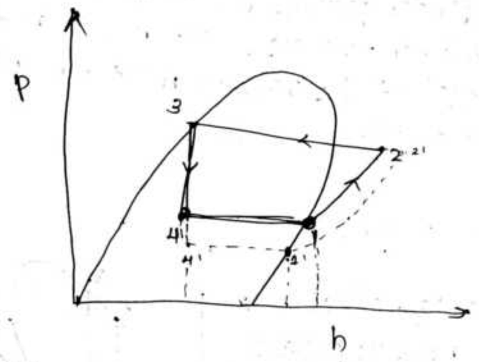
Condenser → give heat to atmos. → > than atmos temp.

heat transfer b/w. two lines take place at LSHE.

Useful superheat → heating in evaporator

Useless superheating → heating take place in pipeline due to loss of insulation of pipe line.

Effect of operating parameters on COP.



1 * • Evaporator pressure (4→1) → If evaporator pressure decrease refrigerating effect reduce.

$h_1 - h_4 \rightarrow h_1' - h_4'$.

~~Wor~~ Work done $h_2 - h_1 \rightarrow h_2' - h_1'$ (more)

∴ work increases.

∴ COP decrease.

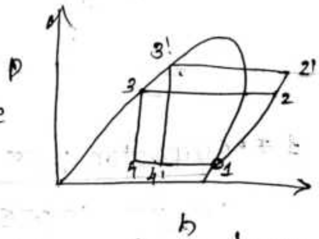
As pressure decrease ; specific volume increase. ∴ large compressor is required. Size of compressor / system increase.

- Refrigeration effect reduces as evaporator pressure decrease. And there is an increase in the workdone. As a result cop of the system decreases. (reduces)

As the evaporator pressure decreases specific volume of the refrigerant increases, As a result size of the compressor has to be increased and that results in increased work done.

2* Condenser pressure :-

Refrigerating effect decrease as condenser pressure increases.



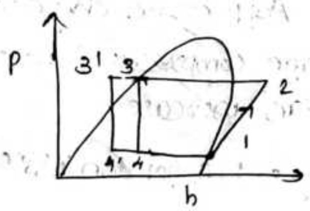
COP decreases.

As condenser pressure increase, condenser temp increase, temp gradient. ~~increase~~ increase. But more work has to be ~~increase~~ done to obtain such large ~~heat~~ compression ratio. ~~We~~ More work by compressor to increase temp of condenser to high value.

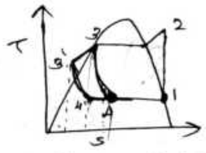
3* Subcooling

$T_3' < T_3$

Subcooling



* Refrigerating effect increase $h_1-h_4 \rightarrow h_1-h_4'$
* No change in work done.
* COP increase
* Reduce throttling ~~process~~ losses.



$h_3-h_4 \rightarrow h_3'-h_4'$

h_3-h_4 – throttling - loss.

$h_3'-h_4'$ → throttling loss

heat loss – no work → enthalpy change.

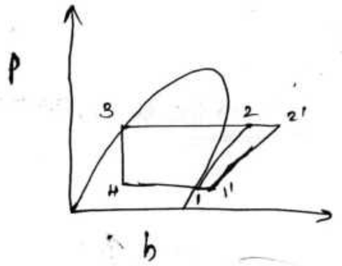
isentropic

→ no heat loss

→ but work done.

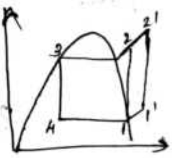
→ Soenthalpy change.

4* Superheating



→ refrigerating effect increase (4-1 to 4-1')

→ work done increase



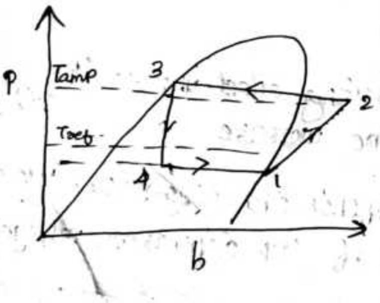
COP depends on refrigerator. We can't predict COP. Value of COP depends on refrigerant.

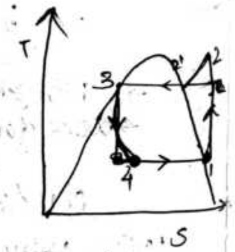
~~A Vapour~~

Q) A vapour compression machine is used to maintain a temperature of -23°C in a refrigeration space. The ambient temperature is 37°C. The compressor takes in dry saturated vapour of R12. A minimum of 10°C temperature difference is required at the evaporator as well as the compressor. There is no subcooling of the refrigerant. If the refrigerant flow rate is 1 kg/min. find out

1) Tons of refrigeration
2) Power required.
3) Ratio of COP of the cycle to the COP of carnot cycle.

Ans.





Evaporator temp = -23-10 = -33°C.

Condenser temp = 37+10 = 47°C

$T_1 = T_4 = -33°C$ $\qquad T_2' = T_3 = 47°C$

$$COP = \frac{h_1 - h_4}{h_2 - h_1}$$

$h_1 \rightarrow h_g$ @ $-33°C$

$h_1 = 336.42$ kJ/kg.

$h = h_f + x(h_g - h_f)$

$h_4 = h_3 = h_f$ @ 47°C

$= 246.32$ kJ/kg.

$S_1 = S_2 = (S_g)_{-33°C} = 1.5824$

$S_2 = S_2' + C_p \ln(T_2/T_2')$ $\qquad C_{pv} = 0.796$

$S_2' = S_g$ at 47°C

$= 1.5440$ kJ/kgK.

given value is saturation temp.
$T_2' = 32$

$1.5824 = 1.5440 + 0.796 \times \ln(T_2/T_2')$

~~$T_2' = 320/$ [illegible] 0.048~~

$T_2' = 304.92$ K.

$$1.5824 = 1.5440 + 0.796 \ln(T_2/320)$$

$$T_2 = \underline{\underline{335.815\ K}}$$

$$h_2 = h_2' + C_{pg}(T_2 - T_2')$$

$$= (h_g)_{at\ 47} + C_{pg}(335.815 - 320)$$

$$= 371.135 + 0.796\ (15.815)$$

$$= \underline{\underline{383.73}}\ kJ/kg$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{336.42 - 246.32}{383.73 - 336.42}$$

$$= \underline{\underline{1.9}}$$

$$\text{Work done} = h_2 - h_1 = 383.73 - 336.42$$

$$= \underline{\underline{47.31}}\ kJ/kg.$$

$$\text{Power required} = \dot{m}(h_2 - h_1) = \frac{1\ kg}{60\ s} \times 47.31 \frac{kJ}{kg}$$

$$= \underline{\underline{0.7885\ kW}}$$

$$\text{Refrig. effect} = \dot{m}(h_1 - h_4)$$

$$TR = \frac{\dot{m}(h_1 - h_4)}{3.5} = \frac{(1/60) \times (336.42 - 246.32)}{3.5}$$

$$= \underline{\underline{0.43}}\ TR.$$

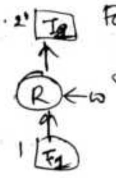
$$\text{Carnot}(COP) = \frac{T_1}{T_2 - T_1}$$

T_1 = ref. space

$= -23°C = 250\ K$

T_2 = Sink temp

$= 37°C = 310\ K.$



For carnot we use source & sink temperature.

$$T_2 / T_1 - T_2$$

$$(COP)_{carnot} = \frac{\text{Temp. of ref. space}}{\text{amb. temp} - \text{ref. space temp}}$$

$$= \frac{250}{310 - 250} = \underline{\underline{4.1667}}.$$

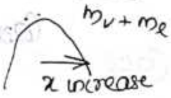
$$\frac{(COP)_{cycle}}{(COP)_{carnot}} = \frac{1.9}{4.1667} = \underline{\underline{0.456}}$$

Multistage VCRS

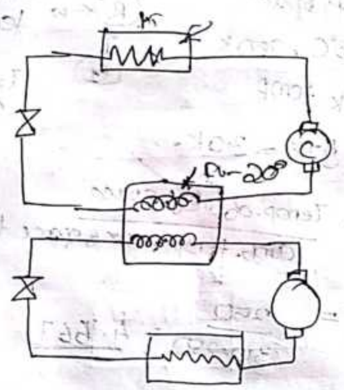
- Multi evaporator system
- Multi compressor system
- Cascade refrigeration.

actual vapour Compression System
ue Khurmi or Rajput.

$x = \frac{m_v}{m_v + m_l}$

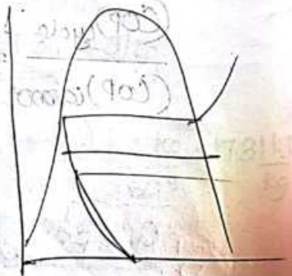


Cascade refrigeration

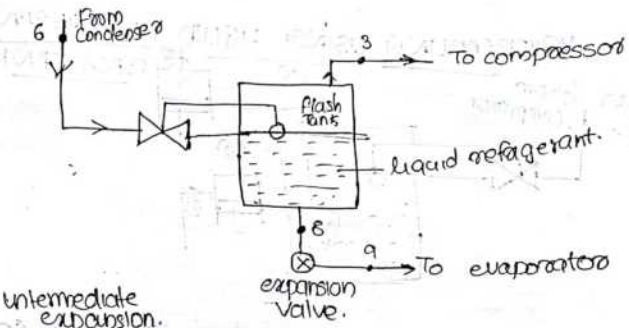


expander of 1 act as evap

Condenser of 1st is evaporator of Second.



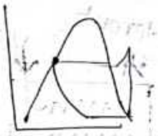
FLASH GAS REMOVAL USING FLASH TANK.



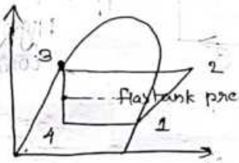
6 → intermediate expansion.

Flash gas has no use.

If more gas, evaporator size needs to be increased.

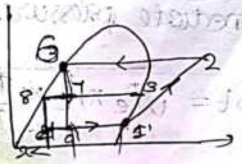


expansion → Quality of gas increase
vapour amount increase
↓
This vapour is flash gas.



(½) intermediate.

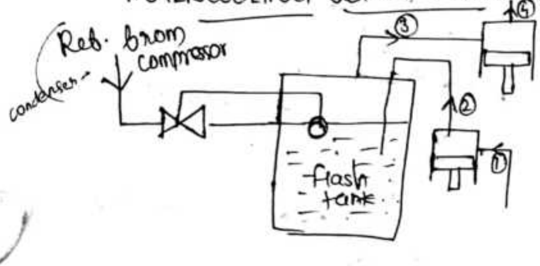
- Vapour to compression
- Liquid to expansion valve, then to evaporator



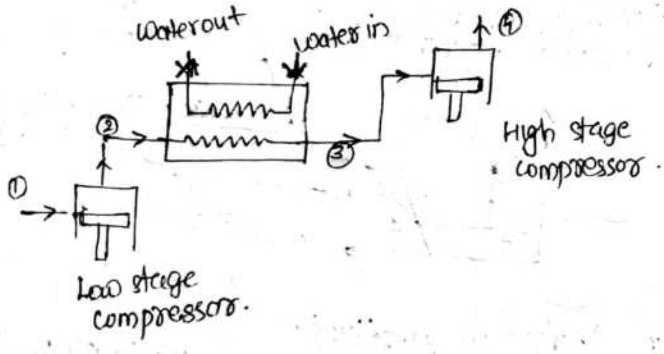
Vapour bypass to com
remaining gas-water to evaporator.

More liquid will be available at evaporator inlet. Quality of liquid is less at inlet.

INTERCOOLING USING LIQUID REFRIGERANT IN FLASH TANK

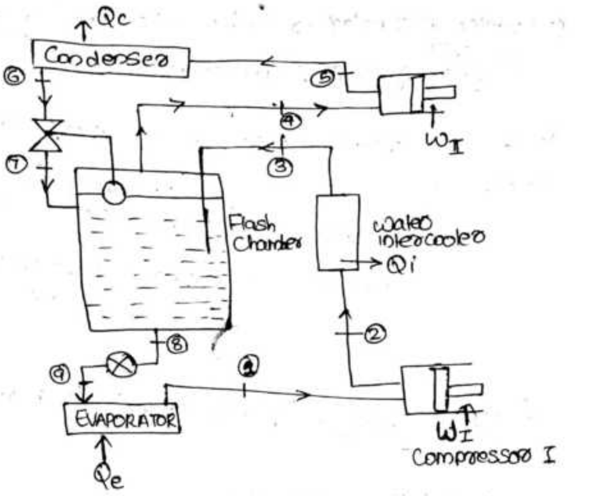


INTERCOOLING USING External water cooled Heat exchanger

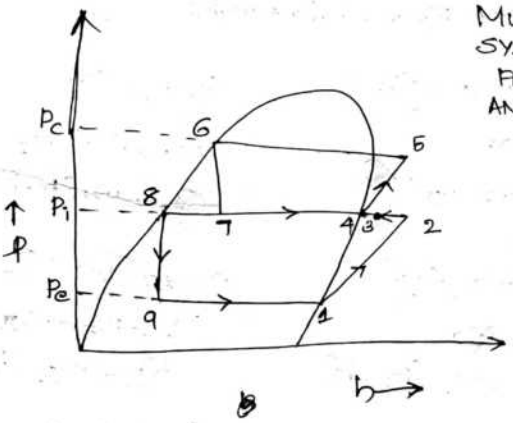


Optimum intermediate pressure for cooling

$$P_{i,opt} = \sqrt{P_e \times P_c \times \frac{T_c}{T_e}}$$



MULTISTAGE SYSTEM WITH FLASH GAS REMOV AND INTERCOOLIN



① In Intercooler : Isobaric heat reject.
Outlet of intercooler — 3
superheated not saturated

① Inter Cooling → Cool spec
volume decrea
work done decreas

(III)) water intercooler → flash chamber .. Flash chamber sat. vapour gas goes to second compressor. (convert superheat to saturated

(IV) 6-7 – intermediate float valve .. Maintain particular level. ~~to~~

Vapour fed to ~~expansion valve~~ (compressor II.

liquid fed to another expansion valve and then to evaporator.

Mass & energy balance of flash chamber.

$$M_7 + m_3 = m_4 + m_8$$

$$m_7 h_7 + m_3 h_3 = m_4 h_4 + m_8 h_8$$

Expansion valve.

$m_8 = m_9$

$h_8 = h_9$ (isenthalpic)

Evaporator

$m_9 = m_1 = m_I$

$Q_e = m_1 (h_1 - h_9)$

Compressor -I

$m_1 = m_2$

$W_I = m_1 (h_2 - h_1)$

Compressor II

$m_4 = m_5 = m_{II}$

$W_{II} = m_{II} (h_5 - h_4)$

Water Intercooler

$m_2 = m_3 = m_1$

$Q_I = m_1 (h_2 - h_3)$

Condenser

$m_5 = m_6 = m_{II}$

$Q_c = m_{II}^{\times} (h_5 - h_6)$

FLOAT VALVE

$m_6 = m_7 = m_{II}$ (valve to flash chamber.

$h_6 = h_7$.

From above equations,

$$m_7 = m_4 = m_{II}$$

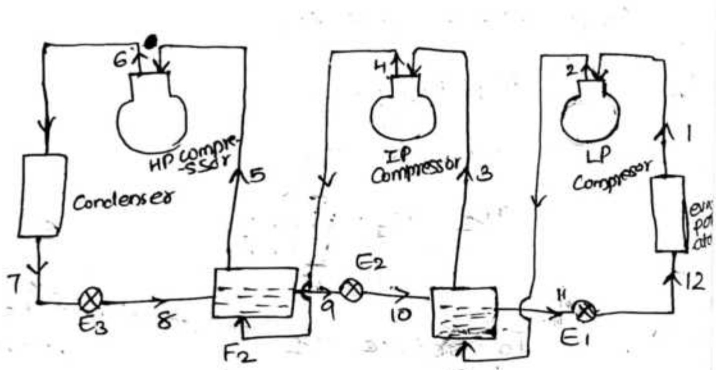
$$m_3 = m_8 = m_I$$

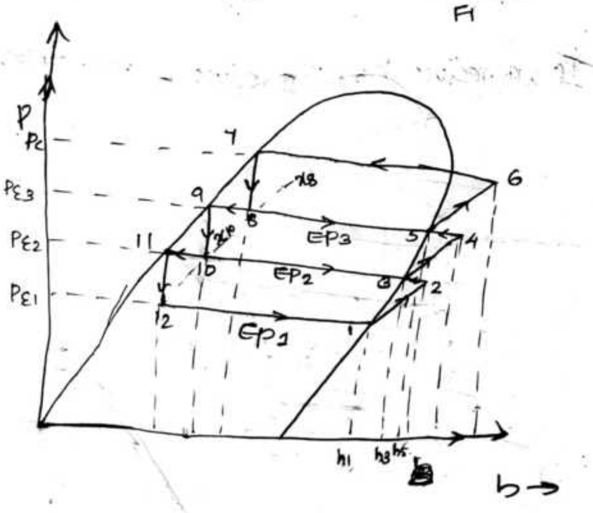
$$m_{II} = m_I \times \left[\frac{h_3 - h_8}{h_4 - h_7}\right]$$

To reduce m_{II}, reduce value of $(h_3 - h_8)$ or increase $(h_4 - h_7)$

If we reduce h_3, m_{II} reduce → work can be reduced.

3 Three stage Compression with Multiple expansion ~~pooce~~ valves and flash Intercooler ~~chamber~~ Intercooler.





eg. $x_8 = 0.2$
(80% water) (20% vapour to 5 (EP3))

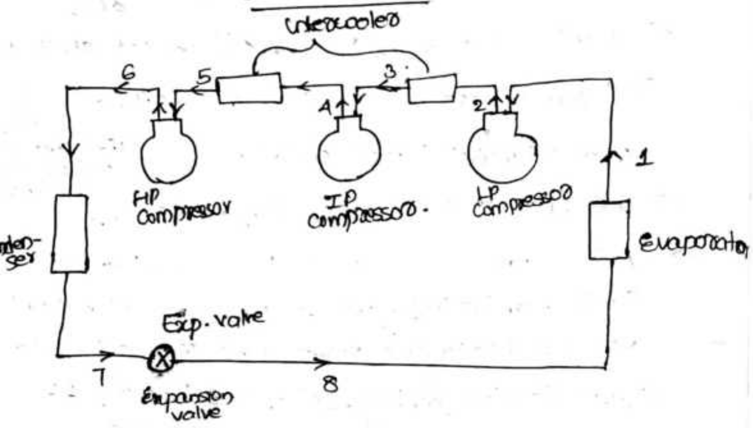
1 : evapor. pressure, small → LP compressor. – P ↑, T ↑
vapour superheated → intercooler flash chamber at P_2
(seperate liquid vapour) → 3 ↑ only saturated vapours move
3-4 IP compressor → 4-5 intercooling. Superheated ↓ saturated (to reduce work) to effective lubrication
: 4-5 saturated liquid cooling.
→ 5-6 HP compressor → condenser 6-7 →
→ 7-8 expansion in E3 to Pressure P8,
Flash chamber → ~~liquid~~ ~~sa~~ to make saturated liquid vapour fed to compressor → 9-10 expn. in E2
again flash chamber, saturated liquid 11.

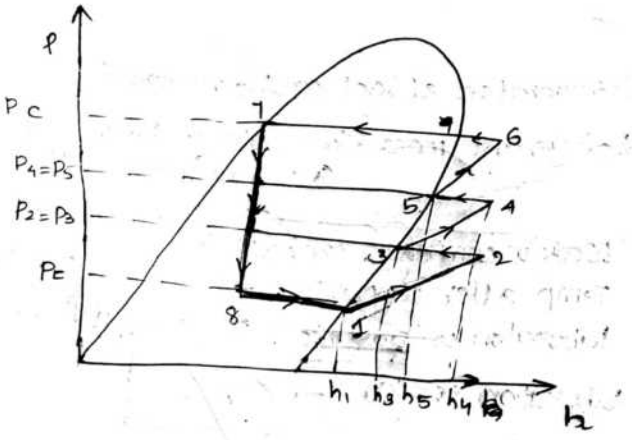
11-12 → expansion in E3.

Refrigeration effect can be increased.
Refrigerant mass flow rate can be ~~increased~~ decreased.
(less mass can produce more effect)

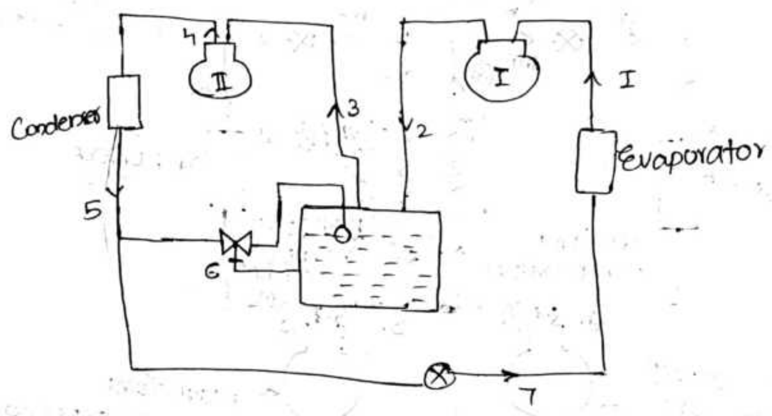
Work in compressor can be reduced.
Temp after compressor can be reduced. Effective lubrication is possible. (stages). If one stage lubrication problems.

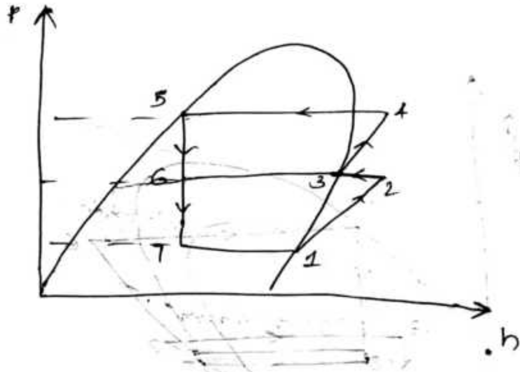
Three stage compression with Water Coolers





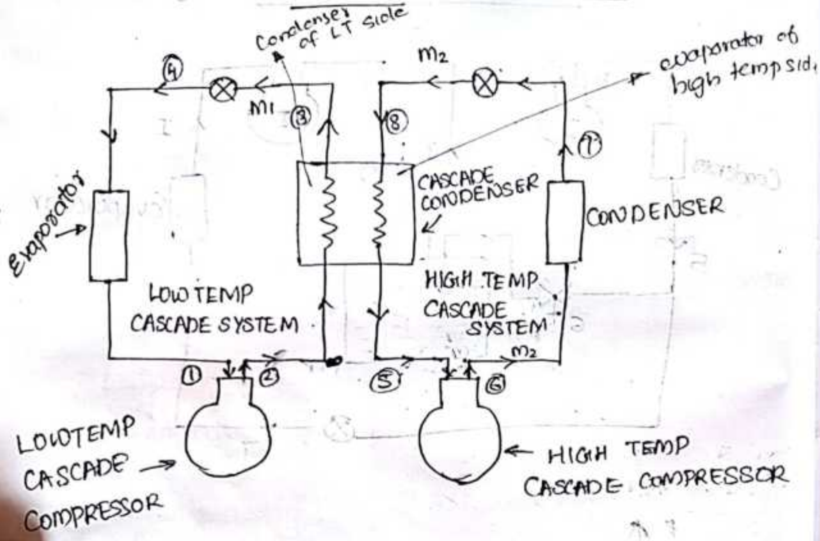
Use of Flash tank for Intercooling.

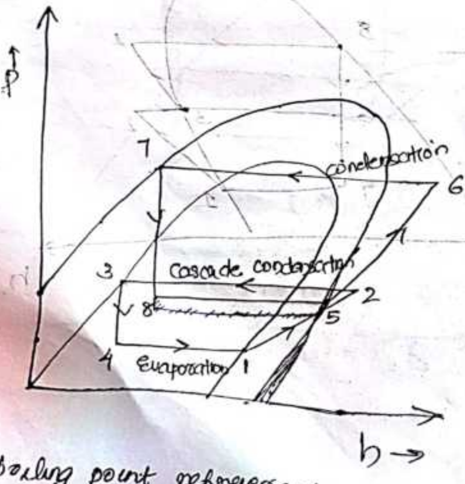




Flash tank is used only for intercool
Not taken during expansion from flash tan

CASCADE REFRIGERATION SYSTEM





2 diff. boiling point refrigerants are used.

Disadvantage of multistage.

- Low freezing point
  refrigerant must have high critical temp } for one refrigerant

Cascade

2 refrigerants are used.

One having low temp freezing point.

Application

→ liquefaction systems

Module 5

PSYCHROMETRY

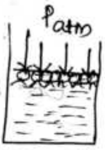
module, then

Dry air :- No water content:

Saturated air - max water content that ~~it~~ can hold.

Boiling :- At saturation temp. Phase change at saturation temp.

Evaporation :-



Boiling is a bulk phenomenon. ← Upper molecules have a tendancy to go out. They escape from surface - evaporation

Evaporation surface phen.

Dry air + water vapour → moist air.

Absolute humidity : mass of water content

Relative humidity : ~~Ratio~~

(RH=80)
Air is 80% saturated
RH=80

RH is very large
They can hold only 20% water ~~can~~ more
ie why air cooler is not effective here

Dry Air

Dry bulb temperature

Wet bulb temperature

$P_t = P_a + P_v$

P_a ↓ Dry air, P_v ↓ water vapour

Partial pressure = $\frac{P_a}{P_a+P}$

$\boxed{PV = mRT}$ $= m\frac{\bar{R}}{M}T$ — universal gas const

R — ~~universal~~ specific gas constant

$\boxed{PV = n\bar{R}T}$

$\bar{R}$ ↓ universal

STP - Standard temp & pressure

NTP - Normal "

At STP, V = 22.48

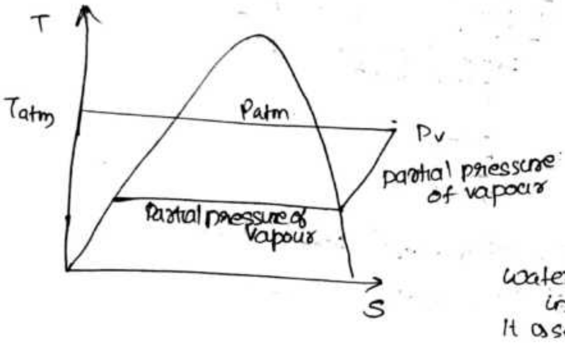
*** ~~For~~ one mole of gas all gases have same volume.

Relative Humidity (ϕ)

It is defined as the ratio of mole fraction of water vapour in moist air to the mol fraction of water vapour in saturated air.
(max. water content is their)

Humidity ratio ~~or~~ specific humidity. (W)

Humidity ratio is the mass of water vapour associated with each kg of ~~air~~ dry air (air collect contain 1 kg dry air – mass of water vapour in that collected ~~ass~~ ~~area~~ air is known as W)

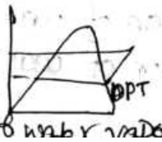


water vapour is in steam condition
It is seen in superheated form

As pressure ↓, boiling point ↑

Dew Point temperature

If unsaturated moisture is cooled at constant pressure, the ~~the~~ temperature at which moisture begins to condense is known as dew point temperature (DPT)



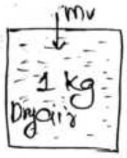
It is the temp ~~of~~ air at which water vapour in air starts condensing into water.

Tsat corresponding to partial pressure of water vapour

~~Degree of sat S~~

Degree of saturation (μ)

Degree of saturation is the ratio of ~~humidity rate~~ humidity ratio to the humidity ratio of saturated mixture at the same ~~temper~~ temperature and pressure.



$$\mu = \left|\frac{W}{W_s}\right|_{t,P}$$

W_s : humidi ratio of saturate mixtu

WBT (wet bulb temperature)

WBT < DBT

Wet Bulb.

~~Vapour~~ depression

Dry BT   It is difference btw DBT & WB

Wet BT

Dewpoint depression

Difference btw ~~DBT~~ dew point temp. and ~~WBT~~ DBT

Humidity ratio = $w = \frac{m_v}{m_a}$

$PV = mRT$

$P_v V_v = m_v R_v T$

v ⇒ vapour

$P_a V_a = m_a R_a T$

$$\frac{m_v}{m_a} = \frac{P_v \cancel{V_v} \times R_a \cancel{T}}{P_a \cancel{V_a} \times R_v \cancel{T}}$$

$V_a = V_v$: same vol is occupied by air and vapour

$$\frac{m_v}{m_a} = \frac{P_v R_a}{P_a R_v}$$

$R_a = 0.287$ kJ/kg K.

$R_v = 0.462$ kJ/kg K.

$$\frac{m_v}{m_a} = \frac{0.287 P_v}{0.462 P_a} = 0.622 \frac{P_v}{P_a}$$

$$W = \frac{m_v}{m_a} = 0.622 \frac{P_v}{P_a}$$

Relative humidity, $\phi = \frac{n_v}{n_s}$

$$\phi = \frac{n_v M}{n_s M} = \frac{m_v}{m_s}$$

n_v = no. of moles of vapour
n_s: no. of moles of vapour of saturated vapour
m_v: mass of vapour
m_s: mass of saturated vapour

$P_v V_v = m_v R_v T$

$P_s V_s = m_s R_s T$

V: vapour in moist air.
s: saturated

$m_v = \frac{P_v V_v}{R_v T}$

$m_s = \frac{P_s V_s}{R_s T}$

m_v: mass of vapour in moist air
m_s: mass of vapour in saturated air.

$R_s = R_v$ since both are ~~air~~ vapour

$$\phi = \frac{m_v}{m_s} = \frac{P_v \cancel{V_v}}{R_v \cancel{T}} \times \frac{R_s \cancel{T}}{P_s \cancel{V_s}}$$

$$\phi = \frac{P_v}{P_s}$$

P_v: partial pressure of vapour in moist air
P_s: partial pressure of vapour in saturated air.

Q. The humidity ratio of atmospheric air at 28°C dry bulb temperature and 760 mm of mercury is 0.016 kg/kg of dry air. Determine

1) Partial pressure of water vapour.
2) Relative humidity.
3) Dew point temperature.

Ans. DBT = 28°C

Total pressure = ~~760 mm of mercury~~ P_b $= P_a + P_v$

$W = 0.016$ kg/kg of air

~~760 m~~

$P_b = 760$ mm of Hg.

$$W = 0.622 \frac{P_v}{P_a} = 0.622 \frac{P_v}{P_b - P_v}$$

$$0.016 = 0.622 \times \frac{P_v}{(760 - P_v)}$$

$760 - P_v = 38.875\ P_v$

$760 = 39.875\ P_v$

~~$P_v = 20.066$ mm of Hg~~.

$P_v = 19.059$ mm of Hg.

19.059
~~Go~~ CONV 27
in calc

$= 2540.99$ Pa

$P_v = 0.025406$ bar.

P_s : Saturation pressure at 28°C

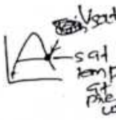
$= 0.03778$ bar

Relative humidity $\phi = P_v/P_s$

$$= \frac{0.025406}{0.03778} = 0.672$$

Dew point temp is the ~~ter~~ saturation temp corresponding to vapour pressure.

(At that pressure, below that temp dews will be formed.)



$T \simeq 21°C$

Q. A room 7×4×4 m is occupied by an air-water vapour mixture at 38°C. The atmospheric press. is 1 bar and relative humidity is 70%. Determine the humidity ratio, dew point, mass of dry air and mass of water vapour. If the mixture of air-water vapour is further cooled at constant pressure until the temp is 10°C. Find the amount of water vapour condensed.

Ans. $V = 7 \times 4 \times 4 = 112\ m^3$.

$\phi = RH = 70\%$.

Dry bulb temp = 38°C.

~~w=?~~ $W = 0.622\ P_v/P_a$

$\phi = \frac{P_v}{P_s} = 0.7$

Table ~~20~~39
page 44
Properties of Steam for Psychrometric appl.

P_s : Satu. pressure at 38°.

$P_s = 0.06632$ bar.

$P_v = 0.7 \times P_s = 0.7 \times 0.06632$
$= 0.04642$ bar.

$W = 0.622 \, P_v/P_a$

Total pressure $P_b = 1$ bar $= P_a + P_v$

$P_a = 1 - 0.04642$
$= 0.9535$ bar

Humidity ratio, $W = 0.622 \times \frac{0.04642}{0.9535} = \underline{\underline{0.03028}}$ kg/kg of air

Dew point temp, Tsat corresponding to Pv.

$P_v = 0.04642$ bar

$$\frac{0.04642 - 0.04496}{0.04759 - 0.04496} = \frac{x - 31}{32 - 31}$$

$x = 31.56°C$

Dew point temp $= \underline{\underline{31.56°C}}$

Mass of dry air

$P_a V_a = m_a R T$ $\qquad T = 311$ K

$0.9535 \times 10^5 \times 112 = m_a \times 287 \times 311$

$m_a = \underline{\underline{119.64 \text{ kg}}}$

$P_v V_v = m_v R T$ $\qquad \boxed{R_v = 461 \text{ J/kgK}}$

$0.04642 \times 10^5 \times 112 = m_v \times \cancel{287} \times 311 \times 461$

$m_v = \underline{\underline{\cancel{5.82 \text{ kg}}}} \; \underline{\underline{3.626}}$ kg.

or

$W = \frac{m_v}{m_a} \Rightarrow m_v = \underline{\underline{3.6226 \text{ kg}}}$

Dew point temp $= 31.56°C$
temp reaches 31.56°C air becomes saturated.

$10 < 31.56°C$

$P_b = P_a + P_s$ $\qquad P_v = P_s$ at 10°C saturated at 10°C

$1 = P_a + 0.012276$ $\qquad P_v = P_s = 0.012276$ b

$P_a = 0.987724$ bar

< 31.56 water got condensed & go out

$P_s V_v = m_v R_v T$

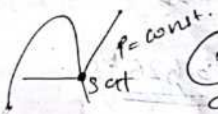
$10^5 \times 0.012276 \times 112 = m_v \times 461 \times 283$

$m_v = 1.0538$ ~~×10⁻⁵~~ kg

Mass of vapour condensed $= (m_v)_{38^\circ C} - (m_v)_{10^\circ C}$

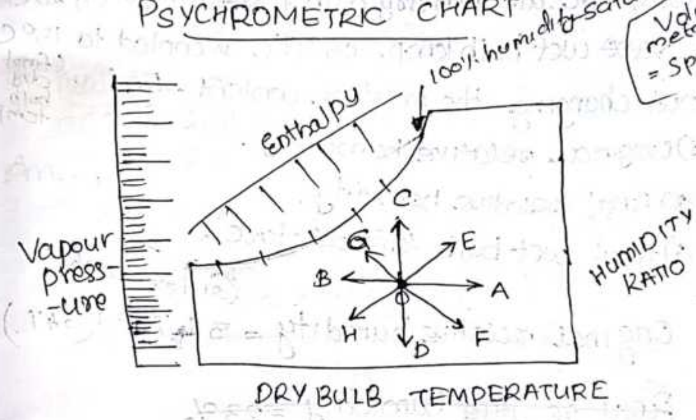
$= $ ~~49~~ $3.626 - 1.0538$

$= \underline{\underline{2.572}}$ kg



(Vapour in air at 38.6 (superheated) and 31.56 (du are same. Below 31.56 water condense & mass decrease)

PSYCHROMETRIC CHART



(Volume cubic metre/kg of dry air = Specific volum

OA - heating OB - Cooling.

OC - humidification OD - Dehumidification

OE - Heating ~~of~~ & humidification.

OF - Heating & dehumidification.

OG - cooling & humidification

OH - cooling & dehumidification.

100% humidity - saturation line.

Dew point temp : temp at saturation line. (Corresponding to vapour pressure)

Q. Atmospheric air with dry bulb temperature of 28°C and ~~vapo~~ wet bulb temp. of 17°C is cooled to 15°C (final dry bulb temp.) without changing the moisture content. Find

1) Original relative humidity.
2) Final relative humidity.
3) Final wet bulb temperature.

(31 from graph)

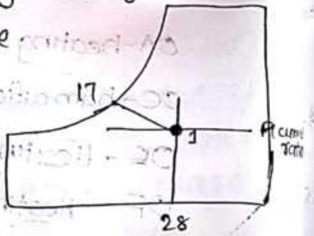
Original relative humidity = 34% (34%)

~~Final relative humidity = 23%~~

~~Final wet bulb temperature~~

Cooled without changing moisture content

Humidity ratio constant



Final relative humidity = ~~43%~~ ~~69%~~ 71%

Cooling means decrease in dry bulb temp

Final wet bulb temp = 12.8°C

(Cooling no humidification.) Air saturated

RH increase

WBT decrease

15/10/15

Q) A quantity of air having a volume of 300 m^3 and at 30°C drybulb temp at 25°C wet bulb temp is heated to 40°C DBT estimate the amount of heat added. Final relative humidity and WBT.

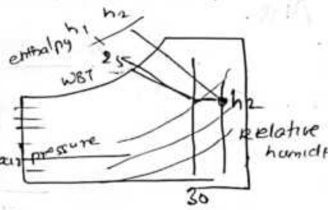
Ans.

~~$h_1 = 82$ kJ/kg~~

$h_1 = \cancel{78.2}$ 77 kJ/kg

$h_2 = 87$ kJ/kg.

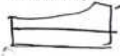
Enthalpy change = heat added (since const pressure)



Specific volume $v_1 = 0.883$ m^3/kg.

$$m_a = \frac{V}{v_1} = \frac{300}{0.883} = 339.75 \text{ kg}$$

(Only heating no moisture content. So specific humidity remains constant)



Heat added $= m_a(h_2 - h_1)$

$= 339.7 \times (87 - 77)$

$= 3397$ kJ

Final relative humidity = 38%.

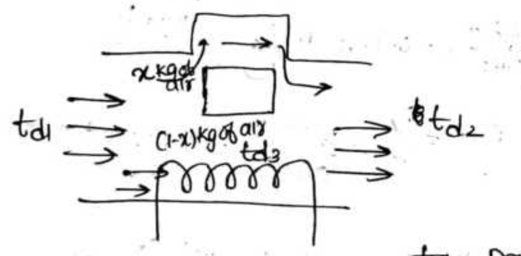
= 38%

Final WBT = 27.5°C

Q. Atmospheric air at 760 mm Hg, dry bulb temp 15°C and WBT 11°C enters a heating paint whose temperature is 41°C. Assuming bypass factor of heating coil as 0.5 determine DBT, WBT and relative humidity of the air leaving the point. Also determine the sensible heat added to the air per kg of dry air.

Ans.

Bypass factor:



x kg of air is bypassed (temp t_{d1} as it bypass

~~t_{d1}~~ Dry BT = t_d

$$x \times \cancel{C_{pm}} \times t_{d1} + (1-x)\cancel{C_{pm}}\, t_{d3} = 1 \times \cancel{C_{pm}} \times t_{d2}$$

$$0.5 \times \cancel{C_{pm}} \times t_{d1} + 0.5\, \cancel{C_{pm}}\, t_{d3} = t_{d2}$$

~~$x = t_{d2}$~~

t_{d3} - coil temp.

$$x\, t_{d1} + t_{d3} - x\, t_{d3} = t_{d2}$$

$$x = (t_{d2} - t_{d3}) / (t_{d1} - t_{d3})$$

Bypass factor, $BPF = \dfrac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$ } for cooling

$BPF = \dfrac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$ → Heating

for cooling t_{d3} is lowest

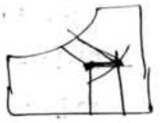
for heating t_{d3} is highest.

Here heating coil. So,

$$BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

$$0.5 = \frac{41 - t_{d2}}{41 - 15}$$

$$t_{d2} = \underline{\underline{28^\circ C}}$$



DBT at exit = $\underline{\underline{28^\circ C}}$.

WBT at exit = $\underline{\underline{15.5^\circ C}}$

Relative humidity of the air leaving the point $= \cdot 25^\circ C$

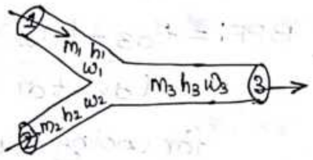
Sensible heat added to air per kg of dry air

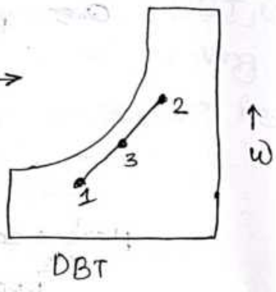
$$Q = mC_p\Delta T = 1 \times 287 \times (28 - 15).$$

$$= \underline{\underline{3731\ J}}$$

$Q = mC_p\Delta T$

Adiabatic mixing of Air streams





ω: humidity ratio/
specific humidity.

$$m_3 = m_1 + m_2$$

$$m_1 h_1 + m_2 h_2 = m_3 h_3$$

$$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3 \quad \cancel{= m_1 h_3 + m_2 h_3}$$

$$m_1 h_1 + m_2 h_2 = m_1 h_3 + m_2 h_3$$

$$m_1 (h_1 - h_3) = m_2 (h_3 - h_2)$$

$$\boxed{\frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3}}$$

Similarly,

$$m_1 \omega_1 + m_2 \omega_2 = m_3 \omega_3$$

$$m_1 \omega_1 + m_2 \omega_2 = (m_1 + m_2) \omega_3$$

$$\boxed{\frac{m_1}{m_2} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_3}}$$

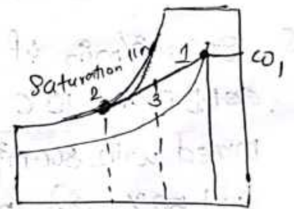
Q. 1 kg of air at 40°C dry bulb temperature and 50% relative humidity is mixed with 2 kg of air at 20°C DBT and 20°C dew point temperature. Calculate the temperature and specific humidity of the mixture.

[DBT = dew point temp : when saturation.]

Ans.

State 1

40° DBT & 50% relative humidity.



~~ω_1 = humidity rato = 240.024~~ kg

~~ω_2 = 15.5~~ 0.0158

ω_1 = humidity ratio = 0.0238 kg/kg of dry air

ω_2 = 0.0145 kg/kg of air

$$\frac{m_1}{m_2} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_3}$$

$$\frac{1}{2} = \frac{\omega_3 - 0.0145}{0.0238 - \omega_3}$$

~~$2\omega_3 - 0$~~.

~~ω_3~~ $2\omega_3 - 0.029 = 0.0238 - \omega_3$

$3\omega_3 = 0.0528$

$\omega_3 = \underline{\underline{0.0176}}$ kg/kg of dry air

$(DBT) = 26°C$

Q. 800 m³/min of free circulated air at 22°C DBT and 10°C dew point temperature is to be mixed with 300 m³/min of fresh air at 30°C DBT and 50% RH. Determine the enthalpy, specific volume, humidity ratio and dew point temperature of the mixture.

Ans. State 1

DBT = 22°C

Dew point = 10°C

State 2

DBT = 30°C

RH = 50%

Specific volume (m³/kg of air)

Specific volume at point 1

= 0.848 m³/kg

$h_1 = 42$ kJ/kg of air

RH = 47%

(Dew point temp is at a particular vapour pressure so ____)

$$\text{Mass} = 800/v_{s1}$$

$= 800/0.848 = \underline{\underline{943.39 \text{ kg/min}}}$

$$\frac{m^3/min}{m^3/kg}$$

$m_1 = 943.39$ kg/min

Specific vol. at point 2,

$v_{s2} = 0.878$ m³/kg

$h_2 = 64$ kJ/kg.

$m_2 = 300/0.878 = 341.685$ kg/min

$$\frac{m_1}{m_2} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_3}$$

$$\frac{943.39}{341.685} = \omega_3$$

$$\frac{m_1}{m_2} = \frac{h_3-h_2}{h_1-h_3}$$

$$\frac{943.39}{341.68} = \frac{h_3-64}{42-h_3}$$

$$39622.38 - 943.39\,h_3 = 341.68\,h_3 - 21867.52$$

$$1285.07\,h_3 = 61489.9$$

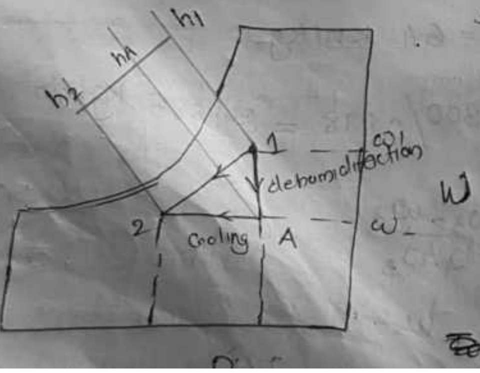
$$h_3 = 47.8494 \text{ kJ/kg}.$$

Dew point temp = 12°C

Specific volume = 0.86 m^3/kg

Humidity ratio = 40%.

Sensible Heat Factor



summer.
atmos temp high

AC – cooling dehumidify

cool ⇒ humidity increases more water vapo in air.
So dehumidify

Total heat = $h_1 - h_2$

Sensible heat = $h_A - h_2$

latent heat = $h_1 - h_A$

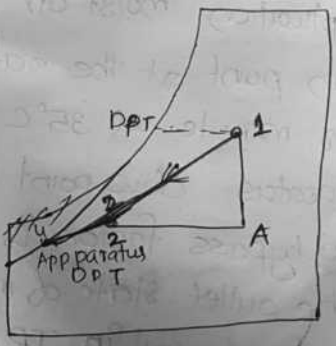
Sensible heat factor, $SHF = \frac{SH}{SH+LH} = \frac{h_A - h_2}{h_1 - h_2}$

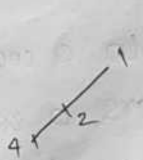
(total heat)

Apparatus Dew Point Temperature (ADP)

ADP < Air DPT → ~~air~~ water will condense.

ADP < supply air temp → Then only dehumidification take place.





4 – apparatus tem

2 – Temp After mixin of two.

4 – eg. 18°C

2 – eg. 22°C

Bypass factor (BPF)

$$BPF = \frac{t_{d_2} - t_{d_4}}{t_{d_1} - t_{d_4}}$$

t_d - dry bulb temp

t_{dp} - dew point temp

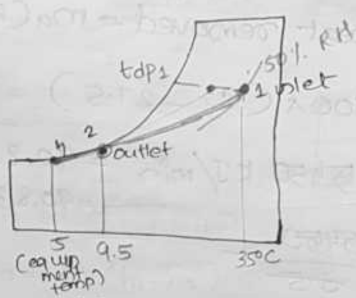
$$= \frac{t_{d_2} - ADP}{t_{d_1} - ADP}$$

$$BPF = \frac{\omega_2 - \omega_4}{\omega_1 - \omega_4}$$

$$BPF = \frac{h_2 - h_4}{h_1 - h_4}$$

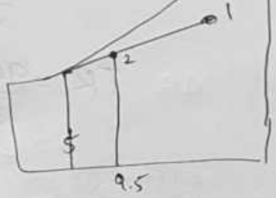
Q. In a cooling application moist air enters a refrigeration point at the rate of 100 kg of dry air per minute at 35°C and 50% RH. The apparatus dew point of the coil is 5°C and bypass factor is ~~0.5~~ 0.15. Determine the outlet state of moist air and cooling capacity of coil in TR.

Ans.



$t_{dp_1} = 23.2$

$t_{d_4} = 5°C$



$$BPF = \frac{t_{d_2} - t_{d_4}}{t_{d_1} - t_{d_4}}$$

$$0.15 = \frac{t_{d_2} - 5}{\cancel{23}\ 35.5}$$

$t_{d_2} = 9.5°C$

$\phi_2 = 96\%$

$\phi_1 = 50\%$

$t_{dp_1} = 23.2°C$

$t_{dp_2} = 9°C$

$h_1 = 82$ kJ/kg

$h_2 =$ ~~30 kJ/kg~~ 27.5 kJ/kg

specific volume at point 2 = 0.815 m^3/kg

27.5

Total heat removed = $m_a(h_1 - h_2)$

$= 100 \times (82 - 27.5)$

$= 5450$ kJ/min $= 90.83$ kJ/s
$= 90.83$ kW

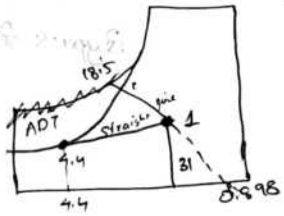
~~$TR = \frac{5450}{3.5}$~~ =

$TR = \frac{90.83}{3.5} = 25.95 = \underline{\underline{26\ TR}}.$

Q. 39.6 m³/min of ~~the~~ a mixture of recirculated room air and outer air enters a cooling ~~point~~ coil at 31°C dry bulb temp & 18.5°C WBT. The effective surface temp of the coil is 4.4°C. The surface area of the coil is such as should give 12.5 kW of refrigeration with given entering air stream. Determine the DBT & WBT of the air leaving the coil and the bypass factor.

Effective coil temp – apparatus dew point temp.

Lewis number



Cooling effect = 12.5 kW

$12.5 = m_a(h_1 - h_2)$

Volume flow rate = 39.6 m³/min

Specific volume at point 1

$v_{s_1} = 0.$~~8[illegible] m³/kg~~ . 0.872 m³/kg

$m_a = \frac{V}{v_s} = \frac{39.6}{0.8[illegible]} = 45.412$ ~~44.097~~ kg/min

= ~~0.734 kg/s~~ $= \underline{\underline{0.756\ \text{kg/s}}}$

$h_2 = \underline{\underline{53\ \text{kJ/kg}}}$

kw =k

$12.5 \times 10^3 = 0.756(53 \times 10^3 - h_2)$

~~$h_2 = 52.$~~

$h_2 = \underline{\underline{36.465\ \text{kJ/kg}}}.$

DBT of mixture leaving = 18°C.

WBT of mixture leaving = $\underline{\underline{12.7°C}}$

~~Bypass factor~~ $= \frac{\cancel{h_1 - h_2}}{h_1 - h_4}$

$= \frac{53 - 36.465}{\cancel{53 - 19}}$

$= \cancel{0.486}$

$$BPF = \frac{h_2 - h_4}{h_1 - h_4}$$

$= \frac{36.465 - 18}{53 - 18}$

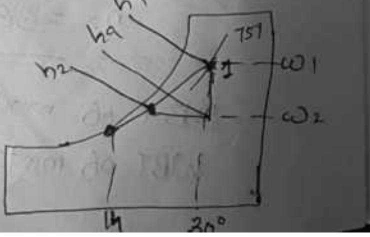
$\cancel{= 0.41} \quad = 0.52$

Q. Atmospheric air at 30°C DBT and 75% RH enters a cooling coil ~~at the~~ rate of 200 m³/min. The coil dew point temp is 14.0°C & BPF of coil is 0.1. Determine the temp of air leaving the coil. The capacity of coil in TR. The amount of ~~air~~ vapour removed per minute & sensible heat factor for the process.

ns.

$h_1 - h_a \rightarrow$ latent

$h_1 - h_2 \rightarrow$ total heat



$h_1 = 83$ kJ/kg

$v_{s1} = 0.888$ m³/kg.

$m_a = \frac{V}{V_s} = 200/0.888 = 225.225$ kg/min

$= 3.75$ kg/s

$$BPF = \frac{h_2 - h_4}{h_1 - h_4}$$

$0.1 = \frac{h_2 - 39}{83 - 39}$

$h_2 = 43.4$ kJ/kg.

$(DBT)_2 = 15.8$ °C $\quad (WBT)_2 = 15.5$°C

Total heat removed $= \frac{m_a(h_1 - h_2)}{\cancel{12N\phi}}$ kg/s

$= 3.75 \times (\cancel{43.40}\ 83 - 43.4)$

$= 148.5$ kJ/s $= 148.5$ kW

$= 148.5 \times 10^3$ J/s

$TR = \frac{148.5 \cancel{\times 10^3} \text{ kW}}{\cancel{210}\ 3.5} = \cancel{707.14}$

Capacity of coil. $= 42.42$ TR

Vapour removed $= (\omega_1 - \omega_2) \times m_a$

$$SHF = \frac{SH}{\text{Total heat}}$$

Vapour removed $= (0.02 - 0.0105) \times 225\ 22$

$= 2.14$ kg/min

$$SHF = \frac{SH}{\text{Total heat}} = 0.377$$

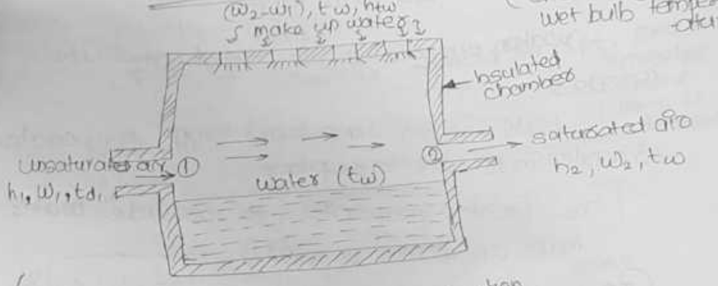
$SH = h_{A} - h_{\overset{A}{2}} =$ ~~43.4 - 83~~ $= 58 - 43.4$

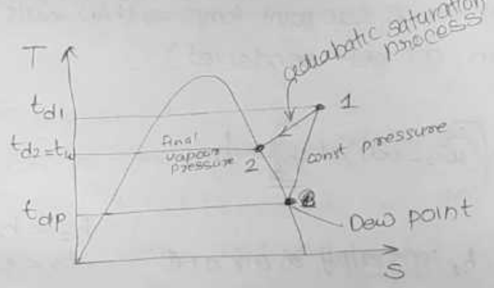
~~25~~ =

$$SHF = \frac{\cancel{25}\,14.6}{83 - 43.4} = 0.36$$

Adiabatic Saturation Temperature

(Thermodynamic wet bulb temperature)





(air - vapour pressure corresponding boiling point if temp < dew point temp water drops will form)

At Point 1, $T_1 >$ dew point temp

water evaporates & fills in air

(If water is at dew point temp then air is saturated with water)

Water bath temp > ~~at~~ dew point temp

② → Air fully water saturated.

Air temp at exit is temp of water bath T_2.

t_2 is called adiabatic saturation temp or thermodynamic wet bulb temp.

water evaporates, so final partial pressure increases.

Until water temp = air temp evaporation of water into air take place.

(water evaporates, to provide water make up water is used.)

(If water temp < dew point temp → then water vapour in air get condensed)

$$h_1 + \boxed{(\omega_2 - \omega_1) \times h_{fg_2}} = h_2$$

$h_2 < h_1$ since $t_2 < t_1$

h_1: enthalpy ~~of~~ air at ①

ω_2: specific humidity of satur. air

h_{fg}: enthalpy ~~of~~ supplied to air by water.

ω_1: specific humidity of unsaturated air.

h_{fg_2}: latent heat ~~of~~ at point 2

$$h_1 + \left[(\omega_2 - \omega_1)\, h_{fg_2}\right] = h_2$$

$$h_1 - \omega_1 h_{fg_2} = h_2 - \omega_2 h_{fg_2}$$

$$h_1 = h_2 - (\omega_2 - \omega_1) h_{fg_2}$$

h can't be replaced by C_pt. since there is enthalpy drop & temp is decrease.

$$\boxed{C_{pm} t_1 = C_{pm} t_2 - h_{fg_2}(\omega_2 - \omega_1)}$$

$$t_1 = t_2 - (h_{fg2}/C_{pm})(\omega_2 - \omega_1)$$

C_{pm} - humid specific heat

$$\boxed{t_2 = t_1 + (h_{fg2}/C_{pm})(\omega_2 - \omega_1)}$$

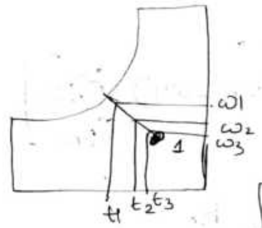
Significance of WBT

For a given WBT, for different RH values (inlet conditions) there are different DBT.

$$t_2 = f(t_1, \omega_1)$$

t_2 — WBT. t_1 — DBT ω_1 — RH or specific humidity.

so it's a thermodynamic property



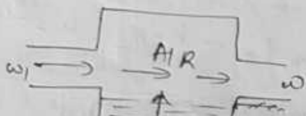
For diff combination of T_1 & ω_1 we get same WBT. Join. An inclined line will be obtained

For diff comb. of T_1 & ω_1 we get same WBT. So joining them we get a line.

For adiabatic sat temp

$h_2 = h_1 + (\omega_2 - \omega_1)\, h_f$

$\omega_2 > \omega_1$ since water vapour is added, humidity increases.



$\omega_2 > \omega_1$
specific humidity

ω_1 : ~~kg~~ kg of vapour / kg of air

$\omega_2 - \omega_1$ is the vapour content change

h_f : enthalpy of saturated liquid kJ/kgK

Water is evaporated using energy from the air, so i.e. temp of air decrease.

$t_2 < t_1$

$$t_2 = t_1 - \left[(\omega_2 - \omega_1)\frac{h_{g2}}{C_{pm}}\right]$$

$$h_1 + \left[(\omega_2 - \omega_1)\, h_f\right] = h_2$$

$$h_1 - \omega_1 h_{fw} = h_2 - \omega_2 h_{fw} \quad \text{——} (i)$$

$h_1 = h_{a1} + \omega_1 h_{s1}$

$h_2 = h_{a2} + \omega_2 h_{s2}$

h_{a1} = Enthalpy of 1 kg of ~~air~~ dry air @ DBT, t_{d1}

h_{s1} = Enthalpy of superheated vapour at t_{d1} per kg of vapour.

h_{a2} = Enthalpy of 1 kg of air @ WBT, t_w.

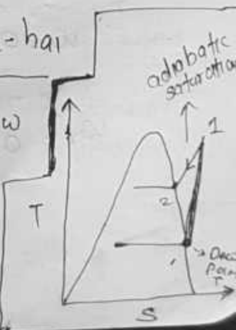
h_{s2} = Enthalpy of saturated vapour @ WBT, t_w per kg of vapour.

Now the equation (i) may be written as,

$$h_{a1} + \omega_1 h_{s1} - \omega_1 h_{fw} = h_{a2} + \omega_2 h_{s2} - \omega_2 h_{fw}.$$

$$\omega_1 (h_{s1} - h_{fw}) = \omega_2 (h_{s2} - h_{fw}) + h_{a2} - h_{a1}$$

$$\omega_1 = \frac{\omega_2 (h_{s2} - h_{fw}) + h_{a2} - h_{a1}}{h_{s1} - \cancel{h_{fa}}\, h_{fw}}$$



At ① → Dry air + moisture content

$x \longrightarrow x$

Q) Atmospheric air at 0.96 bar enters the adiabati saturated. The WBT is 20°C and DBT is 31°C during adiabatic saturation. Determine humidity ratio of entering air, vapour pressure & relative humidity at 31°C and dew point temperature.

here stand. atmos pressure is 0.96 bar

~~Or~~

Psychometric chart is at 1.013 bar

Temp of air at exit of ~~satura~~ adia sat = WBT.

$\omega_1 = ?$

$$\omega = 0.622 \frac{P_v}{P_a} = 0.622 \frac{P_v}{P_b - P_v}$$

$$\omega_2 = 0.622 \frac{P_{v2}}{P_b - P_{v2}}$$

② saturated . vapour line

P_{v2} : vapour pressure at 2 (saturated state

$T_2 = 20°C$

Page 44, table 39

$P_{v_2} = 0.02339$ bar (at 20°C)

P_b = barometer pressure = 0.96 bar

$$\omega_2 = \frac{0.622 \times 0.02339}{0.96 - 0.02339} = \underline{\underline{0.0155}}$$

$$\omega_1 = \frac{\omega_2 (h_{s2} - h_{fw}) + h_{a2} - h_{a1}}{h_{s1} - h_{fw}}$$

$h_{s2} = h_g$ at 20°C = 2538.1 kJ/kg.

h_{fw} = saturated water enthalpy at 20°C

= 83.96 kJ/kg.

air temp = wat
t

air ~~$h_{a2} = 1.005 \times 293$~~

~~= 294.465 kJ/kg~~.

~~$h_{a1} = 1.005 \times$~~

for air

$h = C_p T$

C_p = ~~1.005~~ 1.005

$\Delta h = m C_p \Delta T$

$\Delta T = 293 - 273$

0 i
dat

$h_{a2} = 1.005 (293 - 273)$

or

$= 1.005 \times 20$

$= 20.1$ kJ/kg.

$h_{a1} = 1.005 (31) = \underline{\underline{31.155}}$ kJ/kg

$h_{s1} = (h_{g1})_{31^\circ C} = 2558.1$ kJ/kg

$$\omega_1 = \frac{0.0155 \times (2538.1 - 83.96) + 20.1 - 31.155}{2558.1 - 83.96}$$

$= \underline{\underline{0.0109}}$ kg/kg dry air

$\omega_1 = 0.622 \dfrac{p_v}{p_a}$

$= 0.622 \dfrac{p_{v1}}{p_b - p_{v1}}$

$0.0109 = 0.622 \times \dfrac{p_{v1}}{0.96 - p_{v1}}$

$0.0175 = \dfrac{p_{v1}}{0.96 - p_{v1}}$

$0.0168 - 0.0175\, p_{v1} = p_{v1}$

$p_{v1} = \underline{\underline{0.0165}}$ bar.

Relative humidity $= \dfrac{p_{v1}}{p_{s1}}$

p_{s1} sat vapour pressure at $31^\circ C = 0.04496$

$\phi = \dfrac{0.0165}{0.04496} = \underline{\underline{0.367}} \simeq 36.5\%$

Dew point temp = temp corresponding to vapour pressure

$= 14.5^\circ C = \underline{\underline{14.5^\circ C}}$

1/11/18

Enthalpy of moist air

$h_a = C_{pa} t_d$

C_{pa} = specific heat of dry air

t_d = dry bulb temp.

enthalpy of watervapour associated with 1 kg of dry air $h_v = \omega_{\cancel{hg}}\, h_s$

ω = mass of water in 1 kg of dry air.

h_s = Enthalpy of water vapour / kg of dry air at dew point (t_{dp})

If moist air is superheated then enthalpy of water vapour,

$h_{wv} = \omega C_{ps} (t_d - t_{dp})$

C_{ps} = Sp Heat of superheated vapour = 1.9 kJ/kg

= 1.9 kJ/kgK

$t_d - t_{dp}$ = degree of superheat of the water vapour.

∴ Total enthalpy of superheated water vapour

$$h = C_{pa} t_d + \omega h_s + \omega C_{ps} (t_d - t_{dp})$$

$$= C_{pa} t_d + \omega (h_{f\,dp} + h_{fg\,dp} + C_{ps} (t_d - t_{dp})]$$

$$(\because h_s = h_{f\,dp} + h_{fg\,dp}$$

$$h = C_{pa} t_d + \omega [\overset{4.2\,t_{dp}}{\cancel{h_{f\,dp}}} + h_{fg\,dp} + C_{ps} (t_d - t_{dp})]$$

$$= C_{pa} t_d + \omega\, 4.2\, t_{dp} + \omega h_{fg\,d} + \omega C_{ps} t_d - \omega C_{ps} t_{dp}$$

$$= (C_{pa} + \omega C_{ps}) t_d + \omega [h_{fg\,dp} + t_{dp}(4.2 - C_{ps})]$$

$$= [C_{pa} + \omega C_{ps}] t_d + \omega [h_{fg\,d} + t_{dp}(4.2 - 1.9)]$$

$$= [C_{pa} + \omega C_{ps}] t_d + \omega [h_{fg\,d} + 2.3\, t_{dp}]$$

The term $[C_{pa} + \omega C_{ps}]$ is called humid specific heat (C_{pm})

In general, ~~$h = 1.02$~~

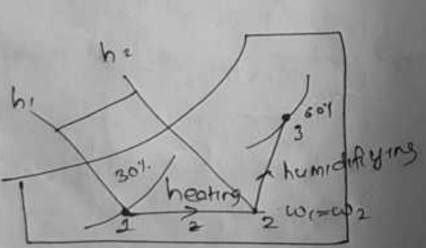
$$h = 1.2022\, t_d + \omega (h_{fg}\, dp + 2.3\, t_{dp}) \frac{kJ}{kg}$$

$h_{fg\,dp}$ = Latent heat of vaporisation of water corresponding to DPT

$$h = 1.005\, t_d + \omega [2500 + 1.9\, t_d]\ kJ/kg$$

An air conditioning system is taking outdoor air at 10 °C and 30 % Rh ~~at~~ a steady rate of 45 m^3/min. and to condition it to 25 °C and 60 % Rh. The outdoor air ~~is~~ first heated to 22 °C ~~in~~ the heating section and then humidified by the injection of hot steam in to the humidifier itself. Assuming the entire process takes place at a pressure of 1 bar. Determine the rate of heat supplied in the heating section. mass flow rate of ~~state~~ steam required in the humidifier itself.

P = 1 bar so we can't use psychometric chart
P < 1.013 bar.



Rate of heat supplied $= \mathring{m}_a (h_2 - h_1)$

~~ma~~ $V = 45$ m^3/min

$m_a = \frac{V}{v_{s_1}}$ v at inlet.

$h_{e_1} = 1.005\, t_d + w \times (2500 + 1.9\, t_d)$

t_d: drybulb temp. in °C

since it is difference.

$(t_d - 0) = 80$ use °C.

$h_1 = (1.005 \times 10) + \omega (2500 + 1.9 \times 10)$

$\omega = \frac{0.622\, P_v}{P_b - P_v}$

Relative humidity $= \frac{P_v}{P_s} = \phi$

$\phi_1 = 0.3 = \frac{P_{v_1}}{P_{s_1}}$

page 44
from properties of steam for psychometric application.

$P_{s_1} = P_s$ at 10°C $= 0.012276$.

$0.3 = \frac{P_{v_1}}{0.012276} \Rightarrow P_{v_1} = 3.6828 \times 10^{-3}$ bar

$\omega_1 = \frac{0.622 \times 0.0036828}{1 - 0.0036828} = 2.29 \times 10^{-3}$

$h_1 = (1.005 \times 10) + 0.00229 \times (2500 + 19)$

$= 15.818$ KJ/Kg

$h_2 = 1.005 t_{d2} + \omega_2 (2500 + 1.9\ t_d)$

$t_{d2} = 22°C$

~~$\omega_2 = 0.622$~~ $\omega_1 = \omega_2$ (since only heating)

so $\omega_2 = 2.29 \times 10^{-3}$

$h_2 = 1.005 \times 22 + 2.29 \times 10^{-3} (2500 + 1.9 \times 22)$

$= 27.93$ KJ/kg

$$\frac{m^3/min}{m^3/kg} = \frac{kg}{min}$$

$m_a = \frac{V}{V_{s1}}$

(V_g in table 39 is specific volume, V_{s1} of saturated steam (air) at enlet steam is not saturated.)

$PV = mRT$

$P_a v_a = RT$

(we find spec. vol of air)

1 bar = $P_v + P_{dair}$

$P_b = P_a + P_v = 1$ bar

$P_a = 1 - 0.003682$

$= 0.996318$

$v_a = \frac{287 \times 283}{0.996318 \times 10^5}$

$= 0.8152\ m^3/kg$.

So mass flow rate of air

$\dot{m}_a = \frac{V}{v_a} = \frac{45}{0.8152} = 55.2\ kg/min = 0.92\ \frac{kg}{s}$

∴ heat supplied $= 0.92 \times (27.93 - 15.818)$

$= 11.14$ KJ/s

W_3 = steam added / kg of dry air [ω = kg of [illegible] / kg of

$= \omega_3 - \omega_2$

$\omega_3 = \frac{0.622\ P_v}{P_b - P_v}$

$\phi = \frac{P_v}{P_s} = 0.6$ $\quad T_3 = 25°C$

$0.6 = \frac{P_v}{0.03169}$

$P_{v_3} = 0.019$

$\omega = \frac{0.622 \times 0.019}{1 - 0.019}$

$= 0.012$

$\omega_3 = 0.012$

Steam added / kg of dry air $= 0.012 - 0.00229$

$= 9.71 \times 10^{-3}$

∴ stead added $= 0.92 \times 9.71 \times 10^{-3}$

$= 8.9332 \times 10^{-3}$ kg/s.

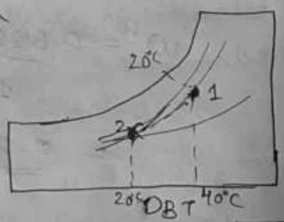
Q. The following data refers to airconditioning of a public wall. Outer door conditions = 40°C DBT, 20°C WBT. ~~Prefered~~ Required comfort condition 20°C DBT, 50% RH. Seating capacity of wall is 1000. Amount of outdoor air supplied is 0.3 m^3/min per person. If the required condition is achieved first by adiabatic humidifying and then cooling find,

(1) the capacity of the cooling coil and
(2) surface temp of the coil if the bypass factor is 0.25.

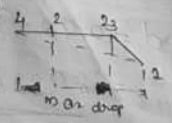
(2) The capacity of the humidifier and its efficiency

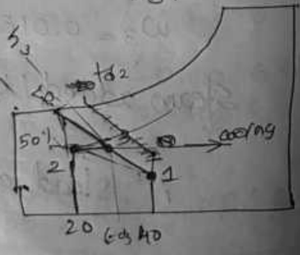
Cooling - horizontal line from right to left

adiabatic humidification - same line as WBT temp.



$\eta_H = \frac{\text{Actual}}{\text{Ideal}}$





$h_3 - h_2$ Cooling

$h_3 = 57$ kJ/kg
$h_2 = 38$ kJ/kg

~~V = 300 m^3/min~~
$V = 0.3 \times 1000$ m^3/min

Outdoor air supplied (Supply at / at point 1)

so $m_a = \frac{V}{v_{s1}}$

$v_{s1} = 0.896$ m^3/kg.

$m_a = \frac{\cancel{0.3}\ 300}{0.896} = \overset{334.8}{\cancel{0.3348}}$ kg/min

Capacity of cooling coil $= m_a(h_3 - h_2)$

$= \overset{334.8}{\cancel{0.3348}}(57 - 38)$

~~= 6.3612 kJ/min~~

$= 6361.607$ kJ/min $= 106.026$ kJ/s

$= 106.026$ kW

(TR) Capacity $= \frac{106.026}{3.5} = 30.3$ TR

BPF $= \frac{\text{Actual temp diff btw coil \& air}}{\text{max temp drop}}$ 3-2 cooling coil

$BPF = \frac{t_{d2} - t_{d4}}{t_{d3} - t_{d4}}$

Coil temp

$0.25 = \frac{~~38 - 20~~ \; 20 - t_{d4}}{38 - t_{d4}}$

~~coil temp, $t_{d4} = -34°C$~~.

$9.5 - 0.25 t_{d4} = 20 - t_{d4}$

$0.75 t_{d4} = 10.5$

coil temp. $\underline{t_{d4} = 14°C}$

(ii) Capacity of humidifier $= \underset{kg/s}{\dot{m}_a} \underset{kg/kg\ of}{(\omega_3 - \omega_2)}$

$\omega_3 = \overset{0.00745}{~~0.75~~}$ kg/kg of dry air

$\omega_2 = 0.0066$ kg/kg of dry air

humidifier - how much water it has given.

Capacity $= 334.8(0.00745 - 0.0066)$

$= \overset{0.30132}{~~0.21762~~}$ kg/min of water $= \underline{18.08 \frac{kg}{hr}}$

Efficiency of humidifier $= \eta = \frac{\text{actual drop}}{\text{ideal drop}}$

$= \frac{t_{d1} - t_{d3}}{t_{d1} - t_{d2}} = \frac{40 - 38}{40 - 20} = 0.1 = \underline{10\%}$

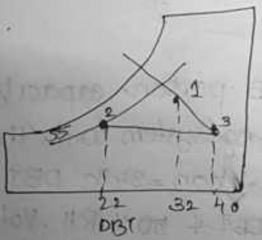
Q. A conference room of 60 seating capacity is to air condition of 22°C dry bulb temp and 55%. RH. The outdoor conditions are 32°C DBT and 22°C WBT. The quantity of air supplied is 0.5 m^3/min per person. The comfort condition are achieved first by chemical dehumidification and by cooling coil. Determine

(i) DBT of air at exit of humidifier

(ii) Capacity of dehumidifier.

(iii) Capacity of surface temp of cooling coil if the bypass factor is 0.3.

Ans.



(i) DBT at 3 = 40°C

(ii) $\omega_1 = 0.0124$

$\omega_3 = 0.009$

$V_1 = 0.5 \times 60 = 30\ m^3/min = 0.5\ m^3/s$

$V_{s1} = 0.881\ m^3/kg$.

$\therefore \dot{m} = \frac{V_1}{V_{s1}} = \frac{0.5}{0.881} = 0.5675$.

$\therefore$ Capacity $= \dot{m}(\omega_1 - \omega_3) = 0.5675(0.0124 - 0.009)$

$= \underline{7.09\ kg/hr}$.

(iii) Capacity of cooling coil $= \dot{m}_a(h_3 - h_2)$

$h_3 = 47$ kJ/kg

$h_2 = 64$ KJ/Kg

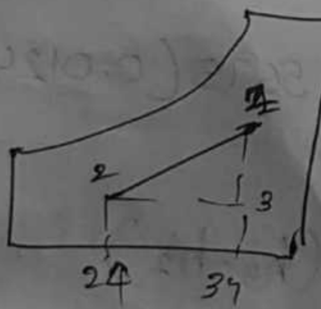
Capa. $= 10.09\,(h_3 - h_2) = 9.69$ kW $= 2.7$ TR

$$BPF = \frac{t_{d2} - t_{d4}}{t_{d3} - t_{d4}}$$

$$0.3 = \frac{22 - t_{d4}}{40 - t_{d4}} \Rightarrow t_{d4} = 14.286°C.$$

$$\text{Efficiency} = \frac{\text{Actual increase in DBT}}{\text{Ideal increase in DBT}}$$

Q. A small office hall of 25 person capacity is provided with air conditioning system with the following data. Outside condition = 34°C DBT & 28° WBT. Inside condition = 24°C DBT & 50% RH. Vol. of air supplied = 0.3 m³/min per person. Sensible heat load in the room is 125600 KJ/hr. Latent heat in the room 42000 KJ/hr. Find the sensible heat factor of the plant.



$S_H = \dot{m}_a (h_3 - h_2)$

$L_H = \dot{m}_a (h_1 - h_3)$

Total sensible heat = SH + sensible heat load

Total latent heat = LH + latent heat load

SHF = Total SH / (Total LH + Total SH).