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UNIVERSITÀ DEGLI STUDI DI GENOVA

Master Degree in Innovative Technologies in Energy Efficient Buildings for Russian & Armenian Universities and Stakeholders (MARUEEB)

Junior Intensive Course
ENERGY EFFICIENT BUILDINGS

Thermodynamics of moist air and air conditioning systems

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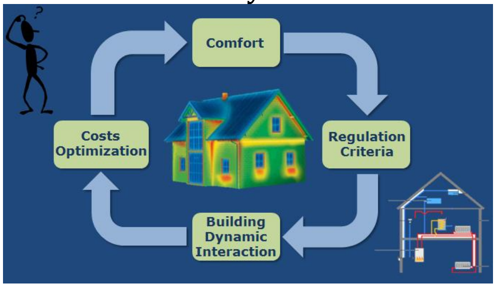
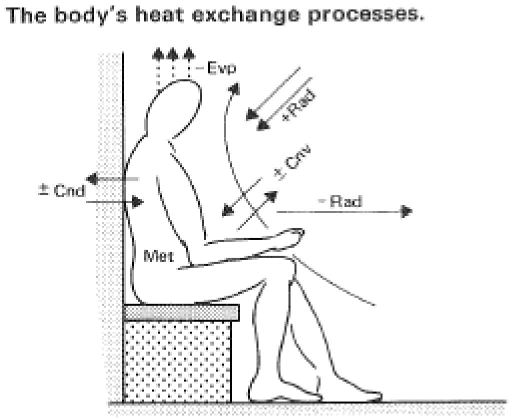


Genoa, November 2016

The body wellness and comfort parameters

We define “environmental conditioning”, the determination of thermo hygrometric conditions, the control and the production of air inside buildings. These conditions must be suitable for the intended use of that environment. Each human body produces heat (metabolic =Met), and exchanges it toward environment through the phenomena of radiation, evaporation, convection and conduction with cold bodies.

figure 1. The body exchange processes



The equation of stationary balance becomes

Met- evaporation ±Rad ±Conv±Cond =0 **(2.1)**

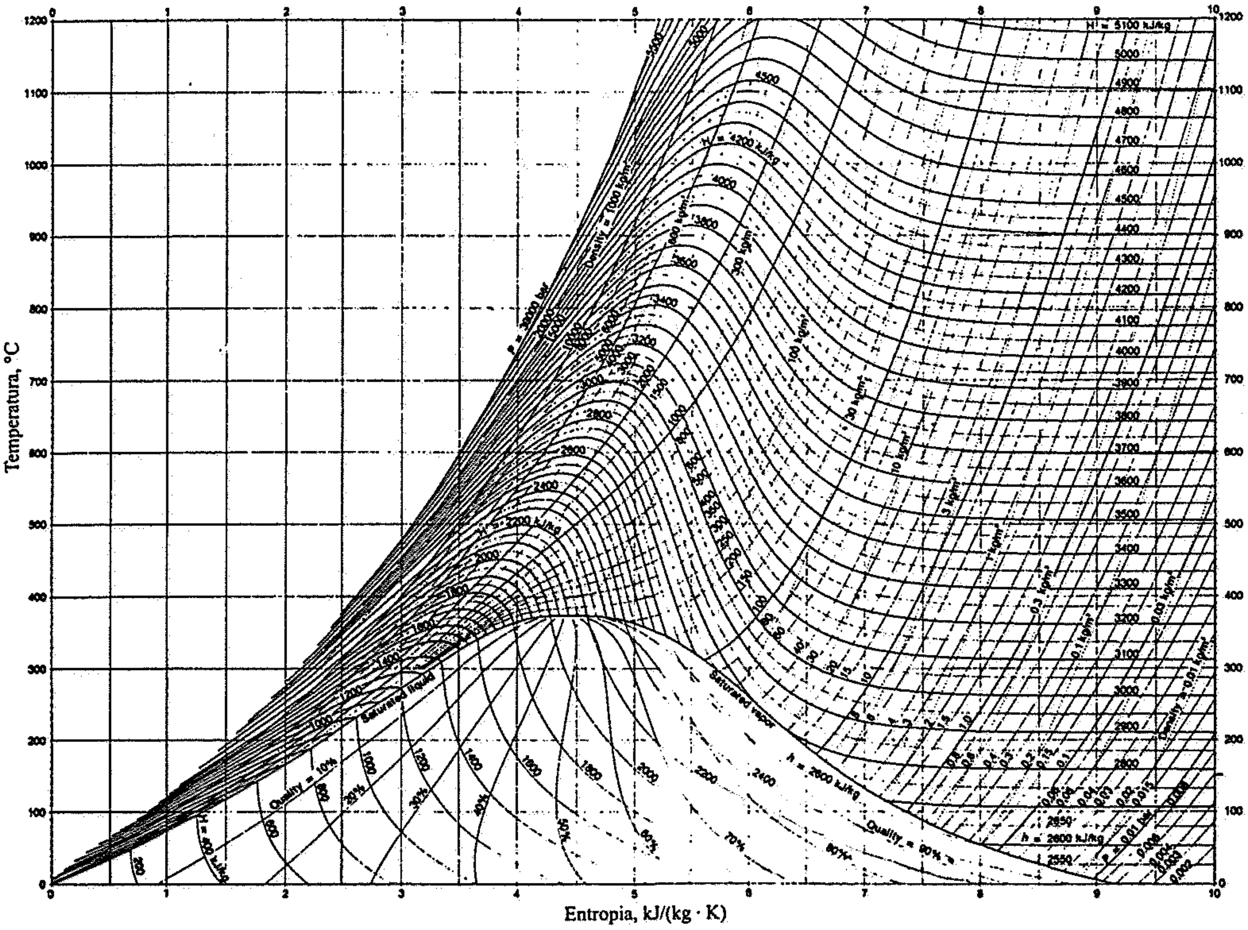
that’s the same with proper arithmetical signs Q_{tot} =Met

$\dot{Q}_{tot} = \dot{Q}_{irr} + \dot{Q}_{evap} + \dot{Q}_{conv} + \dot{Q}_{cond}$, (2.2)

where $\dot{Q}_{cond}$ $\dot{Q}_{conv}$ and $\dot{Q}_{irr}$ depends on ΔT while

$\dot{Q}_{evap} = h_{evap}\, r\, A_{body}\, (p_s(T_{body}) - p_v)$. (2.3)

fig.2 Entropy-Temperatrue Diagram (T-s) for water



In this expression “h_{evap}” is the *mass transfer coefficient* [kg(vapor)/m^2 Pa s], that is the mass that evaporates in the unit time, divided by the unit area, the partial difference in pressure and time. In the case of forced convection, this coefficient is higher than in the case of natural convection (in fact a ventilated body becomes cold faster). It is similar to the coefficient of convective heat transfer. The meaning of “r” is the enthalpy variation of the change phase from liquid to vapor, [kJ/kg] and is named “*enthalpy of vaporization”* or “*enthalpy of change of phase”*.

“A_{body}” indicates the surface of the body [m^2],

it is useful the Du Bois formulation for the determination of A_{body}

$$A_{body}= 0.202\ m^{0.425} h^{0.725} \qquad (2.4)$$

Where “m” [kg] is the mass and “h” [m] is the height of the body.

$p_s(T_{body})$ is the saturation pressure and “p_v” the partial pressure of water vapor dissolved in air (see Dalton's law).

If temperature is high, but the atmosphere is dry, the terms $\dot{Q}_{conv}$ and $\dot{Q}_{irr}$ are small, while $\dot{Q}_{evap}$ is high (physiologically a room warm and dry is better tolerated than another hot and humid).

Main thermo dynamical air properties

Moist air is composed of

- **dry air** (mainly a mixture of N_2(78%), O_2 (21%) and small amounts of pollutants such as CO_2, CO, NO_x) with pressure $p_{d.a}$
- **water vapor** with pressure p_v.

The **hygrometric content [w]** is also called **hygrometric degree** or **absolute humidity** and it can be determined as the ratio between the mass of vapor and the mass of dry air. It is also valuable as the ratio between the partial vapor pressure and the dry air pressure.

$$w = \frac{m_{vap}}{m_{d.a.}} = \frac{p_v}{p_{d.a.}} \qquad (2.5)$$

The former definition is not an operative formula so it is possible to demonstrate that the absolute humidity can be also (better) determined using the ratio between the two pressures.

From Dalton's law

$p_{tot} = p_{atm} = p_{d.a..} + p_v$ (2.6)

$$p_{atm} \approx 1\ bar \quad p_{vap} \approx 100\ Pa \cong 10^{-3}\ bar \rightarrow w \approx 10 \frac{g_{vap}}{kg_{a.s.}} \qquad (2.7)$$

Case study 1

For example the amount of water vapor in a room of V = 300 m^3 and m_{air}=360 kg (with ρ_{air} = 1.2 kg/m^3) is equal to m_{vap} = 3.6 kg , that is about 3.6 liters of water.

Another important parameter is **temperature [T]:** it is always considered uniform in the mixture inside the environment volume. It is important to note that moist air is not a simple thermodynamic fluid: in fact the specification of the two usual independent values (T and p) is no more sufficient to define the thermo dynamical state. It should be also mentioned that “w” provides an indication of the weight composition of the mixture indicating the parts of vapor in ratio with dry air.

In the case of a closed system containing dry air, but in which water can be nebulized, we analyze the *humidification* process: the value of m_{vap} grows, but exists a saturation value, beyond which there is a condensation of particles of water, which no longer pass from liquid to vapor. This value is indicated by the *saturation mass* (m_s). The capacity of air to *absorb* water changes with pressure: it decreases with the increase in pressure and vice versa.

If we set the working total pressure p (generally linked to the altitude), the saturation mass depends only on temperature.

$M_s = M_s(T)$. (2.8)

Relative humidity.

The relative humidity is the ratio between the mass of vapor and the maximum mass of vapor that can be adsorbed by dry air:

$$i = \frac{M_{vap}}{M_s} \quad \in [0,1] \qquad (2.9)$$

As written above it can be also related to the ratio between the partial vapor pressure (p_v) and the saturation pressure (p_s):

$$i = \frac{p_v}{p_s} \qquad (2.10)$$

The physiological values of comfort for the relative humidity “i” depend firstly on the season as shown in diagrams of comfort made by the A.S.H.R.A.E. (American Society of Heating, Refrigerating and Air-Conditioning Engineers) (Fig.3)

fig.3. Ashrae comfort seasonal areas

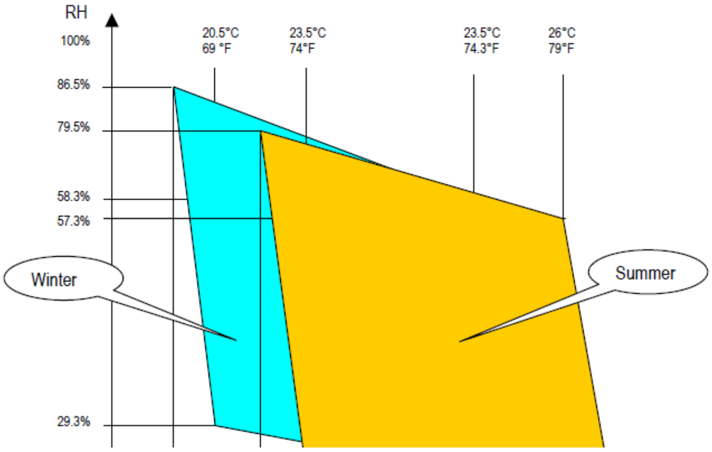


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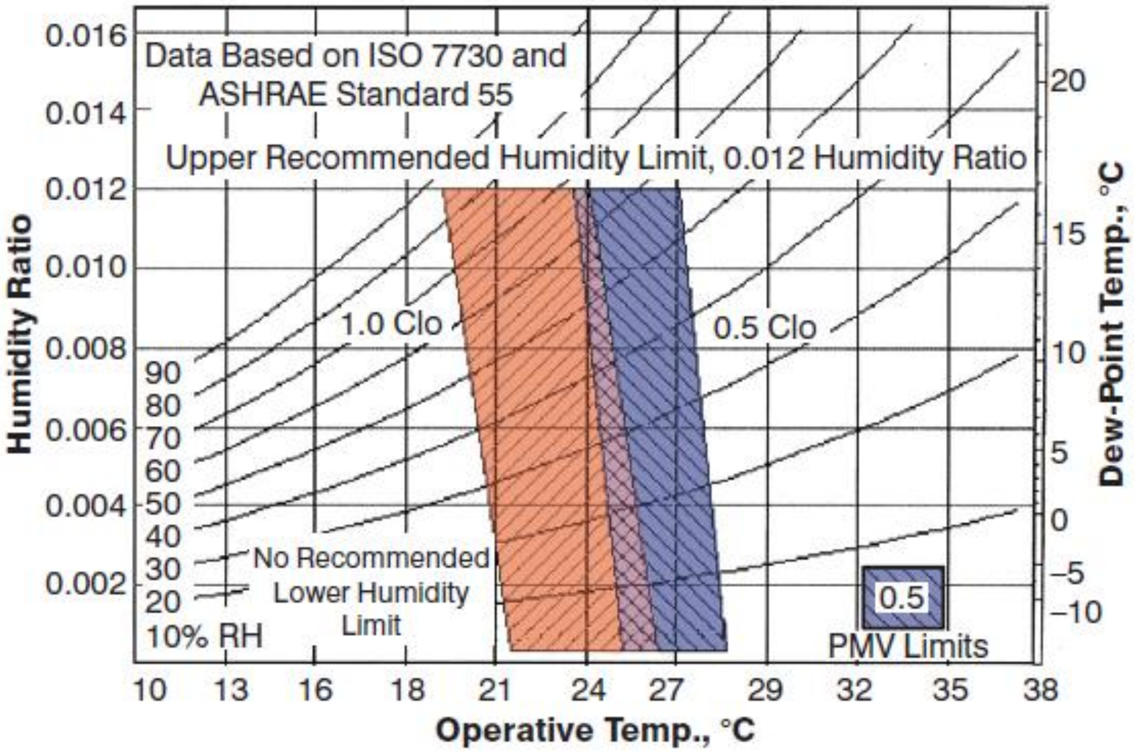
Fig 4 comfort are



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The nd each corr e entire volu

v_{mix} ⁇ $\frac{\cdot}{M_{da}}$ $\frac{\cdots mix}{kg_{da}}$

(2.14)

$$w = \left[\frac{R_{1\,v}}{R_{1\,as}}\right]^{-1}\cdot\frac{p_v}{p_{as}} = \left[\frac{R_{1\,v}}{R_{1\,as}}\right]^{-1}\cdot\frac{p_v}{p_{tot}-p_v} \qquad \frac{R_{1\,v}}{R_{1\,as}} = \frac{R/mm_v}{R/mm_{as\ apparente}} = \frac{mm_{as\ apparente}}{mm_v}$$

$$mm_{as\ apparente} = 29 \quad mm_v = 18 \quad \Rightarrow \quad \left[\frac{R_{1\,v}}{R_{1\,as}}\right]^{-1} = \frac{18}{29} = 0.622 \tag{2.15}$$

$$\Leftrightarrow \quad w = 0.622\cdot\frac{p_v}{p_{tot}-p_v}$$

It can be shown also that (2.10) $i = \frac{p_v}{p_s}$ where p_v is the partial pressure of water vapor and p_s is the partial pressure that vapor would have if the mixture were at *saturation*.

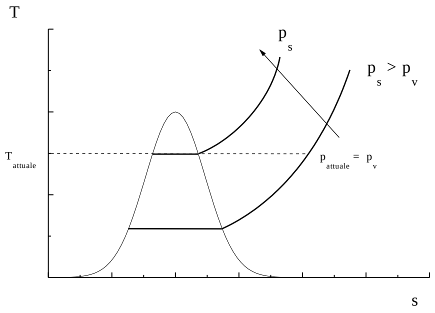

fig 5 – different pressures of water vapor

Consequently follows:

$$w = 0.622 \cdot \frac{i \cdot p_s(T)}{p_{tot} - i \cdot p_s(T)} \qquad (2.16)$$

For a perfect gas a relation exists among three variables (p,v,T), but the independent ones are only two; for humid air the correlation is among four variables (w, p_s, p_{tot} and T), but the independent ones are only three, since p_s is linked to T in a biunique way.

The approach to study moist *air* is to use all the energy equations considered above, with two independent flows $\dot{m}_v$ and $\dot{m}_{da}$ and using the additive characteristic of extensive properties. Actually there is an easier way to lead the discussion. Most of the systems used in air conditioning applications are *open* one (that is with flow), therefore, the most significant energetic contribution consists of *enthalpy*. Furthermore, to study the *"mixture"* system, in other words to determine its specific properties, the mass *kg of dry air* is the reference, since in air conditioning, the amount of dry air is an invariant. The humidity content, that is the quantity of vapor, is on the contrary a controllable quantity from the plant, since it is easier to change the latter, rather than the amount of dry air.

The enthalpy of the mixture is

$$H_{mix} = H_{vap} + H_{as} = M_v \cdot h_v + M_{as} \cdot h_{as} \quad \Leftrightarrow \quad h^o = \frac{H_{mix}}{M_{as}} = \frac{M_v}{M_{as}} \cdot h_v + h_{as} \qquad (2.17)$$

in an equivalent manner

$$h^o = w \cdot h_v + h_{as} \; [J / kg_{da}] \qquad (2.18)$$

is defined as the total *specific enthalpy of humid air relative to kg of dry air*. The relevant quantities in systems are in fact the enthalpy changes and it is therefore necessary to define quantitatively h°.

So it is assumed:

$h_v = 0$ in correspondence of the triple point of water - liquid phase.

$h_{da} = 0$ for T = 0° C and p_{tot} = 1 bar (in fact, in the case of a perfect gas, being h = h(T) the reference to pressure p_{tot} is not significant). $h_{da} = c_{p\,da}\,(t - t_o) = c_{pda}\,t$, with t_o = 0° C.

$c_{p\,da}$ = 1005 J / kg K	constant
$c_{p\,v}$ = 1900 J / kg K	constant
r = 2500 kJ / kg	latent heat of vaporization at the triple point of water
h_v = 2500 kJ / kg + 1.9 t	where t is the temperature in °C, being $t_o \approx 0$.

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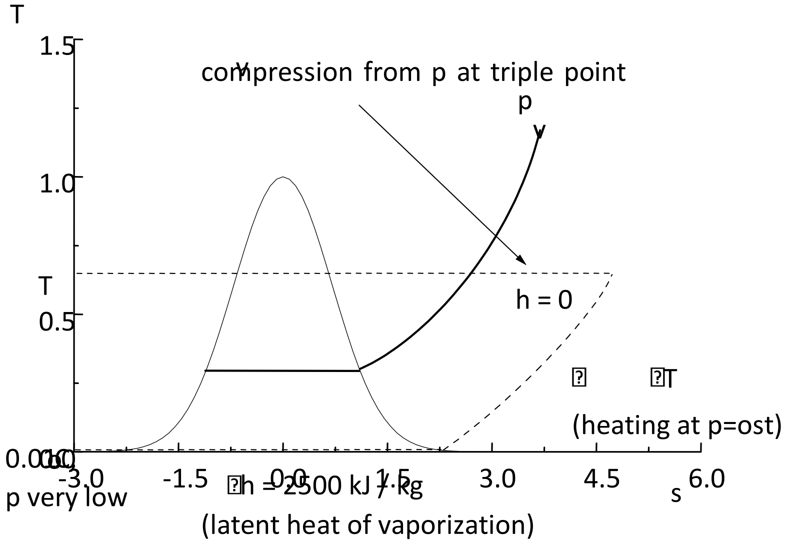


fig 6- reference values for water vapour in moist air

So enthalpy of humid air is:

$$h° = 1.005\, t + (2500 + 1.9\, t)\, w \; [\, kJ / kg \,] \qquad (2.19)$$

with

$$w = 0.622 \cdot \frac{i \cdot p_s(T)}{p_{tot} - i \cdot p_s(T)} \qquad (2.20)$$

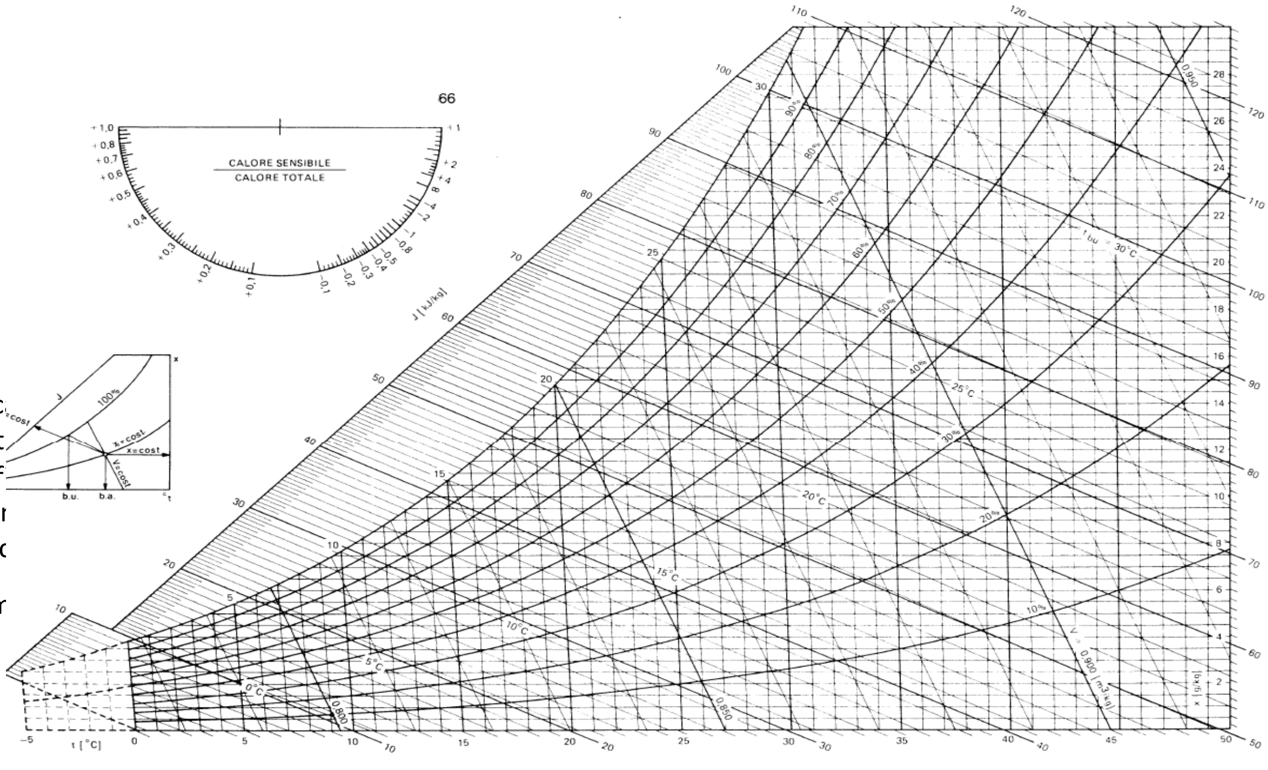
and $p_s = p_s(T)$ is known in tabular or analytical form.

For water it is also possible to use this polynomial formula (with 5% error):

$p_s(T)=1{,}35*(0{,}0001*T^3+0{,}013*T^2+0{,}3336*T+4{,}5111)$ (p [mbar]; T [°C] $\in[-10;+40°C]$. (2.21)

The relations listed above can be expressed in analytical or graph form according to altitude (determining p_{tot} read on the barometer). These allow the resolution of any problem about the humid air conditioning. If we keep p_{tot} constant, is possible to build, in a diagram (diagram ASHRAE), the value of w [g_v / kg_{as}] as a function of temperature [°C]: in fact once we have p_{tot} fixed, the humid air reduces to a system of only two variables. The relation between them is shown in fig 7:

fig 7 Ashrae (Mollier) diagram



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quote 0: $w = 9\ g_v / kg_{da}$ $v = 0.84\ m^3 / kg_{da}$ $h_0 = 42\ kJ / kg_{da}$
quote 2250: $w = 12\ g_v / kg_{da}$ $v = 1.11\ m^3 / kg_{da}$ $h_0 = 50\ kJ / kg_{da}$

The lower is the pressure p_{tot} the higher can be the quantity of water caught by air and in this way enthalpy increases.

Moist air transformations and processes

We consider two types of energy exchanges :

1. *sensible*: the processes of *sensible heating / cooling* in which there is an evident variation of ΔT and $\Delta w = 0$.
2. *latent*: exchanges in which *apparently* there is no ΔT but an evident change in Δw shows the contribution of the latent heat of vaporization.

The properties of a mixture are the weighted average of the component properties: the presence of vapor, regarding to the material property, is not really significant (R_1 varies slightly). However, the small vapor content is fundamental in energy exchanges, since vapor is able to carry around all the big amount of energy coming from vaporization. In fig 8 we show the enthalpy variation of moist air in heating transformation.

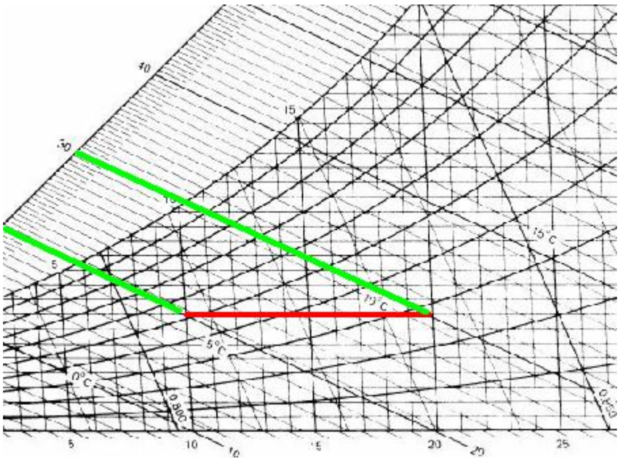


fig 8 variation of enthalpy in heating

Case study 2

Consider a tube in which a certain flow of dry air runs as shown in fig 9

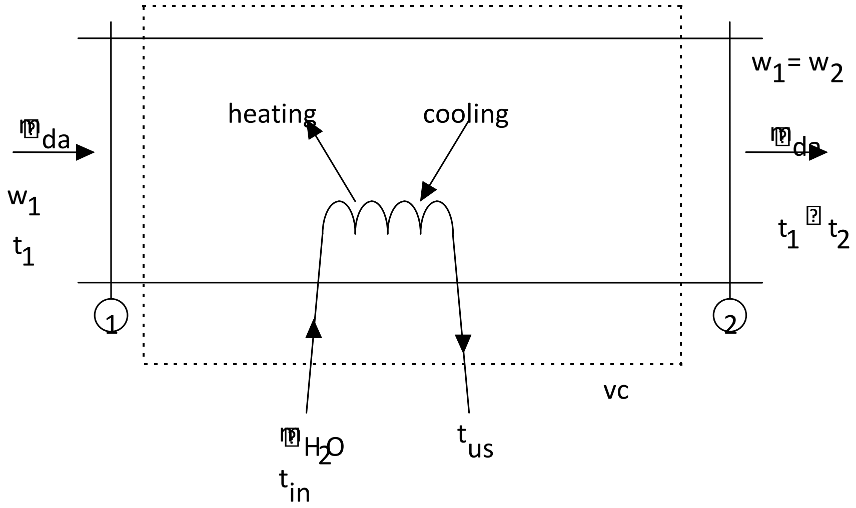


figure 9 Case study 2

$\dot{m}_{as}$ is an invariant of the system, since there are no leaks. This process can be made by letting air flow in a coil where hot (or cold) water is used in a serpentine to heat (or cool) air.

Let's consider the case of heating $t_2 > t_1$. The energy balance is as follows:

$$\dot{Q} = \dot{m}_{as} \cdot \left(h^o{}_2 - h^o{}_1 \right) \qquad (2.22)$$

Since the system is stationary and globally adiabatic, we can write a heat balance of air + water system, including in the volume control both air and water flow (two inputs and two outputs, without mixing):

$$0 = \dot{m}_{as} \cdot h^o{}_1 + \dot{m}_{H_2O} \cdot h_{in} - \dot{m}_{as} \cdot h^o{}_2 - \dot{m}_{H_2O} \cdot h_{us} \qquad (2.23)$$

(In this case) w is constant, so enthalpy changes can be determined using the diagram, or through a linear formula:

$$\Delta h \cong c_{p\ H_2O} \cdot \Delta T \qquad (2.24)$$

from which we can write:

$$\dot{m}_{as} \cdot \left(h^o{}_2 - h^o{}_1 \right) = \dot{m}_{H_2O} \cdot c_{p\ H_2O} \cdot \left(t_{in} - t_{us} \right) \quad \Rightarrow \quad \frac{\dot{m}_{H_2O}}{\dot{m}_{as}} = \frac{h^o{}_2 - h^o{}_1}{c_{p\ H_2O} \cdot \left(t_{in} - t_{us} \right)} .A \qquad (2.25)$$

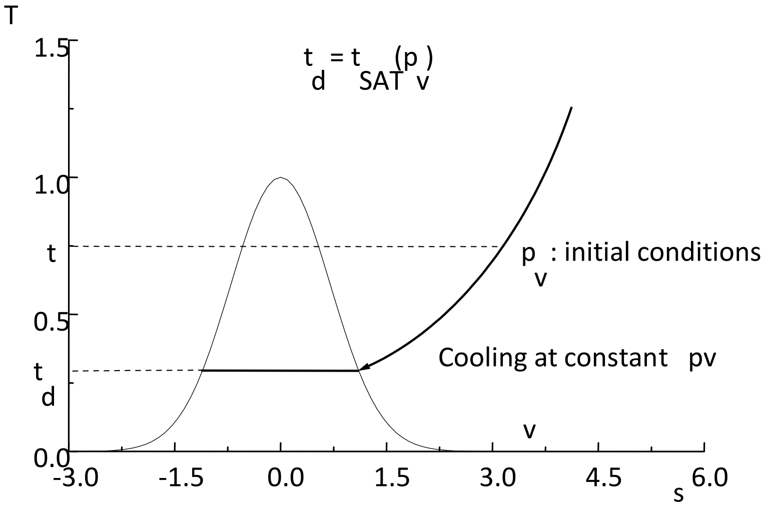
For example, typical values are t_1 =10 °C, t_2 = 20 °C and t_{in} =80°C, t_{out} = 60 °C. The ratio between the two flows can be determined from the balance above.

Cooling

Cooling balances can be still used methods similar to the ones considered for heating, with particular regard to the change of signs in the balances just written. At too low temperatures the water freezes forming brine. This condition creates problems concerned with the correct evaluation of heat exchange and modifies the process since the absolute humidity is no longer a constant. The *dew point temperature* [T_d] is the parameter that indicates if the temperature of the cooling fluid is too low. In fact this is the temperature at which the vapor in moist air begins to condense in form of droplets. This process takes place locally on surfaces at a temperature below T_d and is not a mass phenomenon.

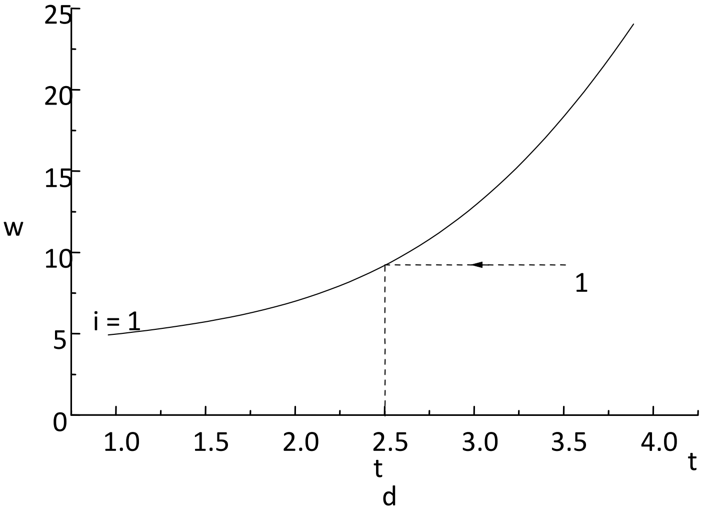
Locally, particles may cool under dew temperature condensing. This is due to the lower cold source in contact with them. This phenomenon leads to saturation conditions and i = 1. It is a process at w constant and a partial pressure of vapor p_v constant (until it starts to condense).

fig 10 Dew point temperature in T-s diagram of water



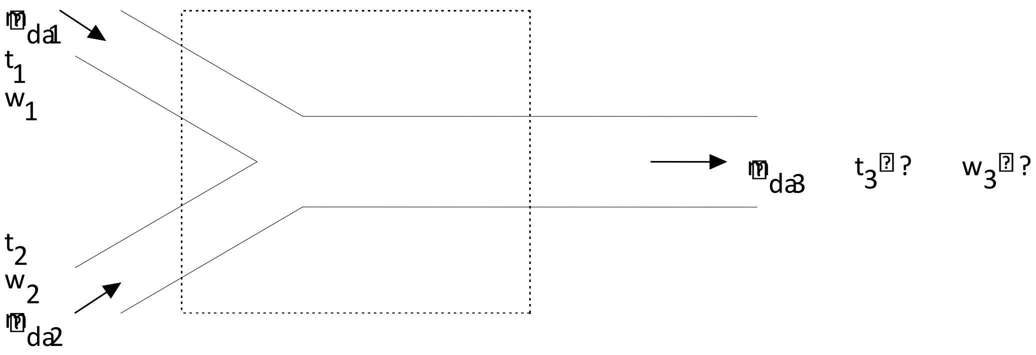
T_d can be obtained considering the tables T - p *in saturation*, or from the graph *w* - *t* for i = 1.

fig 11 dew point temperature



Mixing

fig 12 (mixing flows)



The system is supposed adiabatic: an energy and a mass balance can be written. For the latter we must remember the equation of mass conservation for each elementary part of moist air involved in the process:

$$\dot m_{as\ 3} = \dot m_{as\ 1} + \dot m_{as\ 2}$$

$$\dot m_{v\ 3} = \dot m_{v\ 1} + \dot m_{v\ 2} \quad \text{with } \dot m_v = w \cdot \dot m_{as} \qquad (2.26)$$

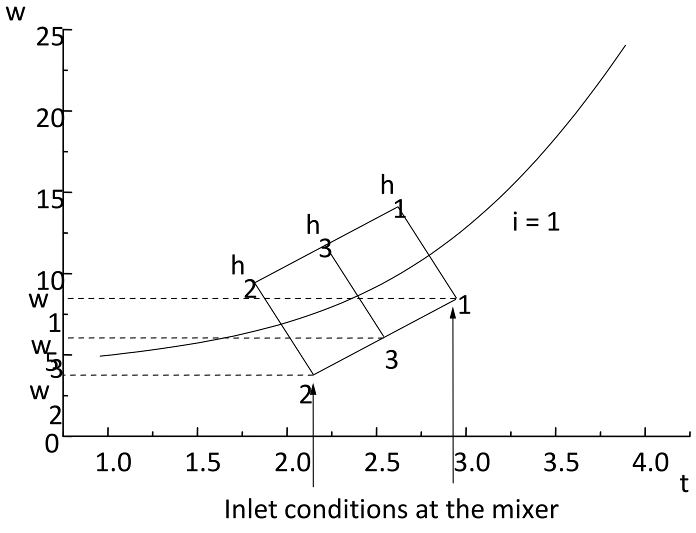
For the conservation of energy :

$$0 = \sum_{in} \dot m_{in} \cdot h_{in} - \sum_{us} \dot m_{us} \cdot h_{us} \quad \Leftrightarrow \quad 0 = \dot m_{as\ 1} \cdot h_{as\ 1} + \dot m_{as\ 2} \cdot h_{as\ 2} + \dot m_{v\ 1} \cdot h_{v\ 1} + \dot m_{v\ 2} \cdot h_{v\ 2} - \dot m_{v\ 3} \cdot h_{v\ 3} - \dot m_{as\ 3} \cdot h_{as\ 3}$$

(2.27)

The conditions of output flow are a weighted average of the conditions of input flows as shown in fig 13.The output conditions are set on the line connecting the input ones separated by segments 1-3 and 2-3 which are respectively proportional to the mass flow 2 (1-3) and 1 (2-3)

fig 13 mixing two flows



$$\frac{\overline{2\,3}}{\overline{2\,1}} = \frac{\dot m_{as\ 1}}{\dot m_{as\ 2} + \dot m_{as\ 1}} \qquad (2.28)$$

In a similar way this happens for T and enthalpies h°: it's easy to determine graphically the output variables thanks to the linearity dependence.

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Humidification process

Humidifying means to increase the hygrometric content, then the variation Δw> 0. It's possible to distinguish between *adiabatic* humidifying, where Δw> 0 and ΔT <0 and the *isothermal* one where Δw> 0 and ΔT = 0. Considering the adiabatic humidification, the variation of T has a negative sign, because if water is introduced in form of droplets, it needs energy to vaporize. Such energy is subtracted from the surrounding air, resulting in a cooling.

Adiabatic humidifying

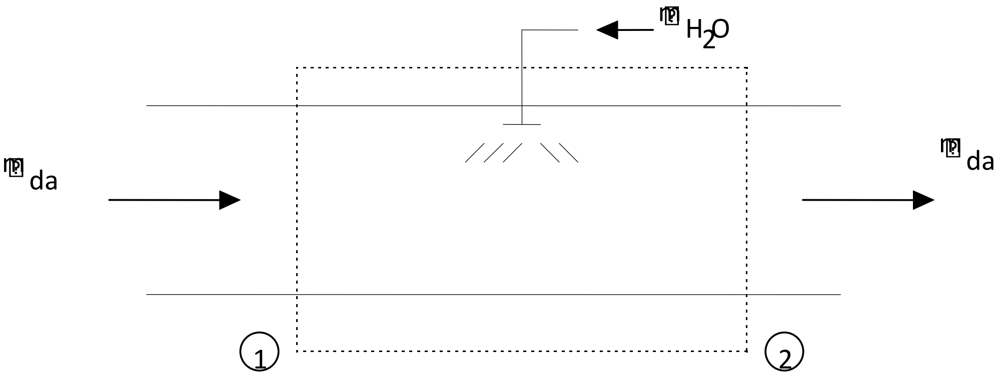


fig 14 (adiabatic humidification)

This system is often used in amusement parks with hot, but dry climates to reduce locally temperature and give some transient comfort to users.

The flow of dry air is constant so we can write

$$\dot{m}_{as\ 1} = \dot{m}_{as\ 2} = \dot{m}_{as} \qquad (2.29)$$

Dividing the mass balance by this flow:

$$0 = \dot{m}_{as\ 1} + \dot{m}_{v\ 1} + \dot{m}_{H_2O} - \dot{m}_{as\ 2} - \dot{m}_{v\ 2} \quad \Leftrightarrow \quad 0 = 1 + \frac{\dot{m}_{v\ 1}}{\dot{m}_{as}} + \frac{\dot{m}_{H_2O}}{\dot{m}_{as}} - 1 - \frac{\dot{m}_{v\ 2}}{\dot{m}_{as}} \qquad (2.30)$$

Which can be written by introducing w and completing the analysis with the energy balance:

$$\dot{m}_{H_2O} = \dot{m}_{as} \cdot (w_2 - w_1)$$

$$0 = \dot{m}_{as} \cdot h^o{}_1 + \dot{m}_{H_2O} \cdot h_{H_2O} - \dot{m}_{as} \cdot h^o{}_2 \quad \Leftrightarrow \quad h^o{}_1 - h^o{}_2 = -\frac{\dot{m}_{H_2O}}{\dot{m}_{as}} \cdot h_{H_2O} = -(w_2 - w_1) \cdot h_{H_2O} \qquad (2.31)$$

The reference temperature is t_{rif} = 0.01°C (liquid phase); since t_{rif} is nearly zero,

$$h_{H2O} = c_{pH2O}\ t_{H2O} \qquad (2.32)$$

$$h^o{}_1 - h^o{}_2 = -(w_2 - w_1) \cdot c_{p\ H_2O} \cdot t_{H_2O} \qquad (2.33)$$

In terms of order of magnitude $h^o{}_1 - h^o{}_2 \cong 0$,when compared to the actual enthalpy values of moist air (of the order of 50kJ(kg)). Indeed $c_{p\ H2O}$ = 4186J/kgK t_{H2O} = 10°C, $(w_2 - w_1) = 5\ g_v / kg_{da}$ and then $h°_1 – h°_2$ = - 200 J / k_{gas} = -0.2 kJ / k_{gas}.

On the ASHRAE diagram, the order of magnitude of total enthalpies h° is about 50 kJ/k_{da}, then the variation of 0.2 kJ/kg_{da} is not significant and the process can be considered as a humidification process approximately isenthalpic.

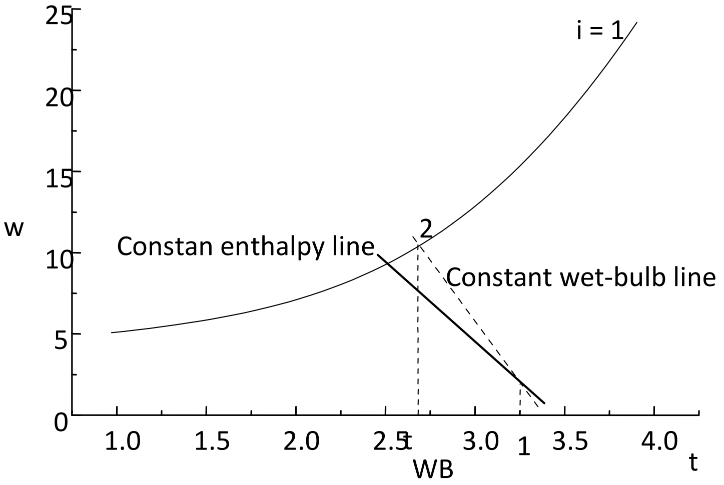


fig 15 adiabatic humidification process on the ashrae diagram

In the *adiabatic saturation process* in which $i_2 = 1$, air is in conditions of saturation at the exit. In this case $t_2 = t_{wet\ bulb}$ namely wet bulb temperature, or the temperature that humidified air through the process of adiabatic saturation will reach.

As regards *isotherm humidification*, water can be introduced in form of vapor to air temperature as shown in fig 16 and 17.

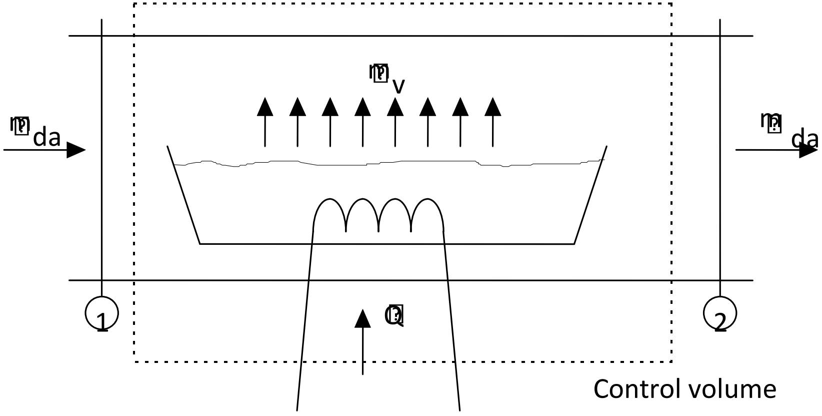


fig 16 isothermal humidification

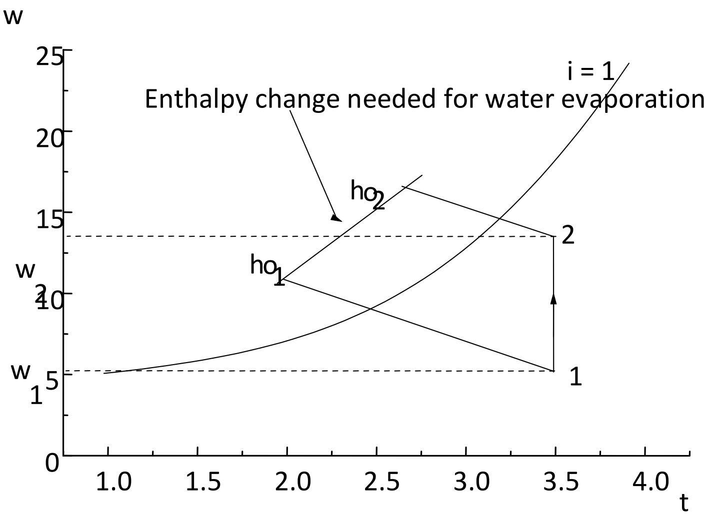


fig 17 isothermal humidification on Ashrae

$$\dot{m}_v = \dot{m}_{as} \cdot (w_2 - w_1)$$
$$\dot{Q} = \dot{m}_{as} \cdot \left(h^o{}_2 - h^o{}_1\right) \qquad (2.34)$$

This is the typical latent heat exchange, enthalpy increase is completely used to raise “w”.

Hassman Hygrometer

It is a system to measure the environmental *relative humidity:*

$$i = \frac{p_{vap}}{p_{sat}} \qquad (2.35).$$

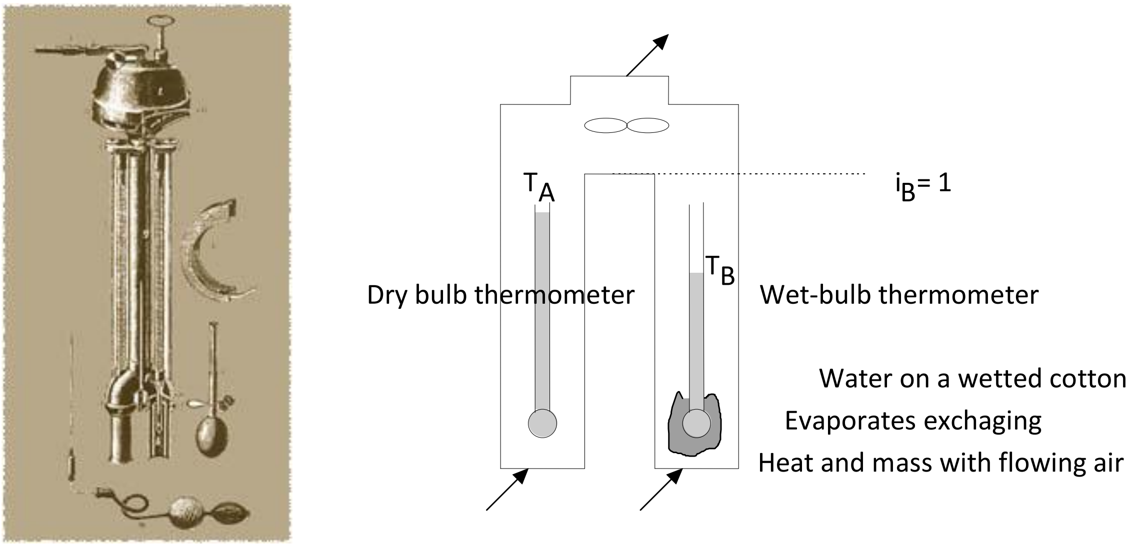


fig 18 Hasmann humidity meter

A fan lets air flow through two pipes, both with thermometers. The air which runs the right part of the meter (fig 18) crosses a wet bulb and is subjected to an adiabatic saturation process. Consider the steady state system:

Until the bulb is wet and we are in steady state, T_B is less than T_A.

We assume that $T_B \approx T_{wet\ bulb}$ temperature of air. This assumption is valid only in the case of air-water processes. At the exit of the pipe, where air humidifies, we have i = 1 (*adiabatic saturation* output of the wet bulb). This transformation consists of an adiabatic process which, as seen before, is isenthalpic, where:

location	T	i
inlet dry bulb	Known (read on thermometer) T_a	to determinate i_a= ?
inlet wet bulb	Known (read on thermometer) the same as environment T_a	
output wet bulb	Known (read on thermometer) T_b	Assumed completely saturated i=100%

The process is shown in fig 19 on the Ashrae diagram. Starting from wet bulb conditions, (T_b,i=100%) following the isenthalpic line we intersect with T_a and we read the proper humidity i-curve passing through the intersection point.

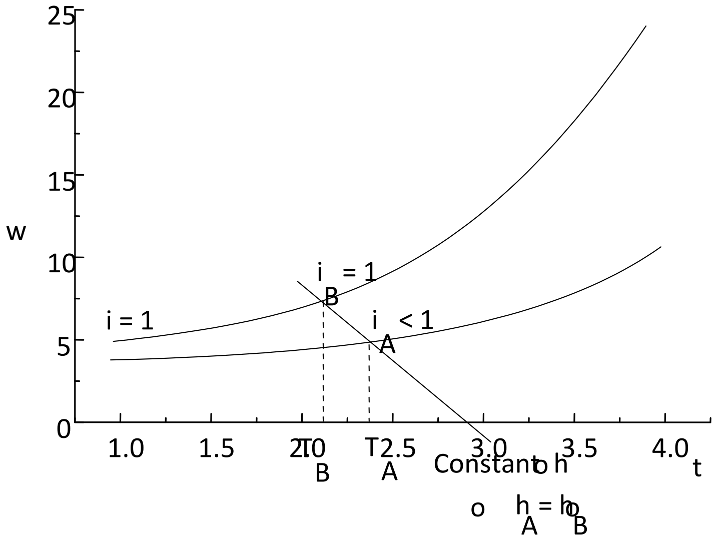


fig 19 Hasmann humidity meter on Ashrae diagram

Cooling and dehumidifying

Sensible cooling is the effect of the condition $T_{cold\ source} < T_{dew\ point}$ with the appearance of local condensation phenomena and a consequent reduction of humidity content. There are no instruments which allow the realization of a "w" reduction in *isothermal* mode and real processes always go through condensation (it will be necessary, cooling + dehumidification). fig 20

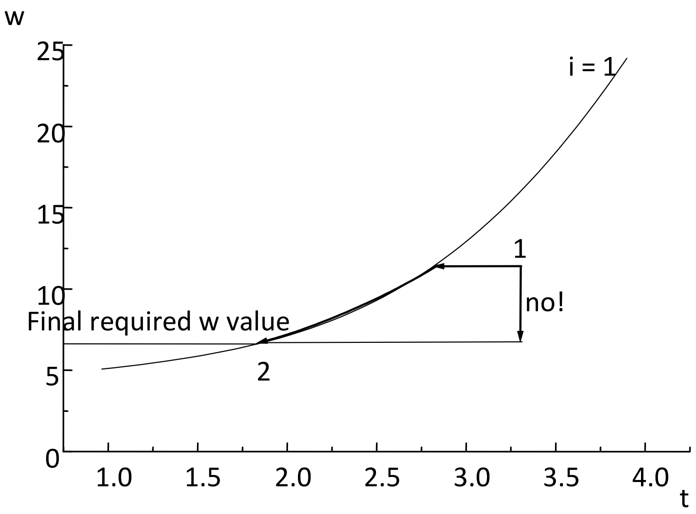


fig 20 cooling and dehumidification on Ashrae diagram

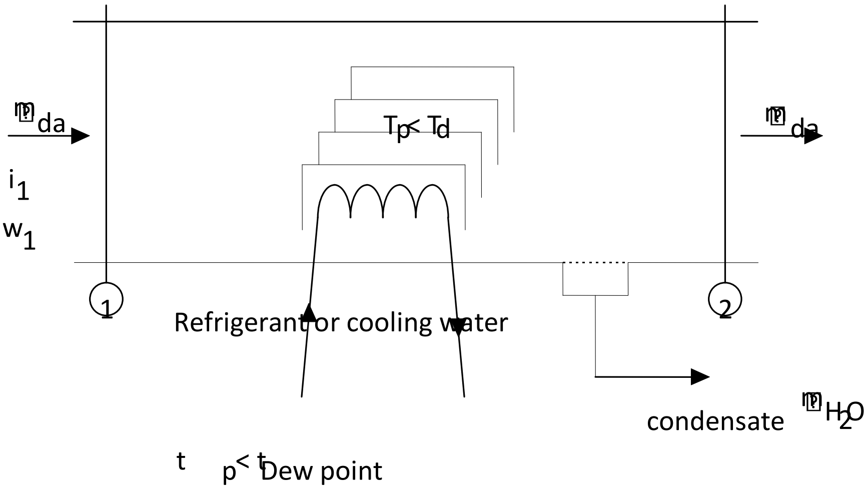


fig 21 cooling and dehumidification systems

In an ideal process, all air is brought to the temperature T_p as shown in fig 22:

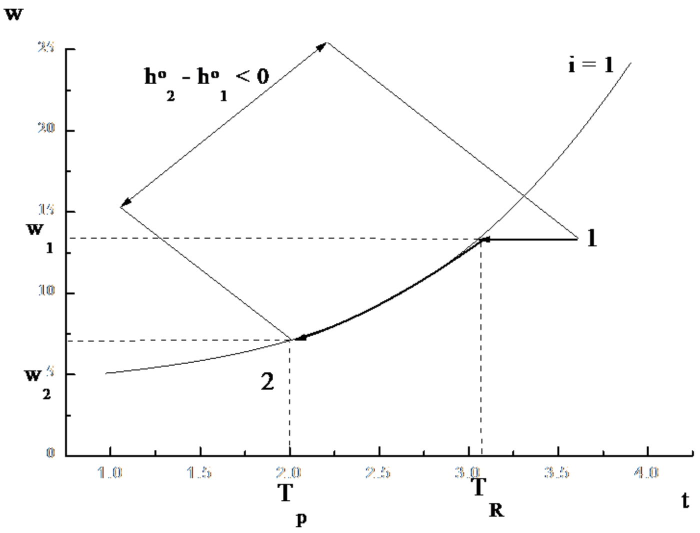
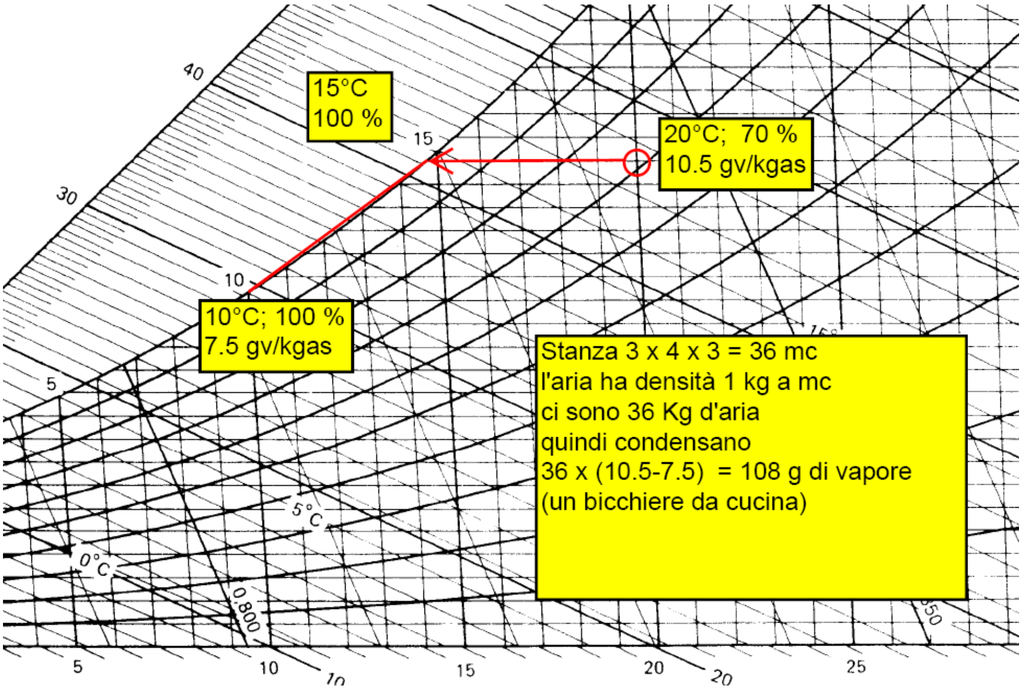


Fig 22 dehumidification on Ashare diagram

The outgoing air is characterized by $T_2 = T_p$ and $i_2 = 1$: changing the T_p value by means of regulation of the cooling system, the finale value w_2 will be easily controlled, too. The lower T_p is, the greater is the amount of water removed from moist air and consequently the lower w_2 will be.
Considering the usual mass and energy balances,

$$\left|\dot{m}_{H_2O}\right| = \dot{m}_{as} \cdot (w_1 - w_2) \qquad (2.36)$$

$$\left|\dot{Q}_{ref}\right| = \dot{m}_{as} \cdot \left(h^o{}_1 - h^o{}_2\right)$$



15°C
100 %

20°C; 70 %
10.5 gv/kgas

10°C; 100 %
7.5 gv/kgas

Stanza 3 x 4 x 3 = 36 mc
l'aria ha densità 1 kg a mc
ci sono 36 Kg d'aria
quindi condensano
36 x (10.5-7.5) = 108 g di vapore
(un bicchiere da cucina)

fig 23 case study 3: quantity of condensed vapor with cooling temperature below dew point temperature

Case study 3

In the fig.23 above the process of moist air cooling and dehumidification is shown. Here air flows in contact with a cold source at 10°C, from 20 °C and 70% of relative humidity at sea level. Initially moist air cools lowing its temperature and increasing relative humidity with absolute humidity constant at 10.5 g_v/kg_{da}. When the relative humidity value reaches 100%, we have the *dew point.* In the example above the dew point temperature is 15°C and at that temperature the first droplets start condensing in moist air. If temperature lowers more, the condensed water is not any more in the moisture but is at liquid state and so the absolute vapor content decreases. At 10°C the maximum content of water in the moist air is 7.5 g_v/kg_{da}. So the condensed water is the difference between the absolute humidity at the start and at the end of the process. If this process has been applied to the whole volume of a room filled with 36 $kg_{a.s.}$, we have 36*(10.5-7.5)=108 g_v condensed. Nearly a glass.

There is a second effect beyond cooling: , a reduction of specific enthalpy:

$$\left|\dot{Q}_{ref}\right| = \dot{m}_{as} \cdot \left(h^o{}_1 - h^o{}_2\right). \qquad (2.37)$$

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This is useful to find the heat flow supplied to the cooling fluid. This fluid will be maintained at low temperature by an inverse refrigeration cycle:

$$\left(\left|\dot{Q}_{ref}\right| = \dot{m}_{ref}\left(h_{us} - h_{in}\right)_{ref} = \frac{\left|L_{ciclofrigo}\right|}{COP_{frigo}} \right) \qquad (2.38)$$

The condensation process is local and not necessarily all the particles are equal to temperature T_p, since it is not assured that they are in contact with the cold surface.

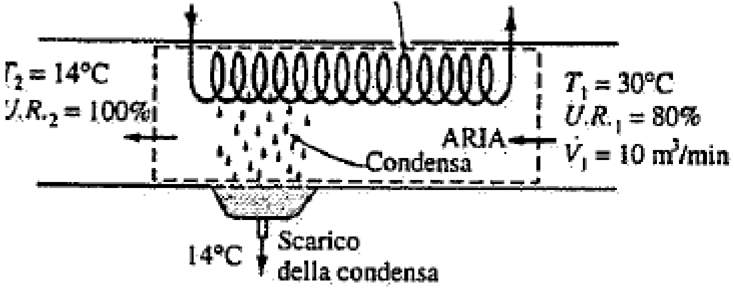


Figure 24

Some parts of the fluid are in contact with the walls and are effectively treated and so they condensate. The parts of fluid flowing in the center of the duct are less influenced by the presence of the walls at lower temperatures and so maybe they are not treated.
We can divide the component into two main parts:

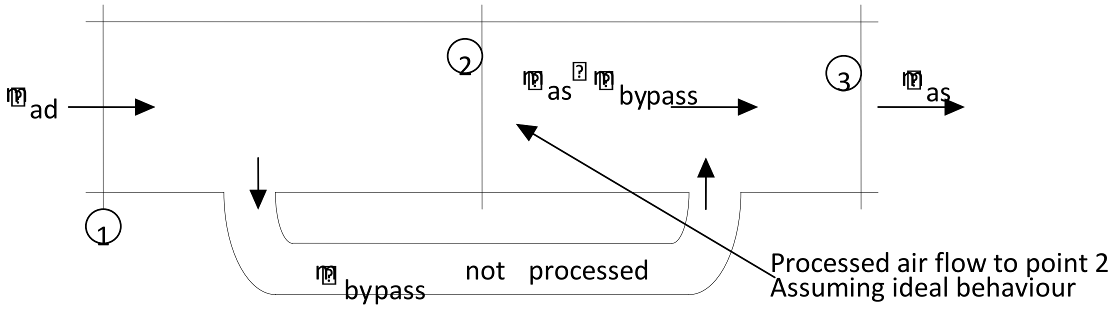


Figure 25

State 3 represents the mix between flows in conditions 1 and 2.

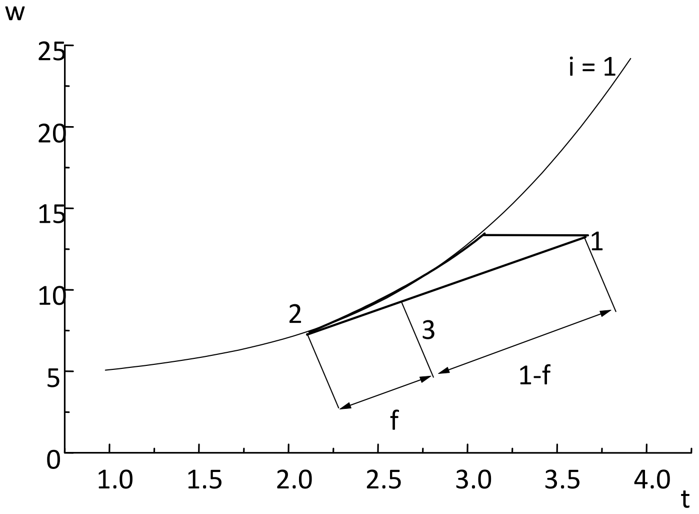


fig 26 mixing flow rates 1 and 2 after dehumidification: geometrical interpretation of the by-pass factor f

As seen before

$$\frac{\overline{2\,3}}{\overline{2\,1}} \propto \frac{\dot{m}_{by\ pass}}{\dot{m}_{as}} = f \qquad (2.39)$$

The case in which $\dot{m}_{by\ pass}$ = 0 implies that segment $\overline{2\,3}$ = 0, that is the component is ideal.

The parameter “f” generally indicates the *by-pass factor* and is provided by the manufacturer in the same way they certify the performance of compressors or expanders. This parameter does not depend on the *system* but on the way in which this is built. In the case of *cooling*, $\dot{m}_{as}$, w_3, t_1, and i_1 are known; from them “w_1”, the choice of the type of exchanger (f factor required) and the working temperature T_p are obtained.

This is part of an iterative procedure:

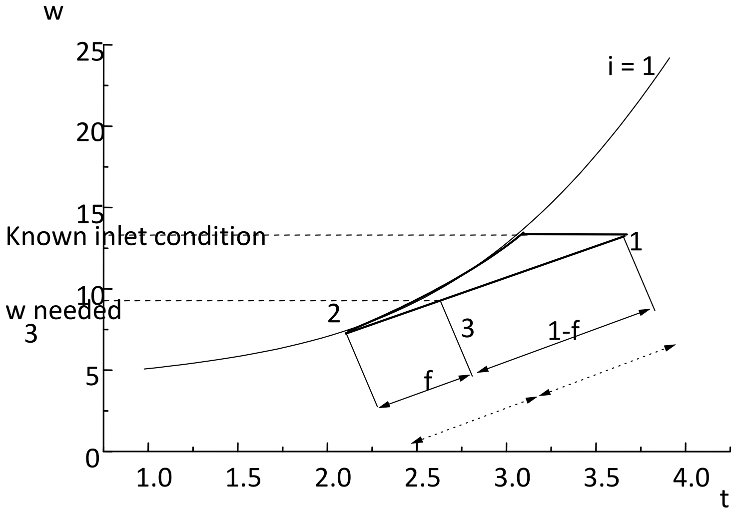


Fig. 27 mixing after dehumidification, with T2 control to determine w

After having chosen an exchanger (set f and T_p), in the optimal conditions there is a point 3 in which $\overline{2\,3} \propto f$ e $\overline{3\,1} \propto 1-f$. If we consider the ideal system (f = 0), it is possible to determine T_p', corresponding to the desired value for w_3. In the real case (f ≠ 0), with the same T_p', $w > w_3$ we must choose $T''_p < T_p'$ and so on, up to the determination of the correct value for T_p. If the assumed values were not realistic or $T_p < 0$ °C, for a water system, it would be necessary to change the factor f (in other words to change the type of plant solution).

Air-conditioning systems

Air conditioning system is based on air treatment and allows to *maintain over time the desired air conditions*. It's essential to know the type of activity carried out in the air conditioned room, in addition to its properties. The significant values are:

room volume	V [m^3]
values of temperature and relative humidity	t_A [°C], i_A

number *n* [volume/hour] of air changes per hour, or the number of times that the air in the room is completely renewed (typical values are n = 0.5 - 1.5 volumes / hour, when air changes are due to doors, windows and air infiltrations, while in an surgery room n = 10 is required and in a sterile room n = 100)

quality and quantity of *thermal exchanges and exchanges of mass* (water vapor) with outside

presence of internal *sources* of heat and vapor (they are computed as external flows to the air conditioning system)

It is important to take into account the exchange of heat and mass (water vapor), in addition to heat exchanged with outside environment (different situations depending on the season). All heat and vapor sources inside the conditioned volume must be regarded as *external* to the air conditioning system. These considerations are summarized by specifying a value for the net heat flux $\dot{Q}$ [W] and for vapor flow $\dot{m}_{H_2O}$ [kg/s].

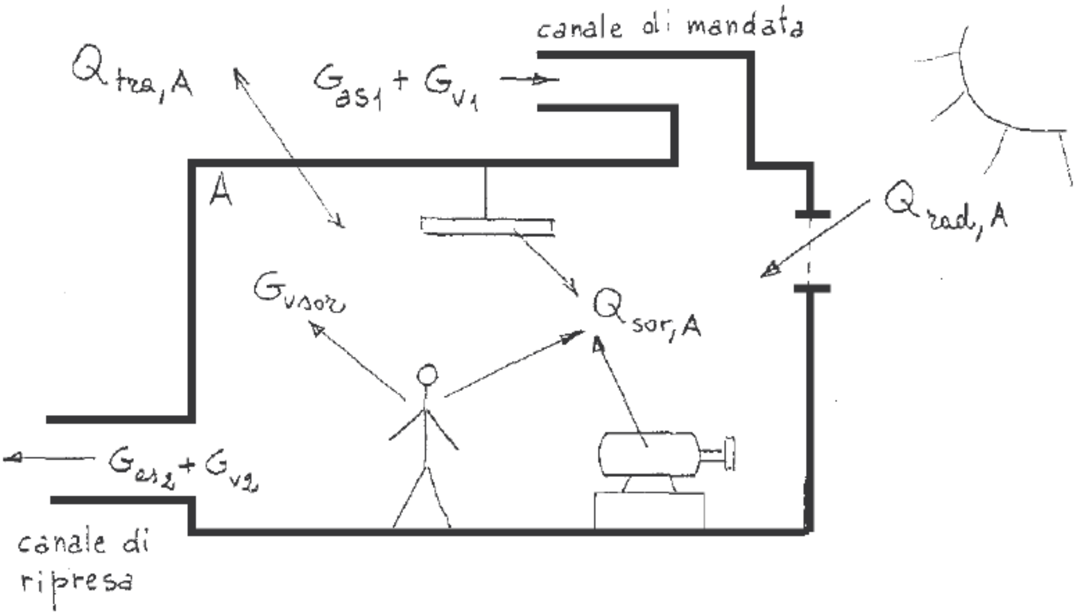
The control variables are:

temperature	T [°C]
relative humidity	i [%]
air speed	w [m/s]
dust and pollutants	

In some environments and for particular processes (ex. in electrical assembly of hard disk drives or in bioengineering processes) the *absence of dust* is crucial. Another important aspect is the effect of dust on humans: particles larger than 3 µm are not let to pass through the nose and those of very small size are expelled. However particles of intermediate size can come in and not go out, causing serious damages (ex silicosis). In all these cases it is necessary to provide for the placement of *filters* in air conditioning systems.

Air speed is important because heat exchange between human body and environment also depends on it. Moreover, this variable is related to another undesirable effect: noise (air passing through pipes can induce vibrations). Typically air speed in the distribution channels is less than 3 m/s; in environment an appropriate value is approximately less than 1.5 m/s. This is controlled through the passage sections (A), with mass balance equation

$$\dot{m} = \rho \cdot W \cdot A \qquad (2.40)$$



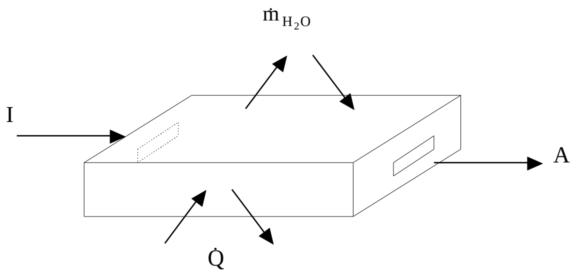


fig 28 Balance of a room (in Italian)

From the psychometric chart or the ideal gas equation, with given data t_A and i_A, it is possible to determinate the specific volume of air, indicated as v_A $\left[\frac{m^3_{mix}}{kg_{as}}\right]$. The relationship for volumetric flow rate is

$$\dot{G}_A = \frac{V \cdot n}{3600} \left[\frac{m^3_{mix}}{s}\right] \qquad (2.\ 41)$$

and the specific volume $\Rightarrow \dot{m}_{as} = \frac{\dot{G}_A}{v_A} \left[\frac{kg_{as}}{s}\right]$. (2.42)

$$\begin{aligned} \Delta h^o &= h^o{}_A - h^o{}_I = \frac{\dot{Q}}{\dot{m}_{as}} \\ \Delta w &= w_A - w_I = \frac{\dot{m}_{H_2O}}{\dot{m}_{as}} \end{aligned} \qquad (2.43)$$

With these equations it is possible to determine the input set values that maintain the environment in the desired conditions.

Winter heating

Fig 29 shows the schematic diagram of a plant for winter environmental conditioning (in the scheme):

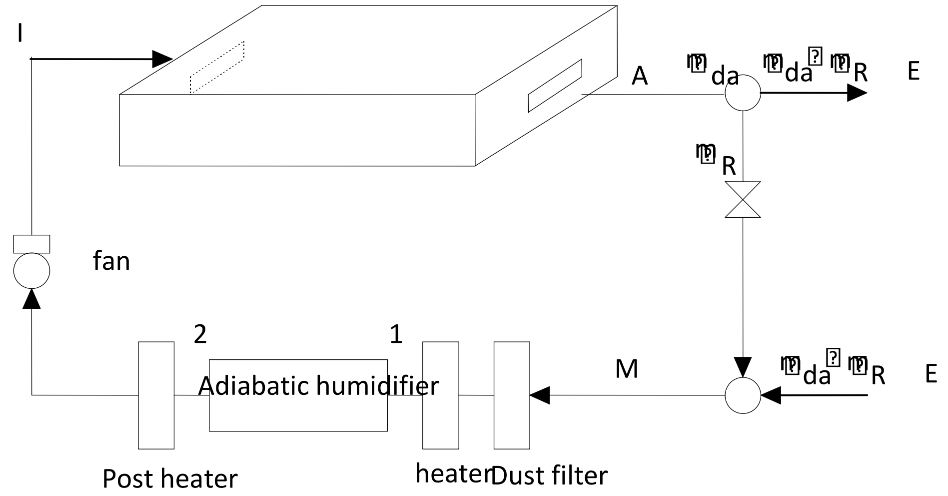


fig 29 schematic diagram of winter air conditioning plant

The water energetic contribution must be considered: the situation in which water is introduced as liquid is different from the case in which vapor is injected.

When there is a change of air, the plant does not take all fresh air from outside because it would be too expensive: in part it is recycled through the system and then mixed with external air.

The partial replacement from outside is still necessary, due to the oxygen supply and the expulsion of pollutants. In the diagram above we see that the filter does not perform thermodynamic operations on air; retained particles are not taken into account, neither in energetic, nor in mass balances. The contribution of air flow from outside, necessary to renew the content of oxygen, is shown in the lower part

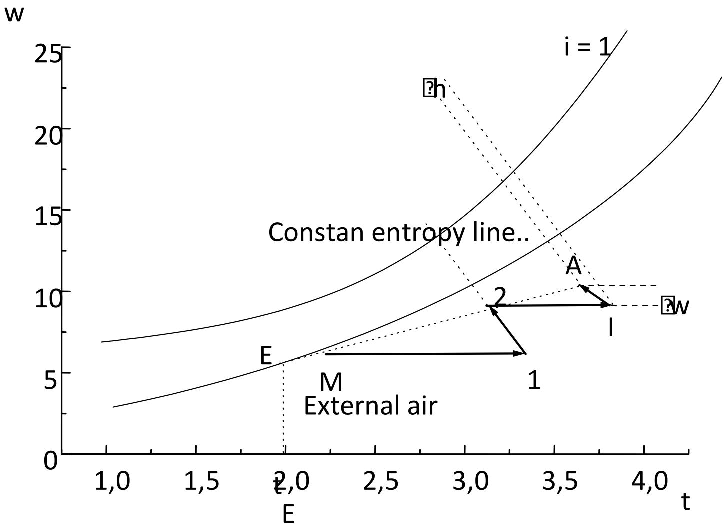


fig 30 schematic process diagram of winter air conditioning plant on ashrae

During winter, there will be a decrease in enthalpy from inlet point (I), to room (A) since air cools by dissipating heat toward outside: so enthalpy input must be higher. Absolute humidity tends to increase because of air humidification: so there should be a minor “w” contribution. External conditions are known parameters, as well as point “A” on the diagram above: we will keep T_e low and i_E (w_E) variable according to weather conditions. Note that in winter low external temperatures always determine low values of absolute humidity even if it may be misty (i=100%).

To consider the presence of people or vapor production in the conditioned room, it is necessary to consider a change ∆w> 0, passing from I to A.

If M is closer to E than to A we have a cycle with much more air input from outside and vice versa. From point M on, there is simply heating (∆w = 0).

Looking at the diagram, it is clear that acting on point 2, you can raise or lower point I, thus varying its humidity content.

Humidification must be adjusted taking into account heating, which is expressed through two phases.

The case of heating and humidification shown in fig 31 would be unacceptable:

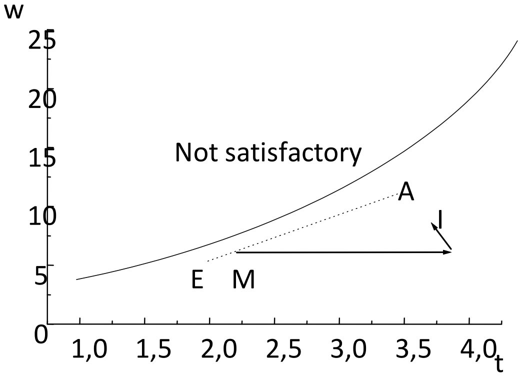


fig 31 heating and humidification with just one heater (not feasible)

In fact a single heater will reach air high temperatures and large losses and there will be problems for plastic components, which usually deteriorate quickly.

Otherwise if we first proceed with humidification and then with heating, there is the possibility to get i = 1 for a value of w less than the one desired.

The isotherm humidification may be an acceptable solution, fig 32. This solution requires anyhow two heaters (one for sensible heating and the other for evaporation during the isotherm humidification):

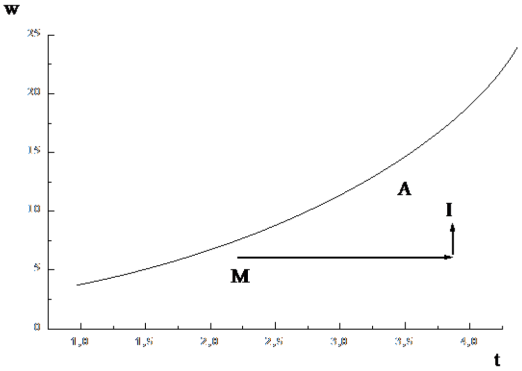


fig 32 isothermal Humidification

In any case the double heating solution is the proper one.

The energy balance for a winter air conditioner, assessing changes in enthalpy on the psychometric chart is:

- thermal heat rate to the first heater:

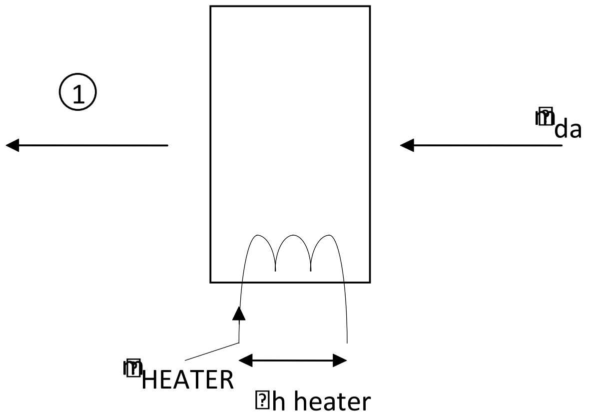


fig 33 energy balance for a winter air conditioner: heater

$\dot{Q}_{1^o} = \dot{m}_{as} \cdot \left(h_1{}^o - h_M{}^o\right)$ (2.44)

The more air is recycled ($h_M{}^o$), the greater is the energy saved.

$\dot{Q}_1 = \dot{m}_{risc} \cdot \Delta h_{risc}$ (2.45)

In case of a resistance-heater the dissipated heat is equal to the heat transferred to air.

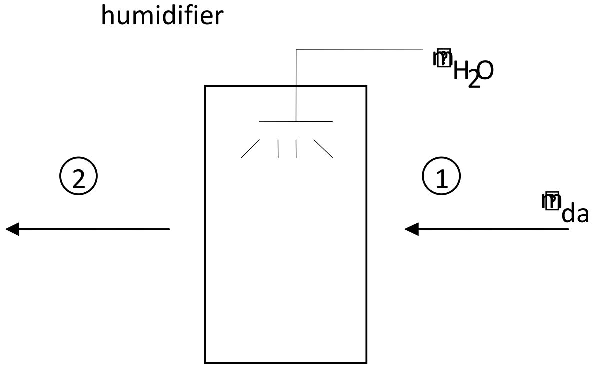


fig 34 energy balance for a winter air conditioner: humidifier

$\dot{m}_{H_2O\ umidificatore} = \dot{m}_{as} \cdot \left(w_2 - w_1\right)$ (2.46)

thermal heat rate to the second heater: (formulas similar to those used for the first heater).

Summer cooling

Let's consider a room of volume V to be maintained in conditions A: t_A and i_A. In summer, outdoor temperature is higher than inside and the exchanged thermal flux is incoming. We need $t_A < t_E$.

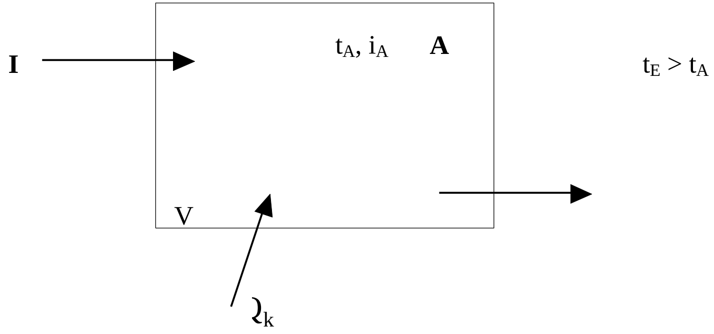


fig 35 energy balance for a room of volume V in air conditions A, in summer . I = Inlet conditions

There are also incoming or outgoing water vapor flows.

Let’s consider the case study in which they are incoming. Under these conditions, the enthalpy that comes out from the conditioned room is higher than the one of the outside air entering the room.

$$\Delta h^{o} = h_A{}^{o} - h_I{}^{o} = \frac{|\dot{Q}|}{\dot{m}_{as}} > 0 \qquad (2.47)$$

$$\Delta w_{AI} = w_A - w_I = \frac{\dot{m}_{H_2O}}{\dot{m}_{as}} > 0 \qquad (2.48)$$

$\dot{m}_{da}$ is evaluated basing on the *number of changes per hour* (n) while the room *volume* (V), $\dot{m}_{da}$ and V are design data.

$$G_{mix} = \frac{n \cdot V}{3600} \ \frac{m^3{}_{mix}}{s} \quad \Rightarrow \quad \dot{m}_{as} = \frac{G_{mix}}{v_A} \qquad (2.49)$$

$$\left[\dot{m}_{as}\right] = \frac{\dfrac{m^3{}_{mix}}{s}}{\dfrac{m^3{}_{mix}}{kg}} = \frac{kg_{as}}{s} \qquad (2.50)$$

v_A is the specific air volume under conditions A.
During summer, external air temperature and humidity vary according to the location.

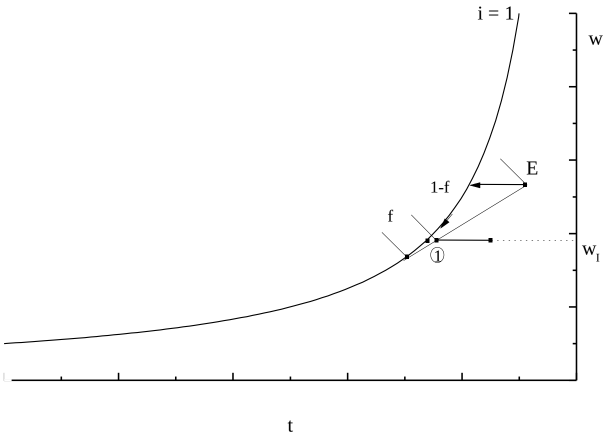


fig 36 processes for a summer air conditioner on Ashrae diagram

The first idea would be to send hot air in a cooling exchanger.
Every possible variation during cooling (at w constant) is very restrained and then from E point we immediately reach the saturation condition i = 1. In this state the content of humidity in the air is too high and not suitable to enter the room.
An exchanger for cooling is not enough: in fact we need cooling and dehumidification. Under ideal conditions, the output of the exchanger might have the desired humidity content.
The relative humidity should be reduced by a post-heating exchanger; this is always necessary even in summer air conditioning, in order to avoid the situation $t_A << t_E$ which it would lead to thermal shocks.
It is advisable to avoid excessive heating or cooling of the body for the transition between outside – inside and vice versa. This can be done by placing approximately $t_A - t_E = 5\text{-}7$ °C.
Instead of an excessive decrease in temperature, we should act on humidity “i”: the comfort condition can be achieved with different conditions (ex 30°C, 50% is equivalent to 27°C, 70%).

The by-pass factor “f” can give further problems, because the end point of de-humidification transformation is not precisely on the line of the desired “w” but is linked to a lower temperature (lower “w”).

The by-pass factor for the exchanger of cooling and drying can be written as

$$f = \frac{\dot{m}_{by-pass}}{\dot{m}_{as}} \quad . \qquad (2.51)$$

Plant for summer air conditioning

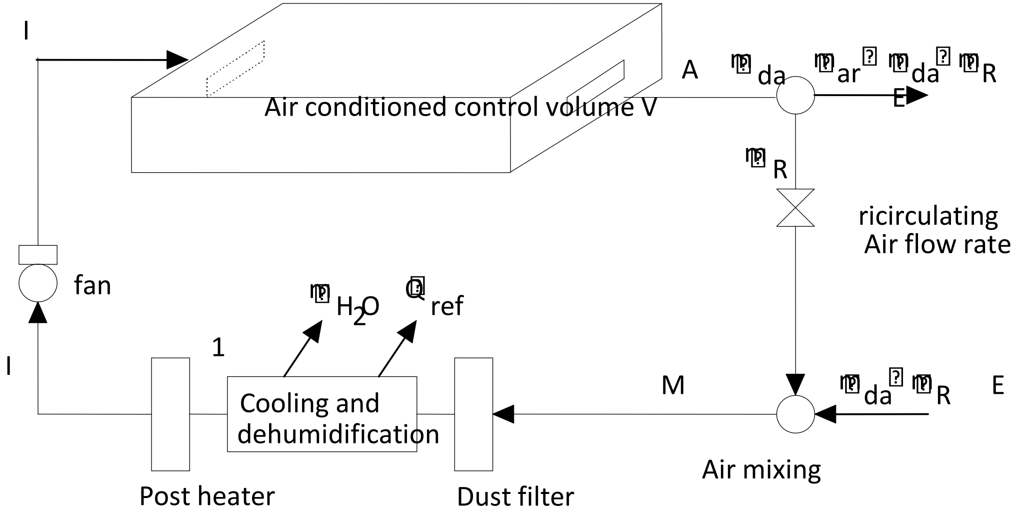


fig 37 Plant scheme for a summer air conditioner

The fan must be dimensioned using the Bernoulli formula, on the basis of pipes and rooms to be conditioned.

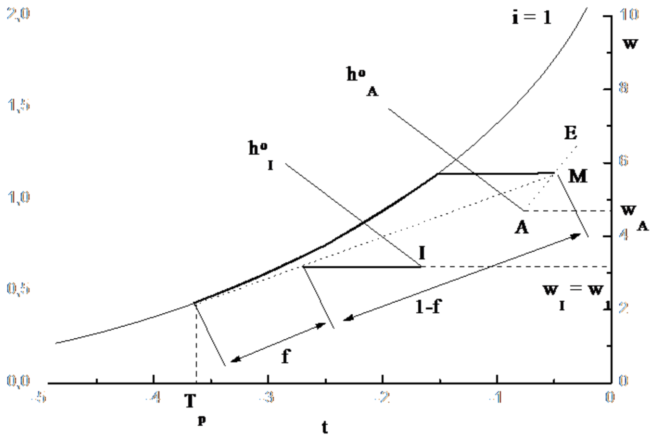


fig 38 processes for a summer air conditioner on Ashrae graphic

In the conditioned room there is an increase of “h°” and “w”, because air receives heat from outside and it is humidified.

“M point” represents the phase of mixing and the segment MA on the diagram is a measure inversely proportional to the flow conditions in A.

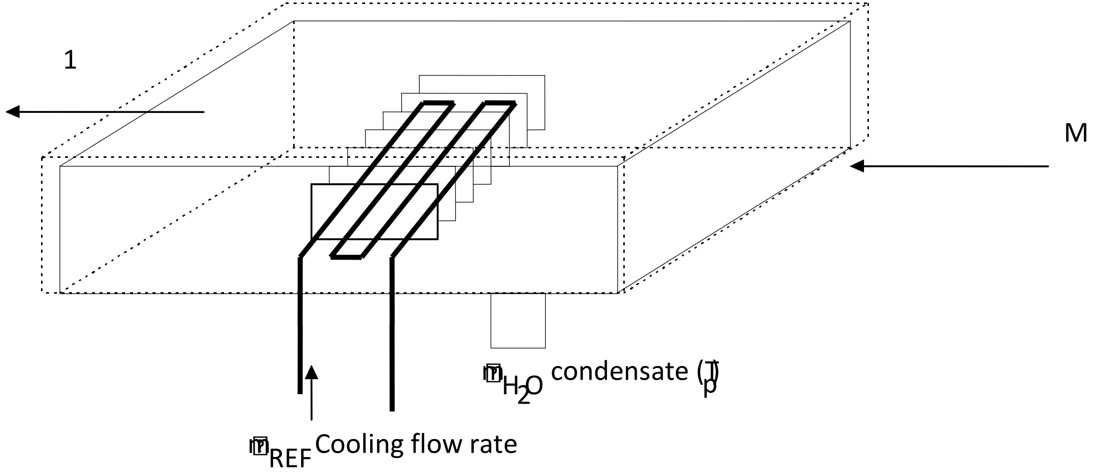
The filter performs no thermo dynamical transformation. The balance equations of each component, with volume V as given data, are as follows:

Input conditions $h_I^{\,o} = h_A^{\,o} - \frac{|\dot{Q}|}{\dot{m}_{as}}$, (2.52)

$$w_I = w_A - \frac{|\dot{m}_{H_2O}|}{\dot{m}_{as}} \quad (2.53)$$

while for the mixer (recycled air - air renewal: A - E ---- M) applies the usual equations previously described.

Exchanger cooling and dehumidification Exchanger
($w_I = w_1$ data project)



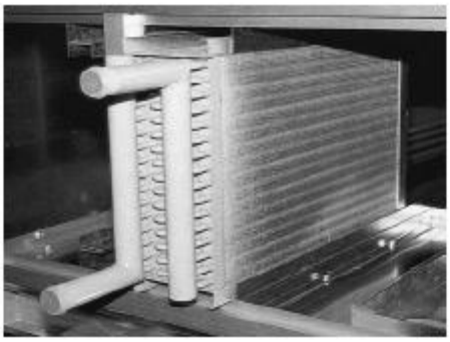


fig 39 cooling and dehumidification heat exchanger

This is an open system with multiple inputs and outputs. From the global point of view, the whole box can be taken and considered as an adiabatic control volume.

At steady state $\frac{\Delta E_{vc}}{\Delta \tau} = 0$ (2.54)

$$\frac{\Delta E_{vc}}{\Delta \tau} = \dot{Q} - \dot{L}_e + \sum_{in} \dot{m}_{in} \cdot h_{in} - \sum_{us} \dot{m}_{us} \cdot h_{us} \quad (2.55)$$

In the previous equation the first member, $\dot{L}_e$ and $\dot{Q}$ are equal to zero, since there are no fans or circulators and the control surface is supposed adiabatic. We obtain:

$$0 = 0 - 0 + \sum_{in} \dot{m}_{in} \cdot h_{in} - \sum_{us} \dot{m}_{us} \cdot h_{us} \quad (2.56)$$

$$0 = \dot{m}_{as}\cdot h_M{}^{o} + \dot{m}_{REF}\cdot h_{REF_{IN}} - \dot{m}_{as}\cdot h_1{}^{o} + \dot{m}_{REF}\cdot h_{REF_{US}} - \dot{m}_{H_2O}\cdot h_{COND} \qquad (2.57)$$

where h_{cond} = 4186 T_p (J/kg) is related to water in liquid form; T_p is the temperature of cold air, with low energy content.

The conservation of mass is

$\dot{m}_{daIN} = \dot{m}_{daOUT}$

and $\dot{m}_{REF_{IN}} = \dot{m}_{REF_{US}}$ (2.58)

Note: it would be also possible to study this system as a set of subsystems, but for the refrigerant subsystem the adiabatic condition is not true anymore since there is heat exchange.

In the particular case $\dot{m}_{H_2O}$ comes from the dehumidifying process which is negligible and then

$$\dot{m}_{H_2O}\cdot h_{COND} \cong 0 \qquad (2.59)$$

in fact it is shown that

$$\dot{m}_{H_2O} = \dot{m}_{as}\cdot\left(w_M - w_1\right) = \dot{m}_{as}\cdot\left(w_M - w_I\right) \qquad (2.60)$$

magnitude order of 5 10^{-3}, while h_{cond} = 4186 T_p

$$\dot{m}_{REF} = \dot{m}_{as}\cdot\frac{h_M{}^{o} - h_1{}^{o}}{h_{REF_{US}} - h_{REF_{IN}}} \qquad (2.61)$$

For example, air-conditioning systems are possible with cold water (typically with t = 3-5 ° C), or with *freon* (in a reverse cycle system). In both cases we have for water:

$$h_{REF_{US}} - h_{REF_{IN}} = c_{p\ H_2O}\cdot\left(t_{us} - t_{in}\right) \qquad (2.62)$$

And we have for Freon (refrigerant inside the refrigeration machine):

$$h_{REF_{US}} - h_{REF_{IN}} = \Delta h_{evap} \qquad (2.63)$$

(variation of the evaporation enthalpy) for *freon.*

In any case, it is necessary to provide a reverse refrigeration cycle: small plants use a reverse cycle cooling system settled in the exchanger, while big plants employ large cooling batteries united with a global cooling cycle.

Post heating heat exchanger

For the correct operation of the exchanger, it is necessary to prevent ice formation on fins and temperature must be at least T_p> 2°C.

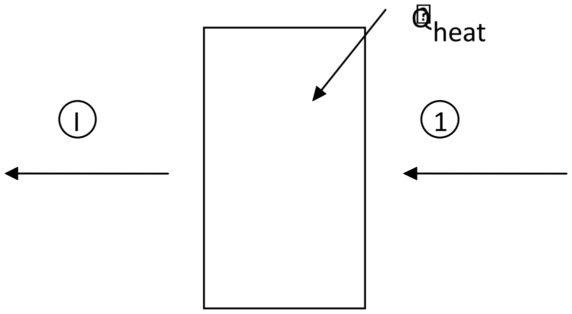


fig 40 post heating exchanger

The post heating exchanger is generally used only in large plants and in its simplest version is composed of an electric system

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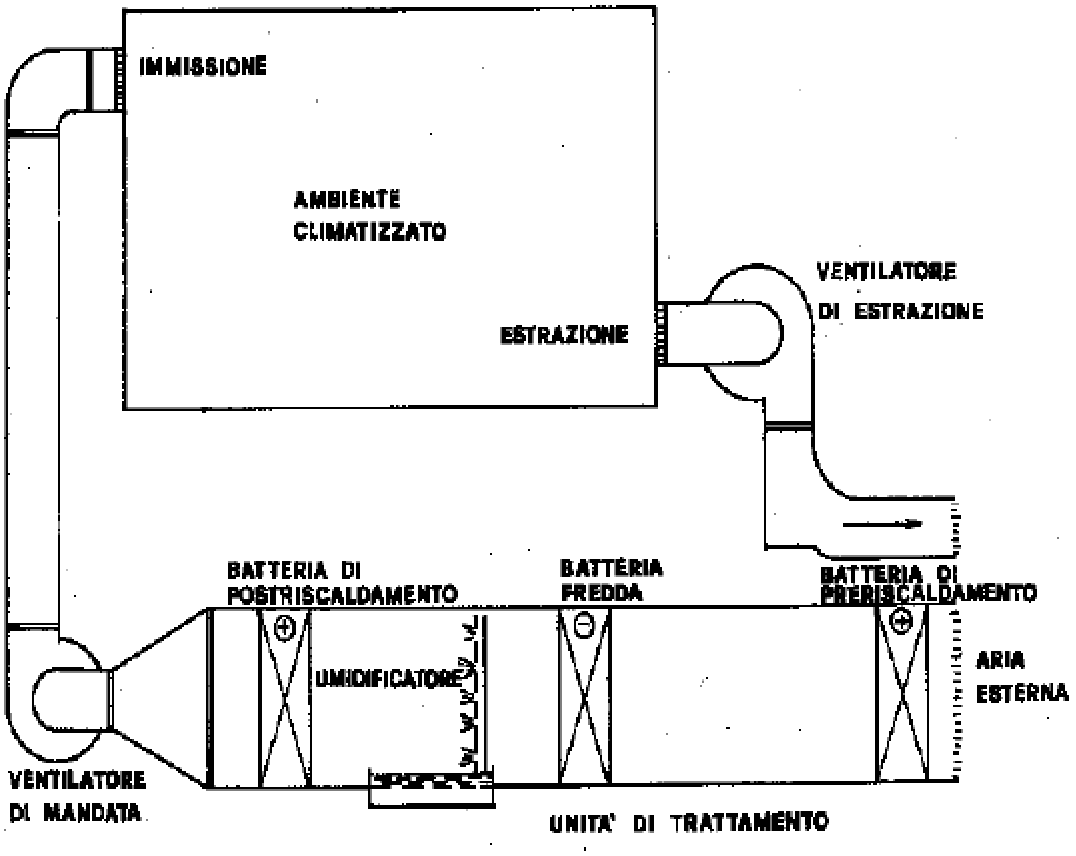
$\dot{Q}_{heat} = V \cdot I \; (Volt \cdot Ampere) = (Watt$

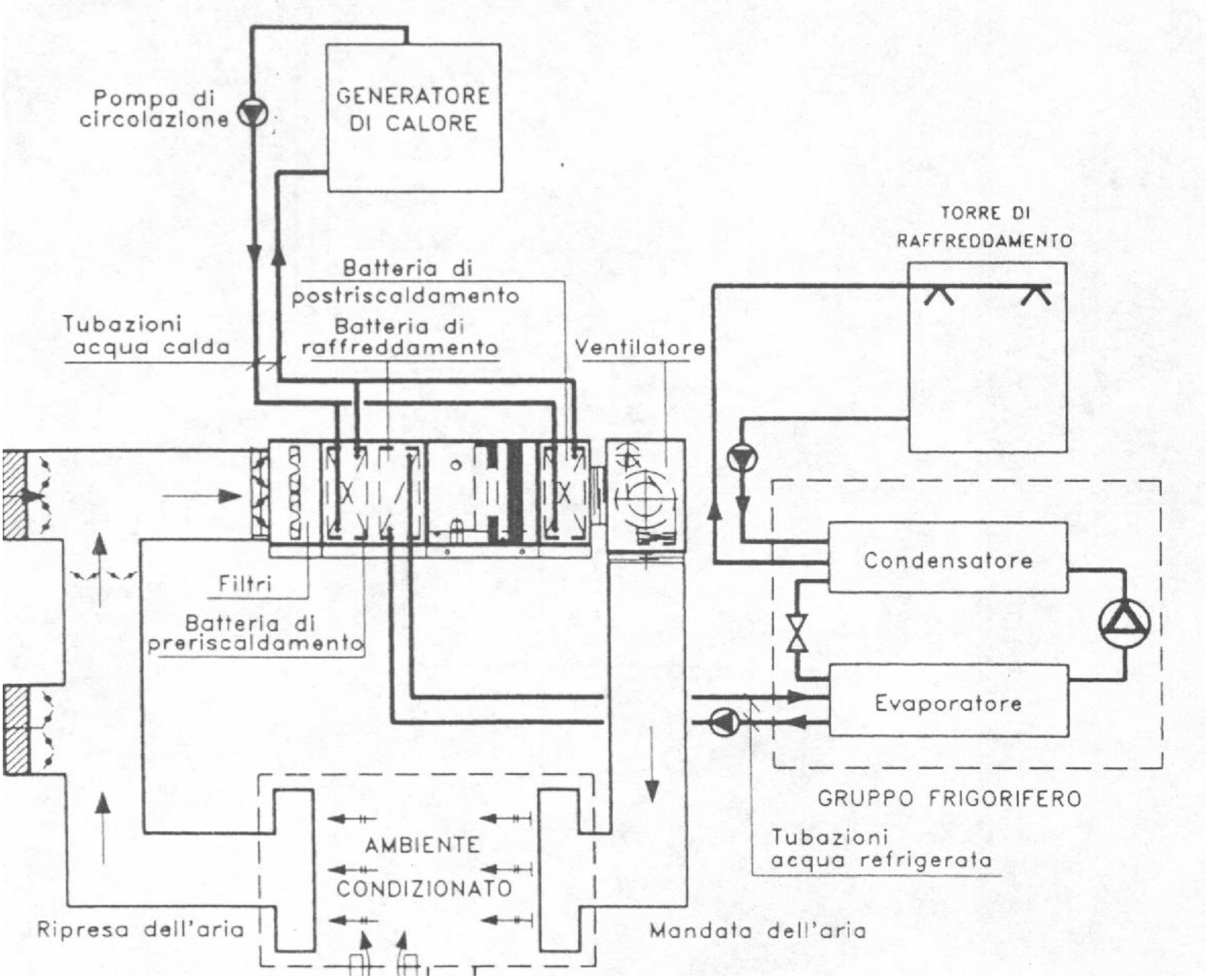
in case of a resistor) (2.64)

Large cooling systems use two circuits of water (hot and cold), in distribution in the various rooms of a building.

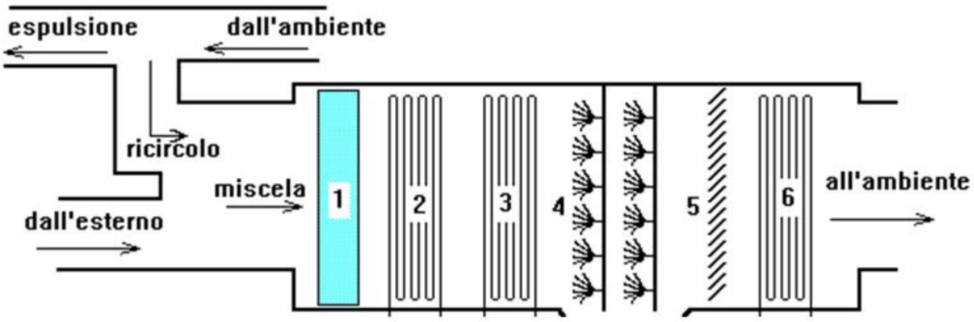
$$\dot{Q}_{risc} = \dot{m}_{as} \cdot \left(h_I{}^o - h_1{}^o \right) \qquad (2.65)$$

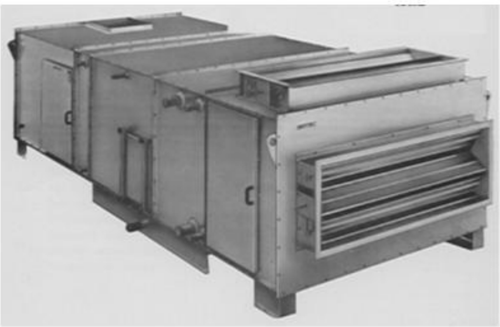
$$\dot{Q}_{risc} = \dot{m}_{H_2O} \cdot c_p \cdot \left(t_{in} - t_{us} \right) \text{ (heating water circulation)} \qquad (2.66)$$




Pompa di circolazione
GENERATORE DI CALORE
TORRE DI RAFFREDDAMENTO
Batteria di postriscaldamento
Tubazioni acqua calda
Batteria di raffreddamento
Ventilatore
Condensatore
Filtri
Batteria di preriscaldamento
Evaporatore
GRUPPO FRIGORIFERO
AMBIENTE CONDIZIONATO
Tubazioni acqua refrigerata
Ripresa dell'aria
Mandata dell'aria

A.T.U. (AIR TREATMENT UNIT)





filter *(1)*

During summer only components #2 *(cooling coil) and* #6 *(post-heating coil) are operating, while*

in winter the working components are Only the #3 *(heating coil),* #4 *(adiabatic saturation) and* #6 *(post-hetaing coil).*

ATU in practice

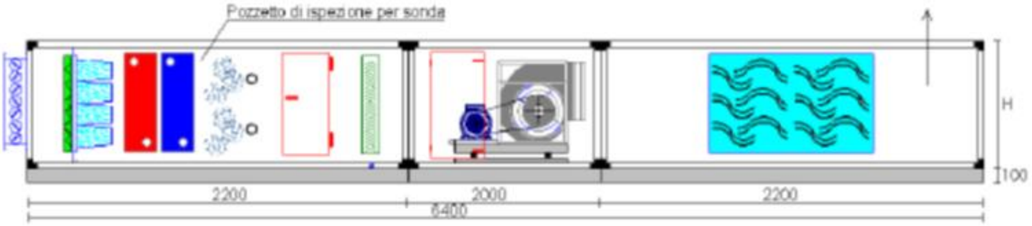


Figura 124: Esempio di selezione di CTA mediante CAD

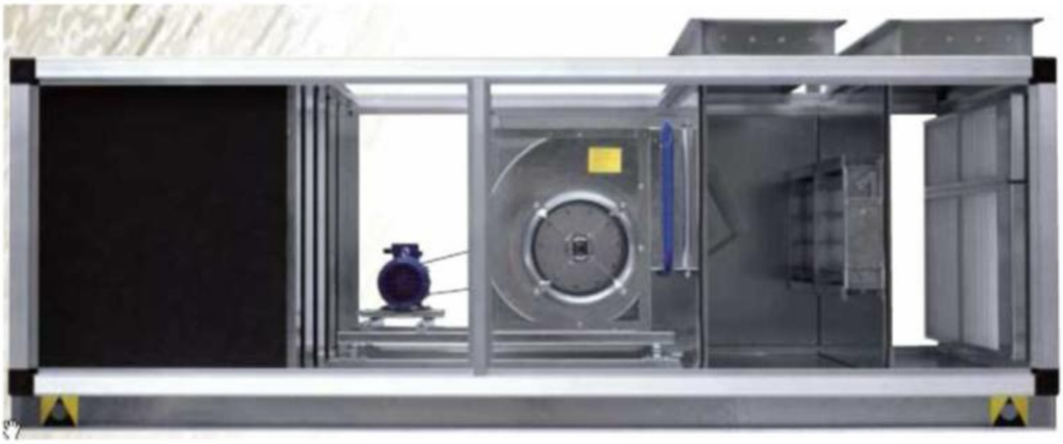


Figura 125: Vista interna di una CTA