① **Air conditioning**: Simultaneous control of Temp., humidity, purity, velocity of air.

* moist air contains 2 parts → Dry air (N_2, O_2, CO_2, Inert gases) ; → water vap. (in SH state, not as liq.) → T-s drawn for this portion.

∴ if atm pressure = P_t Then $P_t = P_v$ (Pr. of w.v.) + P_a (Pr. of dA)

② **psychometry**: branch of sci. which studies moist air.

1- Specific humidity or humidity Ratio ω = Ratio of mass of w.v. to mass of dA in a sample.

$$\omega = \frac{m_v}{m_a} = \frac{P_v \cdot V}{R_v \cdot T} \cdot \frac{R_a \cdot T}{P_a \cdot V} = \frac{P_v}{P_a} \cdot \frac{\bar{R}}{29} \cdot \frac{18}{\bar{R}}$$

$$\boxed{\omega = \frac{m_v}{m_a} = \left(\frac{P_v}{P_t - P_v}\right) \cdot 622 \text{ kg wv/kg dA}}$$

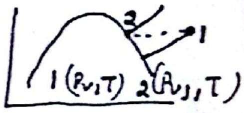
* All properties defined per kg dA → b/c this qty. remains constant.

2- Relative humidity RH or ϕ = Ratio of mass of vapour to the mass of vapour in the same volume and same Temp.

$$\phi = \frac{m_v}{m_{vs}} = \frac{P_v \cdot V}{R_v \cdot T} \cdot \frac{R_v \cdot T}{P_{vs} \cdot V}$$

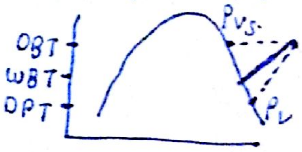
$$\boxed{\phi = \frac{m_v}{m_{vs}} = \frac{P_v}{P_{vs}}}$$

* for saturation pr. ↑ed from P_v to P_{vs} at same temp. DBT



3- DBT- normal temp. of air measured by an ordinary Thermometer.

4- WBT- Temp. measured by a thermometer whose bulb is covered by a wet cloth. water on cloth: In order to saturate the air, it absorbs heat from inner molecules for conversion to vapours. In this process Temp. falls. Incoming air: (i) Due to temp. diff. it transfers heat to water and it T falls (ii) It absorbs moisture from water. This continues till max. moisture absorbed by incoming air. The pressure ↑, T↓ for moist part of incoming air.



5- DPT- The temp. at which w.v. in air starts condensing at same pressure (P_v).

* unsaturated air DBT > WBT > DPT * Sat. air DBT = WBT = DPT

6- Degree of saturation -
$$\boxed{\frac{\omega}{\omega_s} = \frac{P_v}{P_{vs}}\left(\frac{P_t - P_{vs}}{P_t - P_v}\right) = R \cdot H \cdot \left(\frac{P_t - P_{vs}}{P_t - P_v}\right)}$$

7- Enthalpy of moist air $H = m_a \cdot h_a + m_v \cdot h_v$ at DBT $t°C$

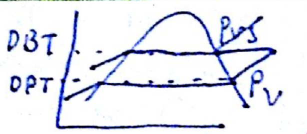
Dry air - Assumed 0 at 0°C ∴ $h_a = C_{pa} \cdot t$ w.v.: assumed 0 at 0°.

LH at 0° = 2500 KJ/kg $C_{p\,w.v.} = 1.88$ $h_v = 2500 + 1.88t$

$H = m_a \cdot C_{pa} \cdot t + m_v (2500 + 1.88t)$

$$\boxed{\frac{H}{m_a} = \text{Enthalpy/kg dA} = C_{pa} \cdot t + \omega(2500 + 1.88t)}$$

8- Saturation pressure corresponding to DBT - P_{vs}
" " " DPT - P_v

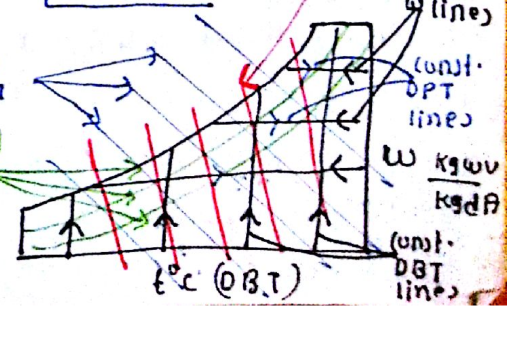


③ **PSYCHOMETRIC CHART**: Development [T-s, P_v-T, Opp. direcn, $P_v \simeq \omega$-T]

* P+F = C+2 1 + F = 2+2 F = 3

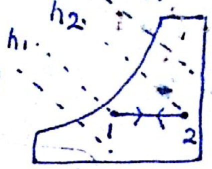
But in P. chart only 2 needed ∴ 3rd P_t = already fixed

* Why closed at ends? In AC. Temp. Range 20°C-45°C.



(4) PROCESSES : I sensible heating or cooling

- No phase change - at constant ω
- m_v, m_a remain const.



- if heat is received or lost due to transmission, it is supposed to have sensible heat load.

$Q = h_2 - h_1$ or $h_1 - h_2$

- Heating can be upto any T
- cooling limited till DPT

$Q = m_a (h_2 - h_1) = m_a \cdot c_{pa} \cdot (t_2 - t_1) + m_a \cdot \omega \cdot c_{pv} (t_2 - t_1)$

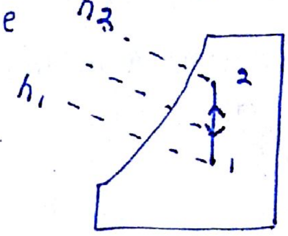
$Q = m_a (1.005 + 1.88\,\omega)\ t_2 - t_1 = m_a \cdot c_p (t_2 - t_1)$

where $c_p = 1.005 + 1.88\,\omega$ = c_p of moist air combined.

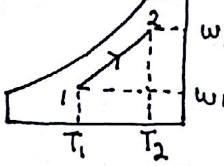
- if volume flow rate given in (cmm) i.e. m^3/min Then $\dot{m}_a = \rho \cdot \frac{(cmm)}{60}$.
- $\dot{m}_a = \left(\frac{cmm}{60}\right) \cdot \rho$ - $\rho = 1.2\ kg/m^3$ std. air (20°, 50% RH) - $c_p = 1.0216$
- $\dot{Q}$ = (cmm) .0204 at kW

II pure humidification or dehumidification - along DBT=C line

- $Q_L = m_a (h_2 - h_1)$ - change in moisture = $m_a (\omega_2 - \omega_1)$
  = change in moisture × $(h_{fg})_{0°} = m_a \cdot (\omega_2 - \omega_1) \cdot 2500$ {Taking $c_{p_1} = c_{p_2}$}
- if bldg. gains or lose moisture it is supposed to have LH load. $\dot{Q}_L = (cmm) \times 50 \times \Delta\omega$ kW



III Heating with humidification



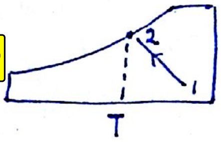
E.g. Steam spray in air [with h_v, m_v]

moisture balance - $\omega_2 = \omega_1 + \frac{m_v}{m_a}$

Energy " - $h_2 = h_1 + \frac{m_v}{m_a} \cdot h_v = h_1 + (\omega_2 - \omega_1) \cdot h_v$

Nozzle
steam
Air →

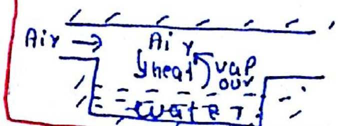
IV cooling with humidification



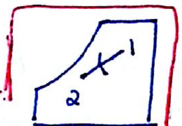
E.g. Air flown over water. it picks moisture from water and loses heat to water for its vaporisation.

T = Adiabatic saturation temp.

* For ~~Air Q=ma(h2-h1)~~ but for system Q=0
∴ 1→2 is const h line. also const WBT line



V cooling with Dehumidification



- In summer AC - cooling coil

VI Heating with "

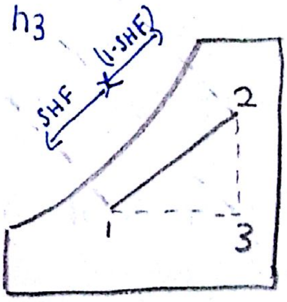
And DBT of air ↑s b/w L.H. of condn is released by silica while absorbing moisture.

E.g. Adiabatic chemical dehumidification
- Some chemicals like silica gel & Alumina used to absorb moisture from air, & hence ω ↓es.

(5) Total heat process :

Total heat = $m_a \cdot (h_2 - h_1)$

Sensible heat = $m_a \cdot (h_3 - h_1)$ LH = $m_a (h_2 - h_3)$



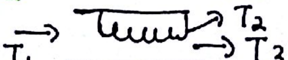
$$SHF = \frac{SH}{TH} = \frac{h_3 - h_1}{h_2 - h_1}$$

↑SHF (Dry climate, close Room)
↓SHF (humid " , open ")

$SH = m_a (1.005 + 1.88\ \omega_1 \text{ or } \omega_3)(t_3 - t_1)$ $LH = m_a (\omega_2 - \omega_3 \text{ or } \omega_1)(h_{fg})_{0°}$

(6) By pass factor: loss factor for a particular coil.

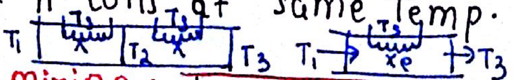
E.g. Heating coil



$BPF = \frac{T_2 - T_3}{T_2 - T_1}$ $[x \cdot T_1 + (1-x) T_2 = T_3]$.

* ↑ velocity → ↓ Time for HT → ↑BPF.

if n coils at same Temp. Then $X_e = x^n$



$xT_1 + (1-x)T_S = T_2$
$xT_2 + (1-x)T_S = T_3$
$x_eT_1 + (1-x_e)T_S = T_3$

(7) mixing : (i) Dry air mass balance $m_{a3} = m_{a1} + m_{a2}$

(ii) w.v. mass balance $m_{a3} \cdot \omega_3 = m_{a1} \cdot \omega_1 + m_{a2} \cdot \omega_2$

(iii) Energy balance $m_{a3} \cdot h_3 = m_{a1} \cdot h_1 + m_{a2} \cdot h_2$

- Adiabatic mixing
- At const. p.
- mixture state divides the line joining in inverse ratio of masses.

$m_{a3} t_3 = m_{a1} t_1 + m_{a2} t_2$
* By approximation.
- c_p all same for all 3 streams

$$\frac{m_{a1}}{m_{a2}} = \frac{t_2 - t_3}{t_3 - t_1} = \frac{h_2 - h_3}{h_3 - h_1} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1}$$

* In inverse Ratio of masses.

E.g. 15.2 1 part of moist air (1) is 2 parts of moist air (2) are mixed. ∴ Ratio of m_a/m_a needed m_{a1} = mass of dry air per unit mass of moist air. $m_{a1} = 1/(1+\omega_1)$ $m_{a2} = 2/(1+\omega_2)$.

Total heat process: $Q_s = (cmm)\cdot 0.0204\,\Delta t$ kW

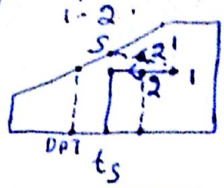
$Q_L = 50\,(cmm)\,\Delta\omega$ kW   Total $\dot{Q} = 0.02\,(cmm)\,\Delta h$
$= 0.02\,(cmm)(h_2-h_1)$

$SHF = \dfrac{0.0204\,\Delta t}{0.0204\,\Delta t + 50\,\Delta\omega} = \dfrac{0.0204\,\Delta t}{0.02\,\Delta h}$ $= cmm\,(0.0204\,\Delta t + 50\,\Delta\omega)$ kW

* process line 1-2 is SHF line or process or condition line.

* $SHF = \dfrac{1}{1+2451\cdot\frac{\Delta\omega}{\Delta t}} = \dfrac{1}{1+\tan\theta\cdot 2451}$ * $\tan\theta = \dfrac{\Delta\omega}{\Delta t} = \dfrac{1}{2451}\left[\dfrac{1}{SHF}-1\right]$ * tanθ is the slope of line 1-2.

Cooling coil: 1- $t_s >$ DPT of incoming air ⇒ Dry coil / Sprayed "



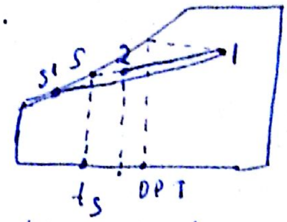
Dry coil: cooling along DPT line * upto t_2 not t_s b/c of BPF

spray coil: {cool + humid} {Textile mills} Along 1-S upto 1-2'.

2- $t_s <$ DPT of incoming air Then t_s called t_{ADP} of coil.

- process 1-2 → not 1-S b/c of BPF. * At t_{ADP} RH = 100%.

* limit of this process: 1-S' where tangent to P-curve

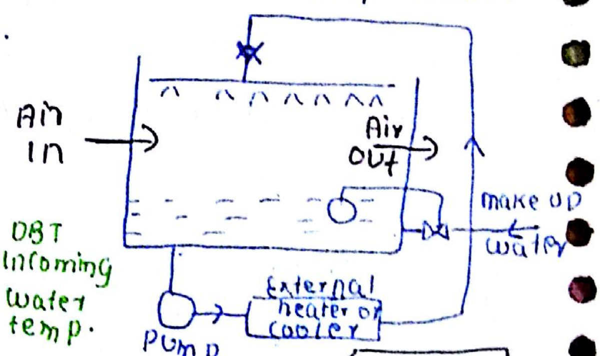


Simple heating coils: $t_s >$ DBT of incoming air

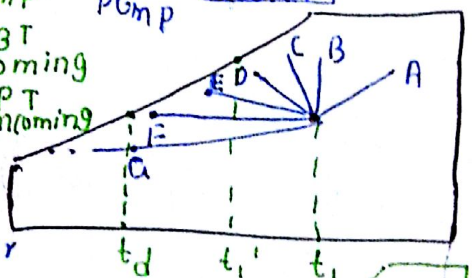
(i) Hot water or steam coils (ii) Furnace gases coil (iii) Finned electric strip heaters.

AIR WASHER: - Flow of air thru spray of water.

Process	Temp.	what done to water	Process
Heat + humid	$t_s > t_1$	Ext. heated	A
humid	$t_s = t_1$	"	B
cool + humid	$t_1' < t_s < t_1$	" (∵ Δh air +ve)	C
Adiabatic saturation	$t_1' = t_s$	Pumped circulation w/o Ext. heat or cool	D
cool + humid	$t_d < t_s < t_1'$	Ext. cooled (Δh air -ve)	E
cooling	$t_s = t_d$	"	F
cool + dehumid	$t_s < t_d$		G



t_1 = DBT incoming
t_s = water temp.
t_1' = WBT incoming
t_d = DPT of incoming



Hygroscopic soln in Air washer - ∵ They exert lower vapour pressures as compared to the water at same temp. Thus more effective dehumidification.

1-A water
1-C H. soln

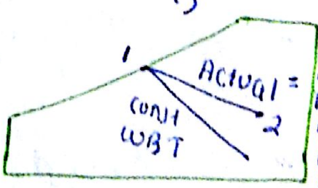
Adiabatic Dehumidifier: - principle of adsorption.

- Adsorbent absorps water from passing air.
- w.v. condn Releases heat which ↑es Adsorbent & air temp. ∴ supply sen. heat to heat air
- Addnal heat Release = heat of adsorption

1-2 : Heat due to condn of moisture = $m_a\cdot\Delta\omega\cdot h_{fg}$ (→2500)

Heat of Adsorption = $m_a\cdot\Delta\omega\cdot h_{ads}$

Rise in Air temp. = (i) + (ii) / $m_a\cdot 1.0216$

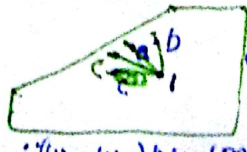


= Due to Adsorbent heat Release.

E.g. 15.5.

Water Injection: water Evaporates (heat comes from enthalpy of air) & condn of air changes.

- if water Evaporated = water Injected. $\omega_2-\omega_1 = \dfrac{m_v}{m_a}$
- $h_2 = h_1 + \dfrac{m_v}{m_a}\cdot h_f$ h_f = Enthalpy of liq. water



* Evaporative cooling becoming popular: ⇒ ↓es water loss / Industrial control of / simple & cost. ∵ $(\omega_2-\omega_1)h_f$ small / supply spaces ∴ process anyway close to WBT line

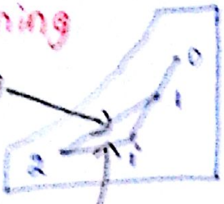
Air conditioning: - Final process b/w supply condn & inside condn

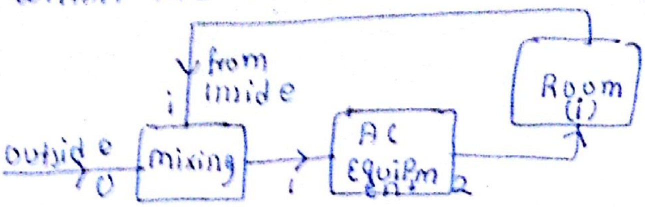
b/w s & i $RSH = (cmm)\times 0.0204\times\Delta t$ $RLH = (cmm)\times 50\times\Delta\omega$

⑧ **ventilation air** : - Amt. of fresh air supplied from outside.
- Recirculated air from within the Room.

summer air conditioning

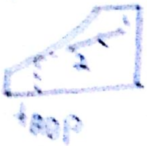
* Grand SHF = line Joining inlet, outlet of AC Equipment





* Room SHF = line Joining supply condn with room condn.
* supply pt = intersection of GSHF and RSHF.

⑨ **ADP** :



$$BPF = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}}$$

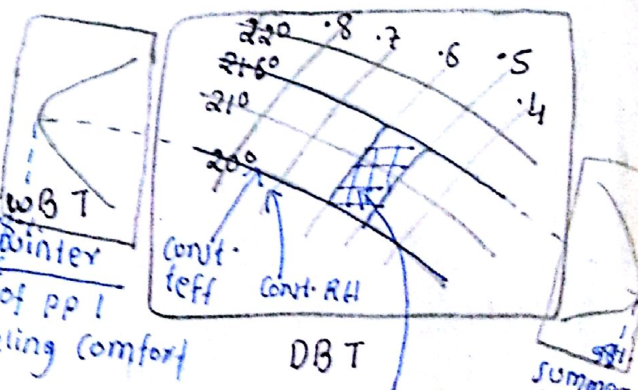
Temp. at which cooling & dehumidification line cuts sat. curve

⑩ **Effective Temp. (ET)** : Parameters governing comfort : 1- Air Temp. 2- humidity 3- air velocity } Thus problem of measuring comfort by a single parameter which combines all 3

ET - A single parameter used as an index of comfort.
That temp. of saturated (100% RH) air at which subject feels same level of comfort as experienced in actual unsaturated Env.
Difficulty in generating universal chart b/c depends on activity, clothing.

⑪ **Comfort chart** : Developed by ASHRAE by study/Experiments to have corelation b/w age, sex, comfort index (Y), humidity, temp. etc.

comfort in ϕ 50% - 60% t_{eff} → 21.5° summer → 20° winter

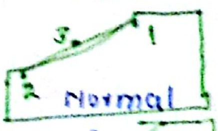


Humid specific heat : $h = C_{p_a} \cdot t + \omega \{ C_{p_v} \cdot t + (h_{fg})_0 \}$

$h = C_p \cdot t + \omega (h_{fg})_0$ ∵ ω↓ ∴ C_p = 1.0216 approx

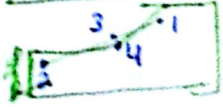
$C_p = C_{p_a} + \omega \cdot C_{p_v} = 1.005 + \omega\, 1.88$ KJ/kgdA-K
↳ specific heat of moist air (1+ω) kg per kg of DA

if mixing with condensation



Actual = 4

$$\omega_3 = \frac{m_{a_1}\omega_1 + m_{a_2}\omega_2}{m_{a_1} + m_{a_2}}$$



$t_4 > t_3$ (∵ heat released by LH of condensate)

$$h_3 = \frac{m_{a_1} h_1 + m_{a_2} h_2}{m_{a_1} + m_{a_2}} \qquad h_4 = h_3 - \omega_c \cdot h_{f4}$$

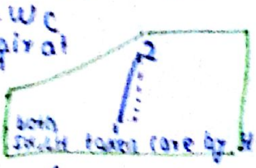
$\omega_4 = \omega_3 - \omega_c$

? org. of cold air + warm air at RH

h_{f4} = Enthalpy of condensate at temp. t_4 of the mixture
$(RH)_4 = 100\%$
ω_c = mass of condensate per unit mass of mixture.

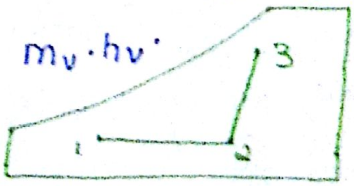
Steam injection :

m_v = mass of steam supplied



h_v = Enthalpy of steam $(\omega_2 - \omega_1) \cdot m_a = m_v$ $m_a (h_2 - h_1) = m_v \cdot h_v$
$t_2 - t_1 \simeq 0°$ * usually steam injected into outside air which then conditioned in tex. mills

E.g. 15.4 Steam injection + sensible heating

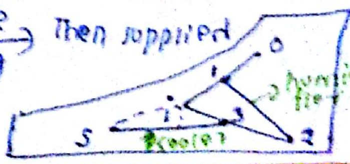


$\omega_3 = \omega_2 = \omega_1 + \frac{m_v}{m_a}$ $m_a \cdot (h_3 - h_1) = SH + m_v \cdot h_v$

E.g. 15.5 inside air + outside air → Adsorption dehumidifier → + more inside air → sensible cooling → Then supplied

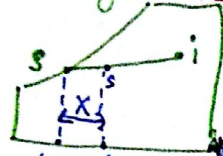
$$\dot{m}_{a_1} = \frac{RSH}{(1.005 + \omega\, 1.88)(t_i - t_s)}$$

$$\omega_s = \omega_i - \frac{RLH}{\dot{m}_{a_1} \times (h_{fg})_0}$$



Summer Air conditioning: usually cool + dehumd < cooling coil $t_s < DPT_{Air}$; Air washer chilled spray $t_s < t_d$

(i) only Recirculated air:



* i-S : locus of all possible supply state
* extremities for supply condn < i ∞ air ; S min. air

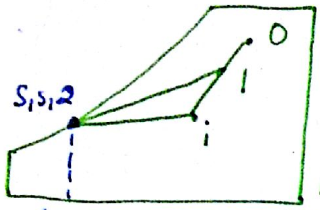
$$(cmm)_{s\,min} = \frac{RSH}{\cdot 0204(t_i - t_{ADP})} = \frac{RLH}{50(w_i - w_{ADP})} = \frac{RTH}{\cdot 02(h_i - h_{ADP})}$$

* with BPF X : use the Δt → ↑ air supply

$t_i - t_s$ = dehumidified Rise $(cmm)_d$: dehumidified air Qty.

$$(cmm)_{d=s} = \frac{RSH}{\cdot 0204(t_i - t_{ADP})(1-X)}$$

(ii) ventilation air + zero BPF:



- Amt. Equivalent to fresh air inducted is ejected out.
- if no ven. air : Then load = RSH + RLH b/w i-S
- when ven. air : Then addnal ven. load on apparatus
- Equivalent the change of condn of ven. air b/w 0 → i

GSHF line = condn line for apparatus 1-2
RSHF line = condn line for Room i-2

* Here coil ADP → Room ADP same

b/w 1-2 in apparatus
$\dot{Q} = \dot{m}_{as}(h_1 - h_2) = (\dot{m}_{ai} h_i + \dot{m}_{ao} h_o) - \dot{m}_{as} \cdot h_2$
$\dot{Q} = (\dot{m}_{as} - \dot{m}_{ao}) h_i + \dot{m}_{ao} h_o - \dot{m}_{as} h_2$
$\dot{Q} = \underbrace{\dot{m}_{as}(h_i - h_2)}_{\text{Room load}} + \underbrace{\dot{m}_{ao}(h_o - h_i)}_{\text{ven. load}}$

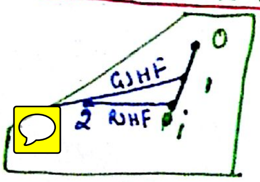
OASH = $\dot{m}_{ao}$ ·0204 $(cmm)_o (t_o - t_i)$
OALH = 50 $(cmm)_o (w_o - w_i)$
OATH = ·02 $(cmm)_o (h_o - h_i)$

Total Equipment load: (RSH + OASH) + (RLH + OALH) = TSH + TLH

$$GSHF = \frac{TSH}{TSH + TLH}$$

E.g. 15.6 Total cooling load = $\dot{m}_{as}(h_1 - h_2)$
Room heat gain = $\dot{m}_s(h_i - h_2)$

(iii) V. air + X BPF:



Winter AC: less load b/c (i) SH losses partially compensated by solar gain & occupancy heat gain, light etc. (ii) LH gains by occupancy.

- pre dominantly SH design & not LH design : LH gain by occupancy always present. SH gains like solar may or may not be present.
- Generally heating & humidyfying using:

(i) pre heating (steam or hot water in coil) → Adiabatic satu → Reheating   1-2-3-S

(ii) Heat + humid (by Air washer ext. heat + pump cir) → Reheat.   1-3-S