Technical College Baghdad

Air condition Engineering Dept.

Refrigeration and air Conditioning
2nd year Lectures

By

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Refrigeration and Air-conditioning Engineering

References

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Fundamental properties of Air and Water vapour mixture

There are a number measures of the amount of moisture contained in atmospheric air, before we begin the discussion of thermodynamics of moist air mixture, we shall define their most important properties and indicate where each is used.

1. Dry Bulb Temperature **DBT**: is the temperature of moist-air mixture measured by a perfectly dry sensor, such as thermocouple or glass thermometer.
2. Wet Bulb Temperature **WBT**: is the temperature of moist - air mixture measured by thermometer's bulb covered by a wick that has been thoroughly wetted with water.
3. Partial pressure **Ps** : is the pressure exerted by one gas component on a mixture of several gases. The partial pressure of water vapour is used in define of relative humidity.
4. Relative humidity ϕ : is the ratio of the partial pressure of water vapour to the saturation pressure of water vapour at the existing dry bulb temperature.
5. Moisture content **g** : Is the ratio of the mass of water vapour to the mass of dry air in a moist-air mixture.
6. Dew point Temperature $\mathbf{T_{dew}}$: is the temperature at which a moist air mixture at a given humidity ratio become saturated.

The composition of dry air

Dry air is a mixture of five main component gas together with traces of number of gases: It is reasonable to consider all these as on homogenous substance but to deal separately with the water vapour present, because the later condensable at every day pressure and temperature. The dry air portion of the atmosphere, then, may be thought of gas being composed of true gases. These gases are mixed together as follows, to form the major part of the working fluid;

Gas	*Proportion%*	*Molecular mass*	*Gas*	*Proportion%*	*Molecular mass*
N_2	*78.03*	*28.02*	H_2	*0.01*	*2.02*
O_2	*20.99*	*32*	*Ar*	*0.94*	*39.9*
CO_2	*0.03*	*44*			

Therefore the mean molecular mass of air is 28.97.

STANDARD ADOPTED

Density of air 1.293 kg/m^3 at 101325 Pa and 0^{o}C
Density of water 1000 kg/m^3 at 4^{o}C & 998.23 kg/m^3 at 0^{o}C
Barometric pressure 101325 Pa
Specific force due to gravity 9.802 N/kg or m / sec^2
Specific heat of air at constant pressure 1.005 kJ/kg K
Gas constant of air 0.287 kJ/kg K
Gas constant of water vapour 0.461 kJ/kg K

The General Gas Law

It is possible, as we saw from the 1st year of thermodynamics subject" to combine Boyle's and Charle's laws as one equation:

PV= m.R.T

P	Pressure of gas	kPa
V	Volume of gas	m^3
m	Mass of gas	kg
R	Gas constant	kJ/kg K
T	Absolute temperature	K

Dalton's law of partial pressure:

If a mixture of gases occupies a given volume at a given temperature, then, the total pressure by the mixture equals the sum of the pressure of the constituents, each being considered at the same volume and temperature.

Dry air
a

Water vapour
b

Air vapour mixture
c

T	20^{o}C
ma	1kg
ms	0kg
Pa	100143Pa
Ps	0Pa
Pat	100143Pa

T	20^{o}C
ma	0kg
ms	0.0073kg
Pa	0Pa
Ps	1182Pa
Pat	1182Pa

T	20^{o}C
ma	1kg
ms	0.0073kg
Pa	100143Pa
Ps	1182Pa
Pat	101325Pa

As shown from the figure an air-tight container under three different conditions, from which it can be seen that the partial pressure exerted by the water vapour in (b) and (c) are equal, as are those exerted by dry air in(a) and (c), and in (a), (b) and (c), the total pressure equals the sum of the partial pressures.

Saturation Vapour Pressure

There are two requirements for the vapour of liquid water to occur:

1. Thermal energy must be supplied to the water.
2. the vapour pressure of the liquid must be greater than that of the steam in the environment

The following equation may be used for the vapour pressure of steam over water up to 100^{o}C .

$$log(p) = 28.59051 - 8.2\,log(t + 273.16) + 0.0024804(t + 273.16) - \frac{3142.31}{t + 273.16}$$

where :

t= temperature in ^{o}C

p= pressure in bar

Over ice, the equation to be used is the following:

$$log(p) = 10.5380997 - \frac{2663.91}{t + 273.16}$$

The vapour pressure of steam in moist-air p_s

The question arises; how do we determine the vapour pressure for relative humidity less than 100%, an empirical equation exist which answer this question;

$$p_S = p_{ssw} - p_{at}.A.(DBT - WBT)$$

p_s	Vapour pressure		
p_{ssw}	Saturation vapour pressure at wet bulb temperature (WBT)		
p_{at}	Barometric pressure		
A	constant	A=6.66x10^{-4} ^{o}C^{-1}	when WBT≥ 0
		A=5.94x10^{-4} ^{o}C^{-1}	when WB<0

Example 1:

Calculate the vapour pressure of moist-air at 20^{o}C DBT, 15^{o}C WBT and 95 kPa.

Solution:

From table at 15^{o}C WBT p_{ssw}=1.7044kPa

$$p_S = 1.7044 - 95 \times 6.66 \times 10^{-4}(20 - 15) = 1.3876\,kPa$$

Relative humidity ϕ :

The ratio of the partial pressure of the water vapour in the moist-air, at a given temperature, to the partial pressure of the water vapour pressure in saturated air, at the same temperature.

$$\phi = \frac{p_s}{p_{ss}}$$

p_{ss}= saturated water vapour pressure at dry bulb temperature kPa.

Example 2:

Calculate the relative humidity of moist air at 30^{o}C DBT, 20^{o}C WBT and 100 kPa barometric pressure?

$$p_S = p_{ssw} - p_{at}.A.(DBT - WBT)$$

p_{ssw} = 2.3373 kPa at 20^{o}C WBT

p_s=2.3373 -100x6.66x10^{-4}(30-20)=2.2704kPa

p_{ss}=4.2431 kPa 30^{o}C DBT

$$\phi = \frac{p_S}{p_{SS}} \times 100\% = \frac{2.2704}{4.2431} \times 100\% = 53.5\%$$

Moisture-content, humidity ratio or specific humidity g

It is the mass of water vapour in kilograms which is associated with one kilogram of dry air in air-vapour mixture. It is some times called humidity ratio or specific humidity.

$$g = \frac{m_s}{m_a}$$

ms; mass of water vapour kg
ma; mass of dry air kg

in general

$$pv = mRT$$

$$For\ air \qquad p_a v = m_a R_a T \rightarrow\rightarrow m_a = \frac{p_a v}{R_a T}$$

$$For\ water \qquad p_s v = m_s R_s T \rightarrow\rightarrow m_s = \frac{p_s v}{R_s T}$$

$$g = \frac{m_s}{m_a} = \frac{p_s v}{R_s T} \cdot \frac{R_a T}{p_a v} = \frac{p_s}{p_a} \cdot \frac{R_a}{R_s}$$

$$R_a = \frac{R_o}{M_a} \rightarrow R_s = \frac{Ro}{M_s} \quad Ma = 28.97 \qquad Ms = 18$$

$$\therefore g = \frac{p_s}{p_a} \cdot \frac{R_a}{R_s} = \frac{p_s}{p_a} \cdot \frac{Ms}{Ma} = \frac{18}{28.97} \frac{p_s}{p_a}$$

$$g = 0.622 \frac{p_s}{p_a}$$

$$g = 0.622 \frac{p_s}{p_a - ps}$$

Another form of relative humidity is:

$$g = 0.622 \frac{p_s \cdot \frac{P_{ss}}{P_{ss}}}{p_{at} - ps \frac{P_{ss}}{P_{ss}}} = 0.622 \frac{p_{ss} \cdot \frac{P_s}{P_{ss}}}{p_{at} - p_{ss} \frac{P_s}{P_{ss}}}$$

$$g = 0.622 \frac{\phi . p_{ss} .}{p_{at} - \phi . p_{ss}}$$

Example:3

Calculate the moisture content of moist air at 20^{o}C BDT, 15^{o}C WBT and 95 kPa

p_s=1.7044-95x6.66x10^{-4}(20-15)=1.387kPa

$$g = 0.622 \frac{1.387}{95 - 1.387} = 9.081 \times 10^{-3}\, kg_w / kg_a$$

Example 4:

Calculate the moisture content and relative humidity of saturated air at 20^{o}C and 95 kPa.

Since the air is saturated, therefore WBT=DBT=20^{o}C p_s=p_{ssw}=2.337kPa $\phi = 100\%$

$$g = 0.622 \frac{2.3373}{95 - 2.3373} = 0.0157 kg_w / kg_a$$

Percentage of saturation or degree of saturation μ

The ratio of moisture content of the moist air at a given temperature, t, to the moisture content of saturated air at the same temperature, t.
It is also called some times the degree of saturation.

$$g = 0.622\frac{p_s}{p_{at} - p_s} \qquad g_{ss} = 0.622\frac{p_{ss}}{p_{at} - p_{ss}}$$

$$\mu = \frac{g}{g_{ss}} = 0.622\frac{p_s}{p_{at} - p_s} \cdot \frac{p_{at} - p_{ss}}{0.622\, p_{ss}} = \frac{p_s}{p_{ss}} \frac{p_{at} - p_{ss}}{p_{at} - p_s}$$

$$\mu = \phi\frac{p_{at} - p_{ss}}{p_{at} - p_s}$$

Dew point T_{dew}

It is the temperature at and below which moist air saturated.
Since the air at dew point temperature contain maximum possible amount of water vapour. The moisture content of air remain constant as its cooled until the dew point reached, below dew point, condensation occurs and the moisture content decreases.
The dew point temperature can be found from $\mathbf{p_s}$ by using the following expression for temperature above freezing.

$$T_{dew} = 6.54 + 14.256\ln(p_s) + 0.7389(\ln p_s)^2 + 0.09486(\ln p_s)^3 + 0.4569\, p_s^{\,0.1984}$$

p_s=vapour pressure in kPa

Example 5:
Calculate the dew point temperature of moist air at 20°C BDT, 15°C WBT and 95 kPa.
Ps=1.7044-95x6.66x10^{-4}(20-15)=1.3876kPa

$$T_{dew} = 6.54 + 14.256\ln(1.3876) + 0.7389(\ln 1.3876)^2 + 0.09486(\ln 1.3876)^3 + 0.4569(1.3876)^{0.1984}$$

T_{dew}=11.78°C
Another solution:
At p_s=1.3876 the air is at dew point temperature, therefore the air is saturated and the saturation pressure is 1.3876 kPa, then the dew point temperature is corresponding to 1.3876 kPa.
By interpolation

11	1.3119
T	1.3876
12	1.4017

$$\frac{1.3119 - 1.4017}{1.319 - 1.3876} = \frac{11 - 12}{11 - T}$$

$$T = 11.842^{o}C$$

Specific Volume

This is the volume in cubic meters of one kg of dry air together with the mass of water vapour associated with it.

Example 5:
Calculated the specific volume of moist air at 45^{o}C DBT, 25 ^{o}C WBT, barometric pressure is 101.25kPa

p_s =3.167-101.25x6.66x10^{-4}(45-25) =1.81835kPa
p_a=p_{at}-p_s=101.25-1.81835=99.43165kPa
For one kg of moist air

$$g = 0.622\frac{1.81835}{99.43165} = 0.01137\,kgs/kga$$

$$ma = 1 - 0.01137 = 0.9886\,kg_a$$

$$V_a = \frac{m_a R_a T}{p_a} = \frac{0.9886 \times 0.2876 \times (45 + 273)}{99.43165} = 0.9093m^3$$

$$Vs = \frac{m_s R_s T}{p_s} = \frac{0.01137 \times 0.461 \times (45 + 273)}{1.81835} = 0.91m^3 \approx Va$$

$$v = \frac{V}{m} = \frac{0.909}{1} = 0.909m^3 / kg$$

<u>Moist Air Enthalpy h</u>

The enthalpy of moist air is the sum of the enthalpy of dry air and the water vapour comprising the mixture:

$$h = h_a + g.h_g$$

where;
ha: enthalpy of dry air kJ/kg
hg: enthalpy of eater vapour kJ/kg
g: moisture content kgs/kga

$$h = c_{pa}.DBT + g(h_{g_{ref}} + c_{pv}.DBT)$$

$h_{g_{ref}}$:enthalpy of water vapour at reference temperature =2501.3 kJ/kg.K

c_{pa} :specific heat of air at constant pressure kJ/kg.K

c_{pv}: specific heat of water vapour at constant pressure =1.86 kJ/kg.K

$$h = 1.005DBT + g(2501.3 + 1.86DBT)$$

DBT; dry bulb temperature of moist air ^{o}C

Example 6:
Calculate the approximate enthalpy of moist air at 35^{o}C DBT, 25^{o}C WBT and 101.325 kPa.
p_{ssw}= 3.1671 kPa at 25^{o}C WBT
p_s=3.1671-101.325x6.66x10^{-4}x(35-25)=2.492 kPa

$$g = 0.622\frac{2.492}{101.325 * 2.492} = 0.0252kgw/kga$$

$$h = 1.005 \times 35 + 0.0252(2501.3 - 1.86 \times 35) = 99.885kJ/kg$$

Sheet No. One

1. *a-Air at 30^oC DBT, $17\,^oC$ and 105 kPa enters a piece of equipment where it undergoes a process adiabatic saturation, the air leaving with a moisture content of 5 gw/kga higher than entering. Calculate i-the moisture content of the air entering the equipment ii-the dry bulb temperature and enthalpy of the air leaving the equipment. (6.15 gw/kga, $18\,^oC$, 46.52kJ/kg)*
2. *calculate the moisture content of air at $17\,^oC$ DBT and 40% relative humidity where the barometric pressure is 95 kPa.(5.18gw/kga)*
3. *calculate from the first principles the enthalpy of air at $28\,^oC$ DBT and 1.926 kPa vapour pressure, at atmospheric pressure. (59.03 kJ/kg)*
4. *calculate the dew point temperature and the enthalpy of air at $28\,^oC$ DBT, $21\,^oC$ WBT and 87.7 kPa barometric pressure. ($18.11\,^oC$, 66.7 kJ/kg)*
5. *calculate the relative humidity and percentage saturation of air at 101.325 kPa, $21\,^oC$ DBT, $14.5\,^oC$ WBT. (48.7%, 48.2%)*
6. *calculate the specific volume of an vapour mixture of dry air at $30\,^oC$ DBT, moisture content of 0.015kgw/kga and barometric pressure of 90kPa . . . ($0.99m^3/kg$)*
7. *a sample of air at $30\,^oC$ DBT, $25\,^oC$ WBT and barometric pressure of 101 kPa, calculate the enthalpy of air if this air is adiabatically saturated, the humidity ratio if this air is adiabatically saturated, the humidity ratio of the sample, the partial pressure of water vapour in the sample, the relative humidity .(0.0201kgw/kga, 76.2kJ/kg, 0.018kgw/kga, 2480Pa, 67%)*

The Psychrometry of Air-Conditioning Process

The specific objectives of this lecture are to:

1. Introduction to psychrometric processes and their representation
2. Important psychrometric processes namely, sensible cooling and heating, cooling and dehumidification, cooling and humidification, heating and humidification, chemical dehumidification and mixing of air streams
3. Representation of the above processes on psychrometric chart and equations for heat and mass transfer rates
4. Concept of Sensible Heat Factor, By-pass Factor and apparatus dew point temperature of cooling coils
5. Principle of air washers and various psychrometric processes that can be performed using air washers
6. Concept of enthalpy potential and its use

At the end of the lecture, the student should be able to:

1. Represent various psychrometric processes on psychrometric chart
2. Perform calculations for various psychrometric processes using the psychrometric charts and equations
3. Define sensible heat factor, by-pass factor, contact factor and apparatus dew point temperature
4. Describe the principle of an air washer and its practical use
5. Derive equation for total heat transfer rate in terms of enthalpy potential and explain the use of enthalpy potential.

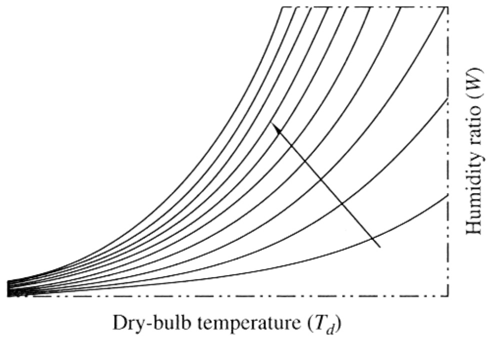
Construction of psychrometric chart

Although the equations that we have developed for the many properties of moist air are used for computer calculation. It is convenient to have the plotted in chart for easy reference during design HVAC system.

The psychrometric chart is an *x y* plot with dry bulb temperature as the abscissa and the moisture content as the ordinate. Since these are two independent thermodynamics variable, all other properties of moist air can be expressed as function of them at a given atmospheric. On the standard psychrometric chart the following moist air properties are plotted:

1-Relativehumidity
2-WBT
3-vapour pressure
4-specfic volume
5-enthalpy

Figure 1a to e shows how line of constant properties listed appear on the psychrometric chart.



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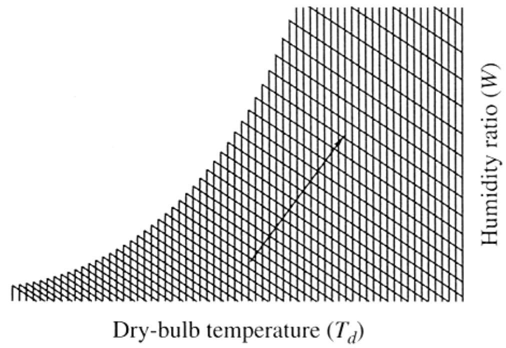

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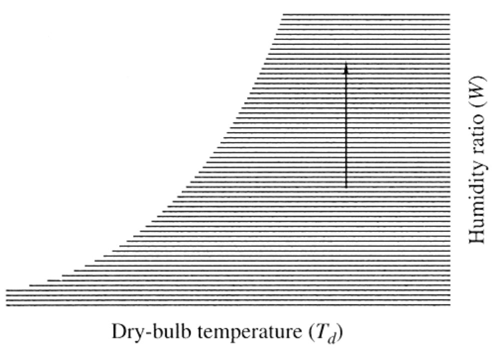

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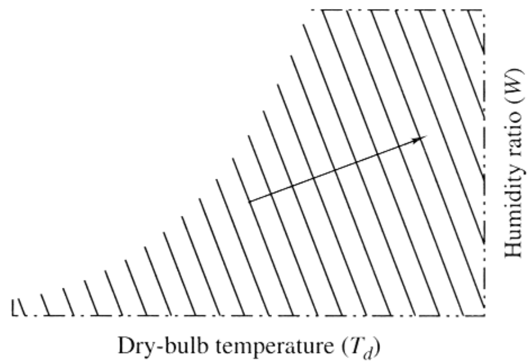

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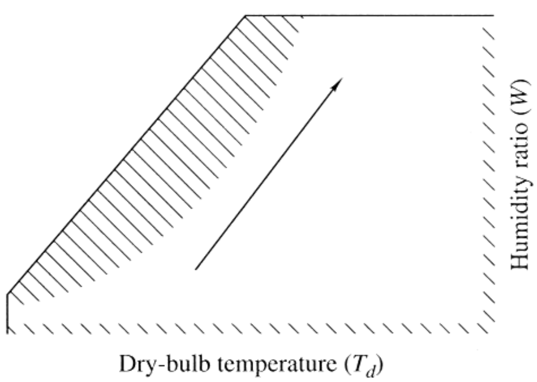

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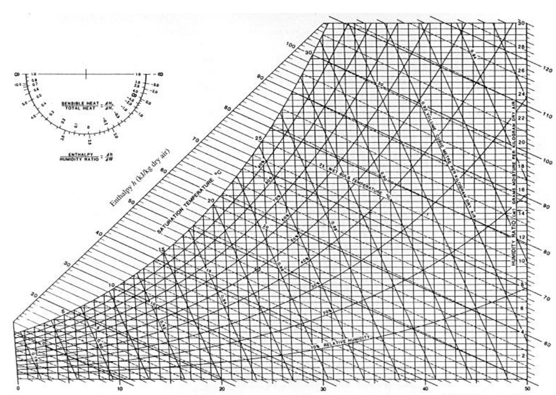

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Figure 1 Skeleton psychrometric chart showing a-constant relative humidity b-constant WBT c-Constant moisture content d-constant specific volume e-constant enthalpy f-the psychrometric chart

Use of the chart

The psychrometric chart can be used in analyzing many processes involving moist atmospheric air.

The Psychrometry of Air-conditioning processes

1- Sensible cooling:

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. Figure 2 shows the sensible cooling process 1-2 on a psychrometric chart. The heat transfer rate during this process is given by:

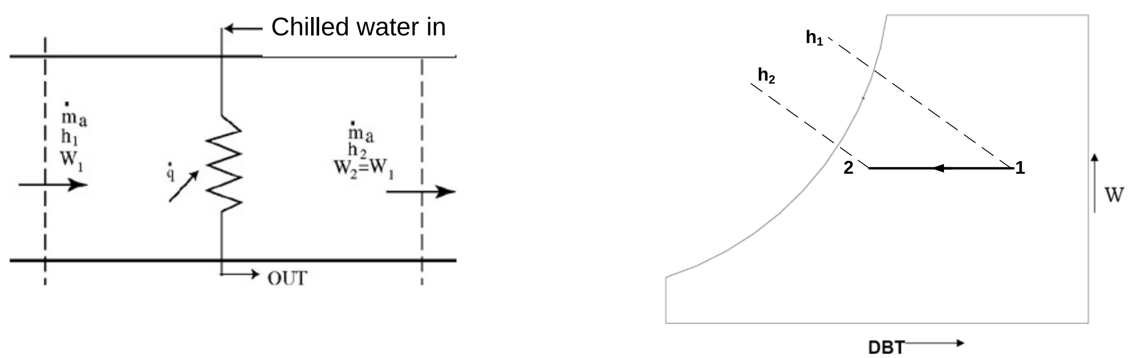


Figure 2 Sensible cooling

$Q_s = \dot{m}_a(h_1 - h_2) = \dot{m}_a.c_{pa}(DBT_1 - DBT_2)$

2- Sensible heating

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil, as shown in figure 3. The heat transfer rate during this process is given by:

$Q_s = \dot{m}_a(h_2 - h_1) = \dot{m}_a.c_{pa}(DBT_2 - DBT_1)$

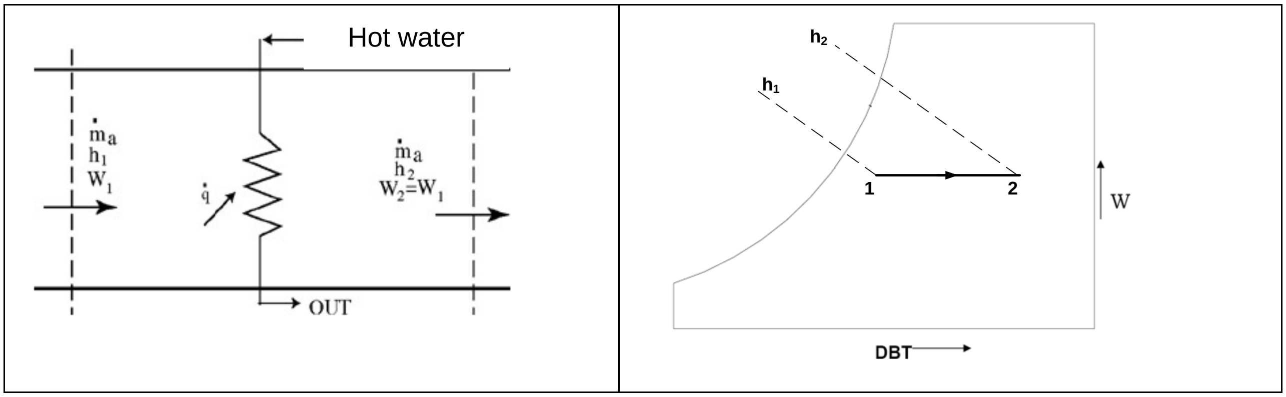


Figure 3 Sensible heating

The variation in the physical properties of the moist- air for the two cases, are summarized below:

	Sensible heating	Sensible cooling
DBT	Increase	Decrease
v	Increase	Decrease
h	Increase	Decrease
WBT	Increase	Decrease
μ	Decrease	Increase
g	Constant	constant
T_{dew}	Constant	Constant
P_s	Constant	Constant

Example1:

Calculate the load on a battery when heats 1.5m^3/s of moist-air, initially at a state of 21^{o}C DBT, 15 ^{o}C WBT and 101.325kPa barometric pressure, by 20 ^{o}C. if low pressure water at 85 ^{o}C flow and 75 ^{o}C return is used to achieve this. Calculate the mass flow rate necessary of water.

Q_{1-2}=ma(h2-h1)

h1=41.88kJ/kg & h2=62.31kJ/kg from the chart

$$v_1 = 0.8439 m^3 / kg$$

$$\dot{m} = \frac{1.5}{0.8439} = 1.777 kg / s$$

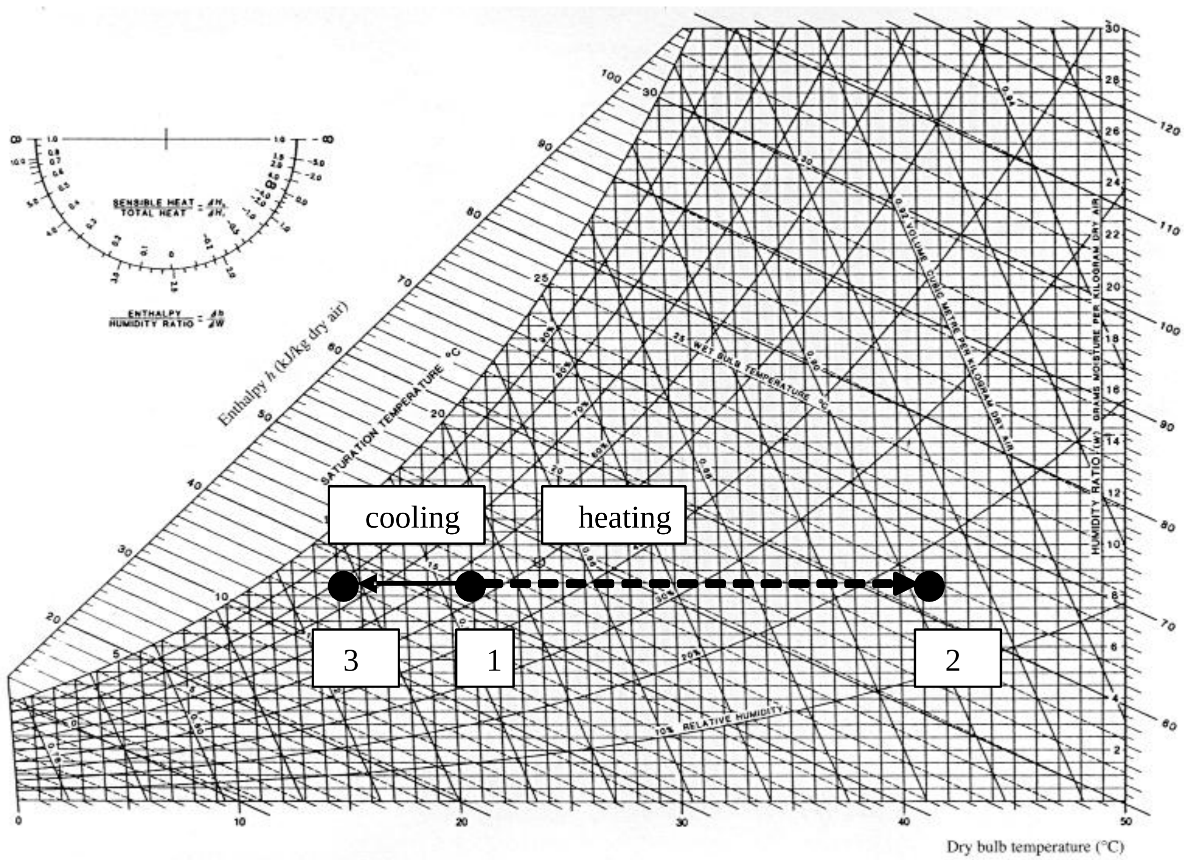
Q_{1-2}=1.777(62.31-41.88)=36.3kW

Heat gain by moist-air= heat lost from the water

$Q=m_w.c_w.(t_{wou}-t_{win})$

$36.3= m_w \, x4.186(85-75)$

$m_w=0.867kg_w/s$



Example2:

If the moist-air mentioned in example 1 is cooled sensibly by 5 ^oC using cooler coil, what is the flow rate of chilled water necessary to effect this cooling if the flow return temperature of 10^oC and 15 ^oC satisfactory.

Q_{1-2}=ma(h1-h3)

h3= 36.77kJ/kg

Q_{1-2}=1.777(41.88-36.77)=9.1kW

Heat lost from air = heat gain by water

$9.1 = m_w x\ 4.186(15-10)$

$m_w = 0.434\ kg_w/s$

3- ***Dehumidification:***

There are four methods whereby moist-air can be dehumidified

I- cooling to a temperature below the dew point
II- adsorption
III- absorption
IV- compression followed by cooling

The first method form is the required matter for this section. Cooling to a temperature below the dew point is done by passing the most air over a cooler coil or through an air washer, as shown in figure 4.

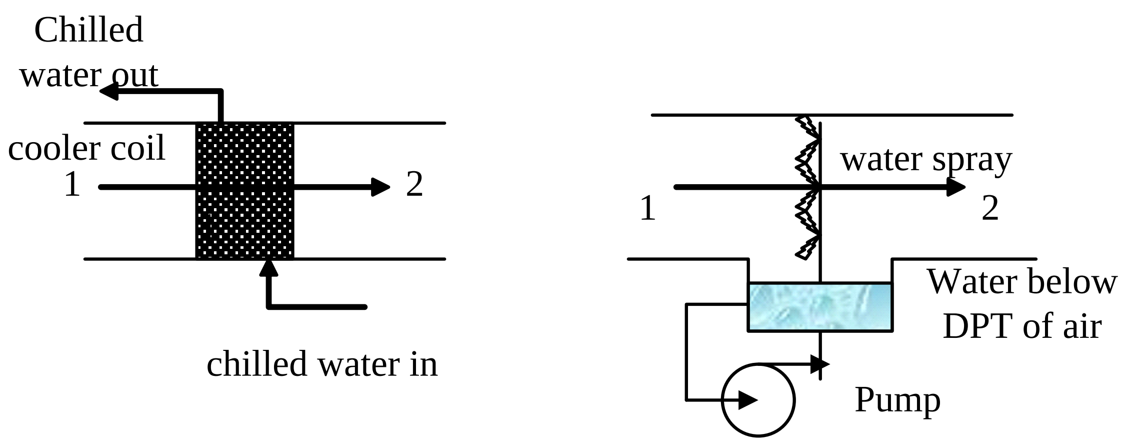


Figure 4 Cooling to a temperature below the dew point

Figure 5 shows a sketch of psychrometric chart what happens when moist air is cooled and dehumidified. Since dehumidification is the aim, some of spray water or same part of cooler coil, must be at a temperature less than dew point temperature of the air entering equipment. The temperature Tc is the ***apparatus dew point***. This term is use for both coils and washer, but in the case of coil alone, Tc is the mean coil surface temperature.

It can be seen from the figure that the moisture content of air is reduced as also is its enthalpy and DBT. The percentage of saturation increases. It might be thought that the point 2 lie on the saturation curve, but there are no cooler coils or air washer have hundred percent efficient. It is unusual to speak of the contact factor β and by by-pass factor $(1-\beta)$.

$$\beta=\frac{g_1-g_2}{g_1-g_c}=\frac{h_1-h_2}{h_1-h_c}=\frac{DBT_1-DBT_2}{DBT_1-T_c}$$

$$1-\beta=\frac{g_2-g_c}{g_1-g_c}=\frac{h_2-h_c}{h_1-h_c}=\frac{DBT_2-DBT_c}{DBT_1-T_c}$$

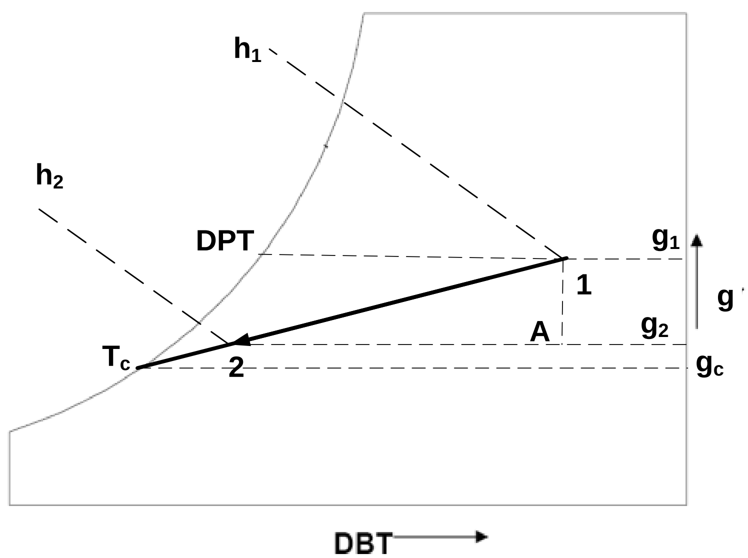


Figure 5 Cooling to a temperature below the dew point

The process is achieved at both sensible (process A-2) and latent heat (process 1-A) rejection, as follows:

$$Q_{s(A-2)}=\dot{m}_a(h_A-h_2)=\dot{m}_a.c_{pa}(DBT_A-DBT_2)$$

$$Q_{l(1-A)}=\dot{m}_a(h_1-h_A)=\dot{m}_a.h_{fg}(g_1-g_2)$$

$$Q_T=Q_s+Q_l=\dot{m}_a(h_2-h_1)\neq\dot{m}_a.c_{pa}(DBT_A-DBT_2)$$

$Q_T=Q_s+Q_l=\dot{m}_a(h_2-h_1)$ $\neq$ **But not equal to** $\dot{m}_a.c_{pa}(DBT_1-DBT_2)$

By separating the total heat transfer rate from the cooling coil into sensible and latent heat transfer rates, a useful parameter called Sensible Heat Factor (SHF) is defined. SHF is defined as the ratio of sensible to total heat transfer rate:

$$SHF=\frac{Q_s}{Q_s+Q_l}=\frac{Q_s}{Q_T}$$

From the above equation, one can deduce that a SHF of 1.0 corresponds to no latent heat transfer and a SHF of 0 corresponds to no sensible heat transfer. A SHF of 0.75 to 0.80 is quite common in air conditioning systems in a normal dry-climate. A lower value of SHF, say 0.6, implies a high latent heat load such as that occurs in a humid climate.

From Fig.6, it can be seen that the slope of the process line 1-2 is given by:

$$\tan(c) = \frac{Q_l}{Q_s} = \frac{\dot{m}_a.h_{fg}(g_1 - g_2)}{\dot{m}_a.c_{pa}(DBT_A - DBT_2)} = \frac{\dot{m}_a.2501(g_1 - g_2)}{\dot{m}_a.1.005(DBT_A - DBT_2)}$$

$$\tan(c) = 2490\frac{(g_1 - g_2)}{(DBT_A - DBT_2)}$$

Thus we can see that the slope of the cooling and de-humidification line is purely a function of the sensible heat factor, SHF. Hence, we can draw the cooling and dehumidification line on psychrometric chart if the initial state and the SHF are known. In some standard psychrometric charts, a protractor with different values of SHF is

provided. The process line is drawn through the initial state point and in parallel to the given SHF line from the protractor as shown in Fig.6

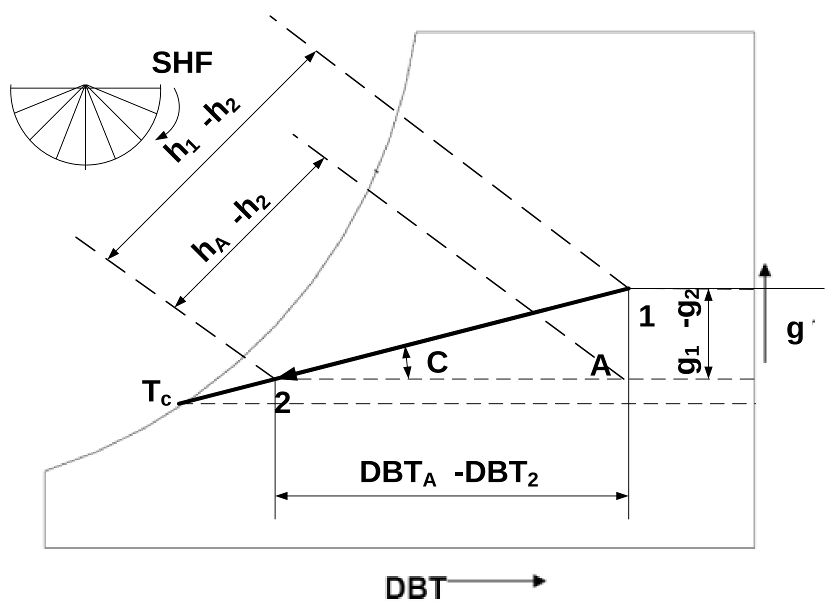


Figure 6 the Sensible Heat Factor (SHF)

Example 3:

1.5 m^3/s of moist air at 28 ^oC DBT, 21 ^oC WBT and 101.325 kPa flow across a cooler coil and leaves at 12.5 ^oC DBT & 8.336 g_w/kg_a. Calculate the apparatus dew point, contact factor and the cooling load.

Tc= 11.5^{o}C h1=60.5 kJ/kg h2=32.5 kJ/kg hc=31.5 kJ/kg from the chart

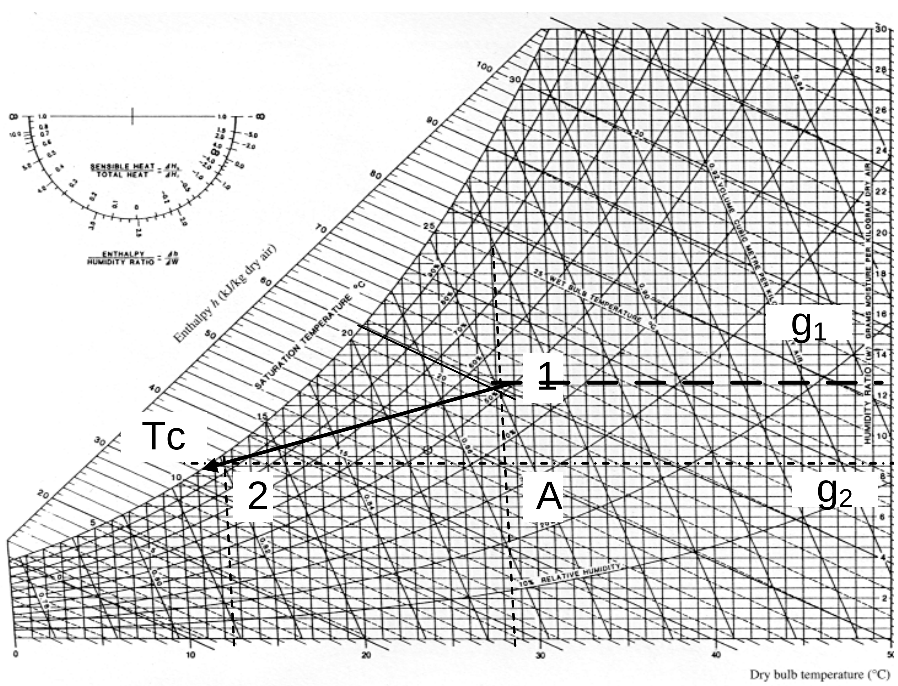
$$\beta = \frac{h1-h2}{h1-hc} = \frac{60.5-32.5}{60.5-31.5} = 0.9665$$

$$1-\beta = 1-0.9665 = 0.034$$

$$cooling\ load = \frac{V}{v_1}(h1-h2) = \frac{1.5}{0.87}(60.5-32.5) = 48.27kW$$

less accurately, one might determine this from the temperature involve:

$$\beta = \frac{DBT_1 - DBT_2}{DBT_1 - Tc} = \frac{28-12.5}{28-11.5} = 0.939$$



Humidification

It is mean that the moisture content of air is increased. This may be accomplished by either water or steam.

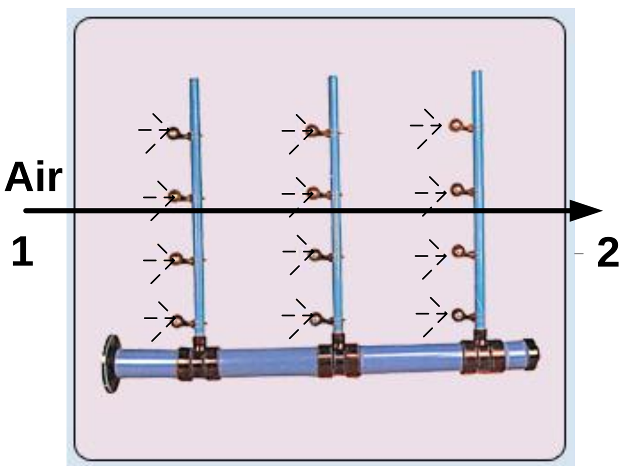
Humidification by water

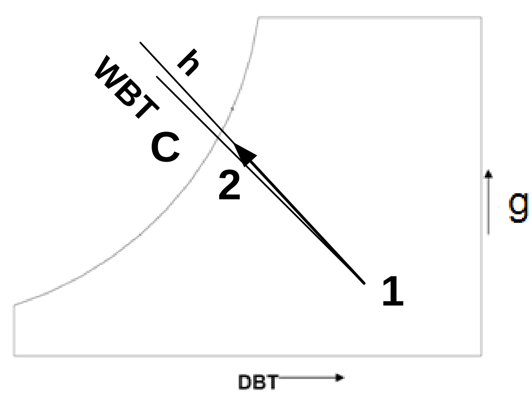
There are three methods of using water as a humidification agent:

- By passing moist air stream through a spray chamber containing a very large number of small water droplets.
- By passing moist air stream through a large wetted surface.
- By direct injection of water droplets aerosol size into the room being conditioned.

passing moist air stream through a spray chamber

In this method a device called an air washer is used, in this method it is customary to speak of *humidification efficiency* or the *effectiveness* ***E*** of an air washer than the contact factor or by a by-pass factor. The effectiveness of air washer may be defined as the extent to which the DBT of the entering moist- air stream approaches it's initial WBT value, or it can be define as the change of state undergoes by the air.





Although humidification efficiency is often expressed in terms of a process of adiabatic saturation, but this process is a special case.

The effectiveness (***E***)of spray chamber is

$$E = \frac{h_2 - h_1}{h_c - h_1} = \frac{g_2 - g_1}{g_c - g_1}$$

And the humidity efficiency η is: $\eta = 100.E$

Consider the special case of adiabatic saturation, for this to occur it is necessary that:

a. The spray water is totally re-circulated, no heat exchange being present in the pipeline or in the waste tank
b. The spray chamber, tank and pipelines are perfectly lugged.
c. The feed water supplied is at the temperature of adiabatic saturation.

Under these conditions it may be assumed that the change of state follows a line of constant enthalpy, and since the lines of enthalpy and WBT are corresponding each other, therefore we can assumed that the process follows the lone of constant WBT.

Example 4:

1.5 m^3/s of moist air at 15^{o}C DBT, 10 ^{o}C WBT and 101.325 kPa enters the spray chamber of an air washer. The humidification efficiency of the washer is 90% all the spray water is re-circulated, the spray chamber and the tank are perfectly lugged, and the feed water at 10 ^{o}C is supplied to make good the losses due to evaporation calculated; a- the state of the air leaving the washer b- the rate of flow of makeup water from the mains.

Since the process is at constant WBT, the point **c** must lie on saturation curve at 10 ^{o}C, i.e. Tc=10 ^{o}C.

$$E = \frac{DBT_1 - DBT_2}{DBT_1 - T_c}$$

$$0.9 = \frac{15 - DBT_2}{15 - 10}$$

$$DBT_2 = 10.5^o C \qquad WBT_2 = WBT_1 = 10^o C$$

$$E = \frac{g_1 - g_2}{g_1 - g_c}$$

$$0.9 = \frac{5.56 - g_2}{5.56 - 7.6}$$

$$g_2 = 7.396 gw/kga$$

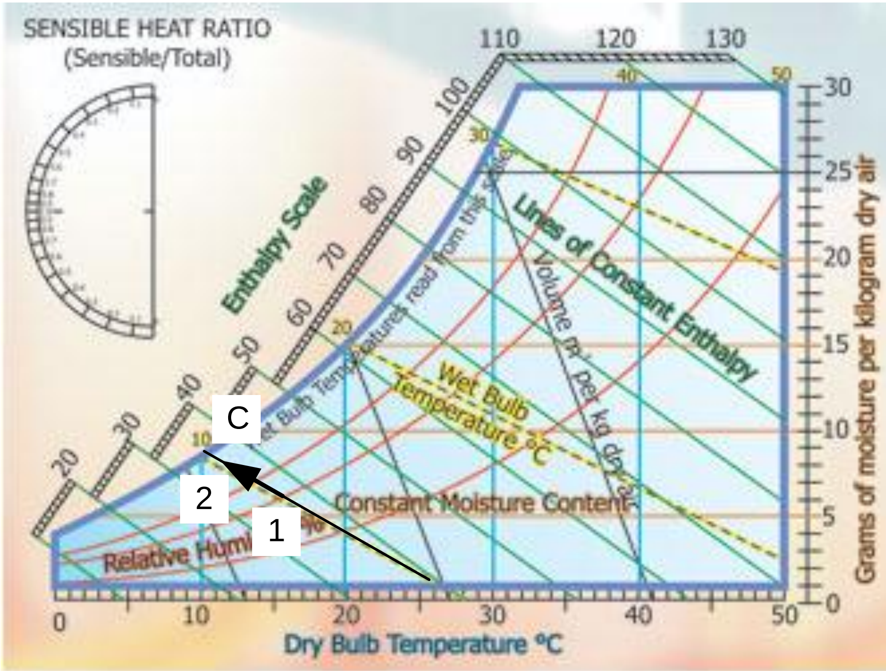
$$the \quad rate\ of\ \text{makeup water} = \dot{m}_a (g_2 - g_1)$$

$$v_1 = 0.824 m^3/kg$$

$$\dot{m}_a = \frac{V}{v} = \frac{1.5}{0.824} = 1.82 kg/s$$

$$\dot{m}_w = 1.82(7.396 - 5.56) = 3.342\ g_w/s$$

$$\dot{m}_w = 3.342 \times 10^{-3} \times 3600 = 12.03 kg_w/hr$$



Water injection;

The simplest case to consider and one that provides the moist insight into change of state of the air stream subjected to humidification by the injection of water is where ***all the injected water is evaporated***. Figure 7 shows what happens when total evaporation occur, air enters a spray chamber, all injected water being evaporated, no falling to the bottom of the chamber to run to waste or to be circulated. The feed water temperature is important to release that, since total evaporation has occurred. State 2, 3 and 4 must be lie nearer to the saturation curve, but just how nearer will depend on the amount of water injected. Two equations, a heat balance and mass balance, provide the answer required:

$h_1 + h_w = h_2$

$m_{a1} + m_w = m_2$

$g_2 = g_a + m_w$ the associated kg of dry air be ignored

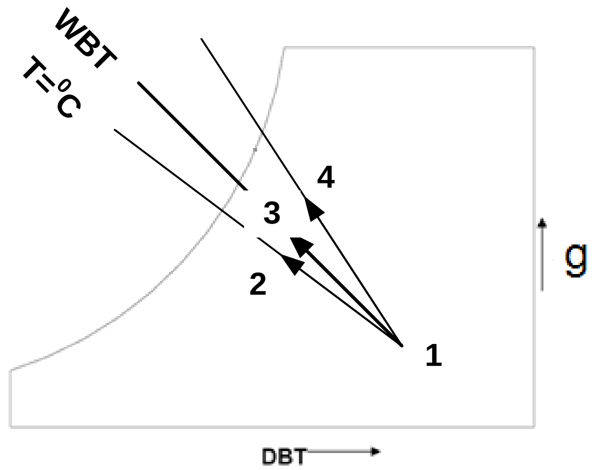


Figure 7 humidification by water injection

m_w : the amount of feed water in kg_w/kg_a flowing through the spray chamber.

Applying the heat balance:

$$h_1 + h_w = h_2 = 1.005DBT_2 + g_2(2501.6 + 1.86DBT_2)$$

One thing is immediately apparent:

- if the feed water is injected at 0°C, the state follows constant enthalpy line, since 0°C is the temperature datum of zero enthalpy for the water associated with 1 kg of dry air.
- If feed water is at a temperature equal to WBT of the air, the state follows constant WBT.
- To see what happens at other water temperature, consider water at 100°C injected into the air stream and totally evaporated.

Example 5:

Moist air at 21 °C DBT, 15 °C WBT and 101.325 kPa enters spray chamber. If for each kg of dry air passing through the chamber, 0.002 kg of water at 100 °C is injected and totally evaporated, calculate the moisture content, enthalpy and DBT of the moist air leaving the chamber.

$h_1 = 42 kJ/kg \qquad g_1 = 8.314 gw/kga$

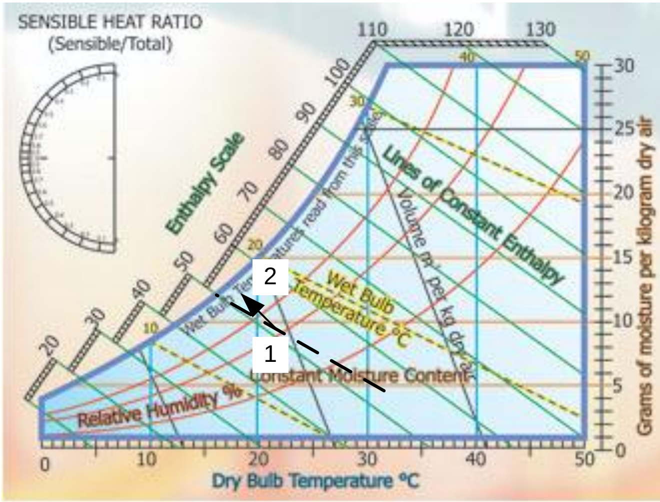
$hw = hf_{100^o C} = 419.06 kJ/kg$

$g_2 = g_1 + m_w = 8.314 + 2 = 10.13 gw/kga$

$h_2 = h_1 + gh_w = 42 + 0.002 \times 419.06 = 42.839 kJ/kg$

$42.839 = 1.005 DBT_2 + 10.13 \times 10^{-3}(2501.3 + 1.86 DBT_2)$

$DBT_2 = 17.09^o C$



From above it can be seen that the condition of line 1,(2,3 and 4) will lie between 1-2(t_w=0^{o}C) and 1-4(t_w=100^{o}C). It follows that for all practical purpose the change of state **for process of so-called adiabatic saturation may be assumed to follow a line of WBT.**

Humidification by steam injection

Steam injection may be dealt with by consideration at a mass and energy balance.

$g_2 = g_1 + m_s$

m_s : mass of steam injected in kg into one kg/s of dry air stream

$h_2 = h_1 + m_s h_s$

The change of state takes place almost along a line of constant DBT between limits defined by smallest and largest enthalpies of injected steam, provided the steam is in dry saturated condition. Figure 8 shows the possibility of steam injection

The lowest possible enthalpy is for dry saturated steam at 100^{o}C, the other extreme is provided by the steam which has maximum enthalpy at 2803 kJ/kg, which is exist at 30 bar and 234^{o}C.

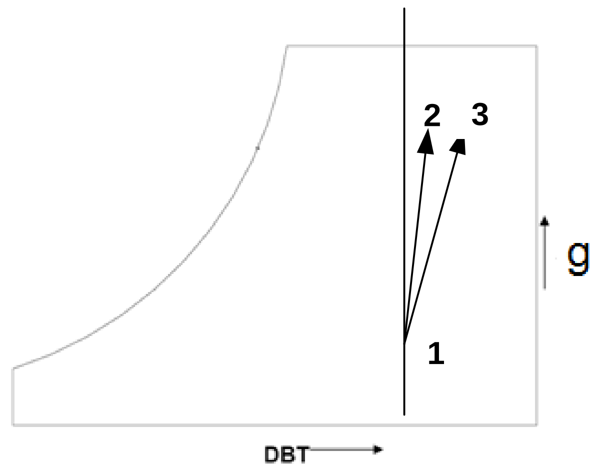


Figure 8 steam injection

Example 6:

Dry saturated steam at 100^{o}C is injected at a rate of 0.01 kg/s into a moist air-stream moving at a rate of 1 kg of dry air per second and initially at a state of 28 ^{o}C DBT, 12 ^{o}C WBT and 101.325 kPa. barometric pressure. Calculate the leaving state of moist air-stream.

from the psychrometric chart h_1=33.2 kJ/kg g1=1.937 g/kga

hg=2675.8 kJ/kg

g2=1.937+10=11.937gw/kga

h2=33.2+0.01x2675.8=59.9kJ/kg

59.9=1.005 DBT_2 + 0.011937(2501.3+1.86DBT_2)

DBT2=29.2^{o}C

Example 7:

Dry saturated steam at with maximum enthalpy is injected at a rate of 0.01 kg/s into a moist air-stream moving at a rate of 1 kg of dry air per second and initially at a state of 28 ^{o}C DBT, 12 ^{o}C WBT and 101.325 kPa barometric pressure. Calculate the leaving state of moist air-stream.

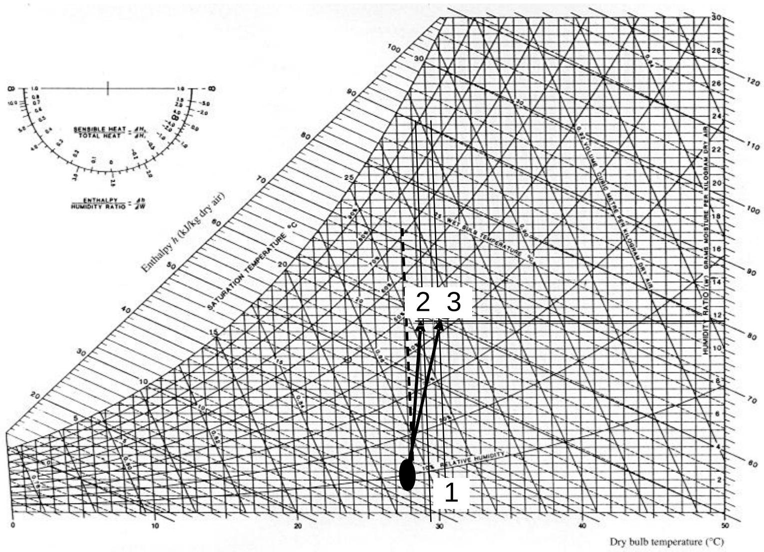
from the psychrometeric chart h_1=33.2 kJ/kg g1=1.937 g/kga

the maximum enthalpy of steam is 2803 kJ/kg at 30 bar and 234^{o}C saturated

h3=33.2+0.01x2803=61.2kJ/kg

61.2=1.005DBT3+0.011937(2501.3+1.86DBT3) DBT3=30.4^{o}C

It can be seen from examples 7 and 8 for the range of states considered, the change in DBT is not very great, so, we can conclude that, the change of state following **steam injection is up a line of constant Dry Bulb Temperature.**



Mixing of air streams:

Mixing of air streams at different states is commonly encountered in many processes, including in air conditioning. Depending upon the state of the individual streams, the mixing process can take place with or without condensation of moisture. Figure 9 shows an adiabatic mixing of two moist air streams during which no condensation of moisture takes place. As shown in the figure, when two air streams at state points 1 and 2 mix, the resulting mixture condition 3 can be obtained from mass and energy balance. From the mass balance of dry air and water vapor:

$$m_{a1} + m_{a2} = m_{a3} \qquad dry - air$$

$$g_1 . m_{a1} + g_2 m_{a2} = g_3 . m_{a3} \qquad associated\ water\ vapour$$

$$g_1 . m_{a1} + g_2 m_{a2} = g_3 . (m_{a1} + m_{a2})$$

$$\frac{m_{a2}}{m_{a1}} = \frac{g_1 - g_2}{g_3 - g_2}$$

or

$$g_3 = \frac{g_1 . m_{a1} + g_2 m_{a2}}{m_{a1} + m_{a2}}$$

making use of conservation of energy:

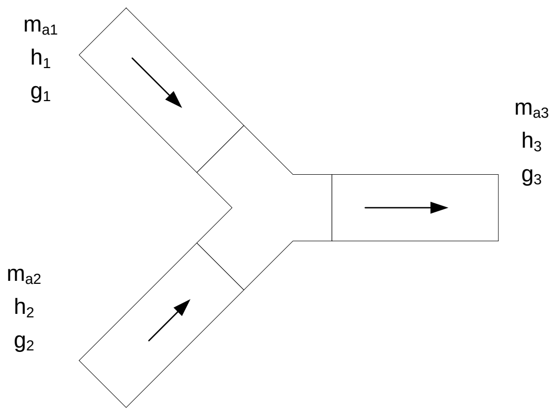
$h_1.m_{a1} + h_2 m_{a2} = h_3.m_{a3}$ *associated water vapour*

$$\frac{m_{a2}}{m_{a1}} = \frac{h_1 - h_2}{h_3 - h_2}$$

or

$$h_3 = \frac{h_1.m_{a1} + h_2 m_{a2}}{m_{a1} + m_{a2}}$$

from this it follows that the three points must be lie on a straight line in a mass - energy co-ordinate system.



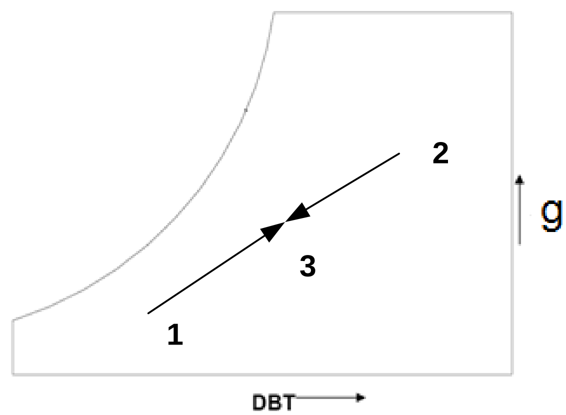


Figure 9 adiabatic mixing of two moist air streams during which no condensation of moisture takes place.

Example 8:

Moist air at state 50oC DBT, 32 oC WBT and 101.325 kPa barometric pressure mixes with moist air at 5 oC DBT, 1 oC WBT and 101.325 kPa barometric pressure. If the mass of dry air are 3kg/s and 2 kg/s respectively. Calculate for the mixture the DBT, enthalpy and moisture content.

a- using equations and chart

b- using psychrometeric chart.

1-air-stream 1

DBT=50 oC WBT=32 oC Pat=101.325kPa

p_{ssw}=4.7556 kPa

p_s=4.7552-101.325x6.66x10-4(50-32)=3.54kPa

$$g_1 = 0.622 \frac{3.54}{101.325 - 3.54} = 0.0225 kgw/kga$$

$$h = 1.005DBT + g(2501.3 + 1.86DBT)$$

h1=1.005x50+0.0225(2501.3+1.86x50)=108.6kJ/kg

2-air-sream2:

DBT=5 ^oC WBT=1 ^oC Pat=101.325kPa

p_{ssw}=0.6566 kPa

p_s=0.6566-101.325x6.66x10-4(5-1)=0.3866kPa

$$g_2 = 0.622\frac{0.3866}{101.325 - 0.3866} = 2.38227 \times 10^{-4}\, kgw/kga$$

$$h = 1.005DBT + g(2501.3 + 1.86DBT)$$

h_2=1.005x5+0.0225(2501.3+1.86x5)=11kJ/kg

3-mixture

$$g_3 = \frac{g_1.m_{a1} + g_2 m_{a2}}{m_{a1} + m_{a2}} = \frac{0.0225 \times 5 + 2.3827 \times 10^{-3} \times 2}{5} = 0.01446\, kgw/kga$$

$$h_3 = \frac{h_1.m_{a1} + h_2 m_{a2}}{m_{a1} + m_{a2}} = \frac{108.6 \times 5 + 11 \times 2}{5} = 69.56\, kJ/kg$$

$$h_3 = 1.005DBT_3 + 0.01446(2501.3 + 1.86DBT_3)$$

$$69.56 = 1.005DBT_3 + 36.168 + 0.0219DBT_3$$

$$DBT_3 = 32.36^o C$$

But if the DBT were calculated by proportion according to the mass of dry air in the two mixing air streams a slightly answer result.

$$DBT_3 = \frac{m_{a1}.DBT_1 + ma_2.DBT_2}{m_{a1} + m_{a2}} = \frac{5 \times 50 + 2 \times 5}{5} = 32^o C$$

This clearly the wrong answer, both numerically and by the method of it's calculation. However the error is small.

The conclusion to be drawn is that the method used to obtain the answer is quite accurate enough for A/C purpose.

Sheet No. Two

1. Which of the following statements are TRUE?

a) During sensible cooling of air, both dry bulb and wet bulb temperatures decrease

b) During sensible cooling of air, dry bulb temperature decreases but wet bulb temperature remains constant

c) During sensible cooling of air, dry and wet bulb temperatures decrease but dew point temperature remains constant

d) During sensible cooling of air, dry bulb, wet bulb and dew point temperatures decrease

Ans.: a) and c)

2. Which of the following statements are TRUE?

a) The sensible heat factor for a sensible heating process is 1.0

b) The sensible heat factor for a sensible cooling process is 0.0

c) Sensible heat factor always lies between 0.0 and 1.0

d) Sensible heat factor is low for air conditioning plants operating in humid climates

Ans.: a) and d)

3. Which of the following statements are TRUE?

a) As the by-pass factor (BPF) of the cooling coil increases, temperature difference between air at the outlet of the coil and coil ADP decreases

b) The BPF of the coil increases as the velocity of air through the coil increases

c) The BPF of the coil increases as the fin pitch increases

d) The BPF of the coil decreases as the number of rows in the flow direction increase

Ans.: b), c) and d)

4. Which of the following statements are TRUE?

a) During cooling and humidification process, the enthalpy of air decreases

b) During cooling and humidification process, the enthalpy of air increases

c) During cooling and humidification process, the enthalpy of air remains constant

d) During cooling and humidification process, the enthalpy of air may increase,
decrease or remain constant depending upon the temperature of the wet surface

Ans.: d)

5. An air stream at a flow rate of 1 kg/s and a DBT of 30°C mixes adiabatically with another air stream flowing with a mass flow rate of 2 kg/s and at a DBT of 15oC. Assuming no condensation to take place, what is the temperature of the mixture.

6. Which of the following statements are TRUE?

a) In an air washer, water has to be externally cooled if the temperature at which it is sprayed is equal to the dry bulb temperature of air

b) In an air washer, water has to be externally heated if the temperature at which it is sprayed is equal to the dry bulb temperature of air

c) In an air washer, if water is simply recirculated, then the enthalpy of air remains nearly constant at steady state

d) In an air washer, if water is simply recirculated, then the moisture content of air remains nearly constant at steady state

Ans.: b) and c)

7. Which of the following statements are TRUE?

a) When the enthalpy of air is equal to the enthalpy of saturated air at the wetted surface temperature, then there is no sensible heat transfer between air and the wetted surface

b) When the enthalpy of air is equal to the enthalpy of saturated air at the wetted surface temperature, then there is no latent heat transfer between air and the wetted surface

c) When the enthalpy of air is equal to the enthalpy of saturated air at the wetted surface temperature, then there is no net heat transfer between air and the wetted surface

d) When the enthalpy of air is equal to the enthalpy of saturated air at the wetted surface temperature, then the wet bulb temperature of air remains constant

Ans.: c) and d)

8. What is the required wattage of an electrical heater that heats 0.1 m^3/s of air from 15°C and 80% RH to 55°C? The barometric pressure is 101.325 kPa.

9. 0.2 kg/s of moist air at 45°C (DBT) and 10% RH is mixed with 0.3 kg/s of moist air at 25°C and a humidity ratio of 0.018 kg_w/kg_a in an adiabatic mixing chamber. After mixing, the mixed air is heated to a final temperature of 40°C using a heater. Find the temperature and relative humidity of air after mixing. Find the heat transfer rate in the heater and relative humidity of air at the exit of heater. Assume the barometric pressure to be 1 atm.

10. A cooling tower is used for cooling the condenser water of a refrigeration system having a heat rejection rate of 100 kW. In the cooling tower air enters at 35^{o}C (DBT) and 24oC (WBT) and leaves the cooling tower at a DBT of 26oC relative humidity of 95%. What is the required flow rate of air at the inlet to the cooling tower in m^3/s. What is the amount of make-up water to be supplied? The temperature of make-up water is at 30oC, at which its enthalpy (h_w) may be taken as 125.4 kJ/kg. Assume the barometric pressure to be 1 atm.

Inside and Outside Design Conditions

The specific objectives of this lecture are to:

1. Describe a typical air conditioning system and discuss the need for fixing suitable indoor and outdoor design conditions.
2. Discuss the criteria used for selecting inside design conditions
3. Define thermal comfort, metabolic rate and response of human beings to variation in body temperature
4. Present heat balance equation, equations for convective, radiative and evaporative losses from the skin, metabolic rates for various types of activities and discuss the thermo-regulatory mechanism used by human body to fight against heat and cold .
5. Discuss the factors affecting thermal comfort .
6. Discuss the various thermal indices used for evaluating indoor environment and present ASHRAE comfort chart, recommended inside design conditions and discuss the concept of Predicted Mean Vote (PMV) and Percent of People Dissatisfied (PPD).
7. Discuss the criteria used for selecting outside design conditions and present typical summer design conditions for major Indian cities as suggested by ASHRAE.

At the end of the lecture, the student should be able to:

1. Explain the need for selecting design inside and outside conditions with respect to a typical air conditioning system
2. Define thermal comfort, metabolism, metabolic rate and discuss the effects of variation in body temperatures on human beings
3. Write the heat balance and heat transfer equations from a human body and using these equations, estimate various heat transfer rates
4. List the factors affecting thermal comfort
5. Define the various thermal indices used in evaluating indoor environment
6. Draw the ASHRAE comfort chart and mark the comfort zones for summer and winter conditions
7. Select suitable indoor design conditions based on comfort criteria
8. Define PMV and PPD and explain their significance

9. Explain the method followed for selecting suitable outside design conditions

1. Introduction:

Design and analysis of air conditioning systems involves selection of suitable inside and outside design conditions, estimation of the required capacity:

- of cooling or heating equipment,
- selection of suitable cooling/heating system,
- selecting supply conditions,
- design of air transmission and distribution systems etc.

Generally, the inputs are the building specifications and its usage pattern and any other special requirements. Figure .1 shows the schematic of a basic summer air conditioning system. As shown in the figure, under a typical summer condition, the building gains sensible and latent heats from the surroundings and also due to internal heat sources (RSH and RLH).

In general, the sensible and latent heat transfer rates (GSH and GLH) on the cooling coil are larger than the building heat gains due to the need for ventilation and return duct losses. The building heat gains depend on:

- the type of the building,
- outside conditions and
- the required inside conditions.

Hence selection of suitable inside and outside design conditions is an important step in the design and analysis of air conditioning systems.

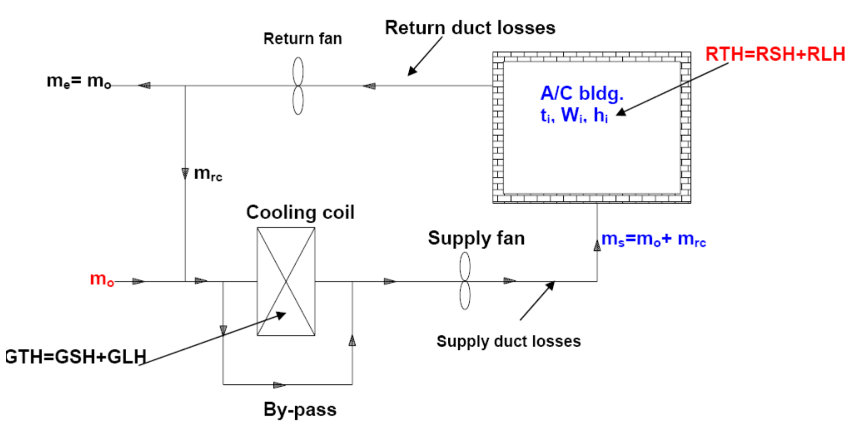


Fig..1: Schematic of a basic summer air conditioning system

2. Selection of inside design conditions:

The required inside design conditions depend on the intended use of the building. Air conditioning is required either for :

- providing suitable comfort conditions for the occupants (e.g. comfort air conditioning),
- or for providing suitable conditions for storage of perishable products (e.g. in cold storages) or
- conditions for a process to take place or for products to be manufactured (e.g. industrial air conditioning).

The required inside conditions for cold storage and industrial air conditioning applications vary widely depending on the specific requirement. However, the required inside conditions for comfort air conditioning systems remain practically same irrespective of the size, type, location, use of the air conditioning building etc., as this is related to the thermal comfort of the human beings.

3. Thermal comfort:

Thermal comfort is defined as "***that condition of mind which expresses satisfaction with the thermal environment***". This condition is also sometimes called as "neutral condition", though in a strict sense, they are not necessarily same. A living human body may be likened to a heat engine in which the chemical energy contained in the food it consumes is continuously converted into work and heat.

"metabolism" : Is the process of conversion of chemical energy contained in food into heat and work .

"metabolic rate": Is the rate at which the chemical energy is converted into heat and work

thermal efficiency of a human being: Is the ratio of useful work output to the energy input.

<u>The thermal efficiency of a human being can vary from 0% to as high as 15-20% for a short duration</u>.

thermal efficiency of 0%. Irrespective of the work output, a human body continuously generates heat at a rate varying from about 100 W (e.g. for a sedentary person) to as high as 2000 W (e.g. a person doing strenuous exercise). Continuous heat generation is essential, as the temperature of the human body has to be maintained within a narrow range of temperature, irrespective of the external surroundings.

A human body is very sensitive to temperature. The body temperature must be maintained within a narrow range to avoid discomfort, and within a somewhat wider range, to avoid danger from heat or cold stress. Studies show that at neutral condition, the temperatures should be:

$$\text{Skin temperature, } t_{skin} \approx 33.7^{o}\text{C}$$

$$\text{Core temperature, } t_{core} \approx 36.8^{o}\text{C}$$

At other temperatures, the body will feel discomfort or it may even become lethal. It is observed that when the core temperature is between:

- **35 to 39^{o}C**, the body experiences only a mild discomfort.
- When the temperature is **lower than 35^{o}C or higher than 39^{o}C**, then people suffer major loss in efficiency.
- It becomes lethal when the temperature **falls below 31^{o}C or rises above 43^{o}C**.

This is shown in Fig. 2.

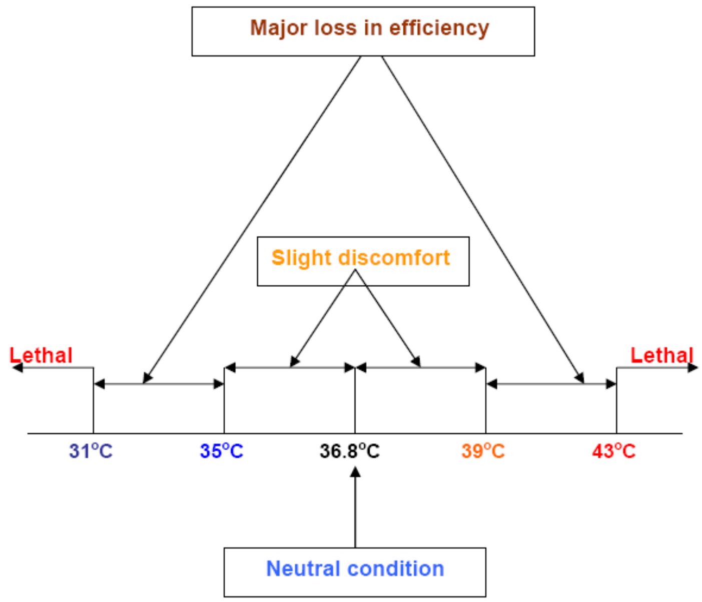


Fig.2: Effect of the variation of core temperature on a human being

4: Heat balance equation for a human being:

The temperature of human body depends upon the energy balance between itself and the surrounding thermal environment. Taking the human body as the control volume, one can write the thermal energy (heat) balance equation for the human body as:

$$Q_{gen} = Q_{sk} + Q_{res} + Q_{st} \qquad (1)$$

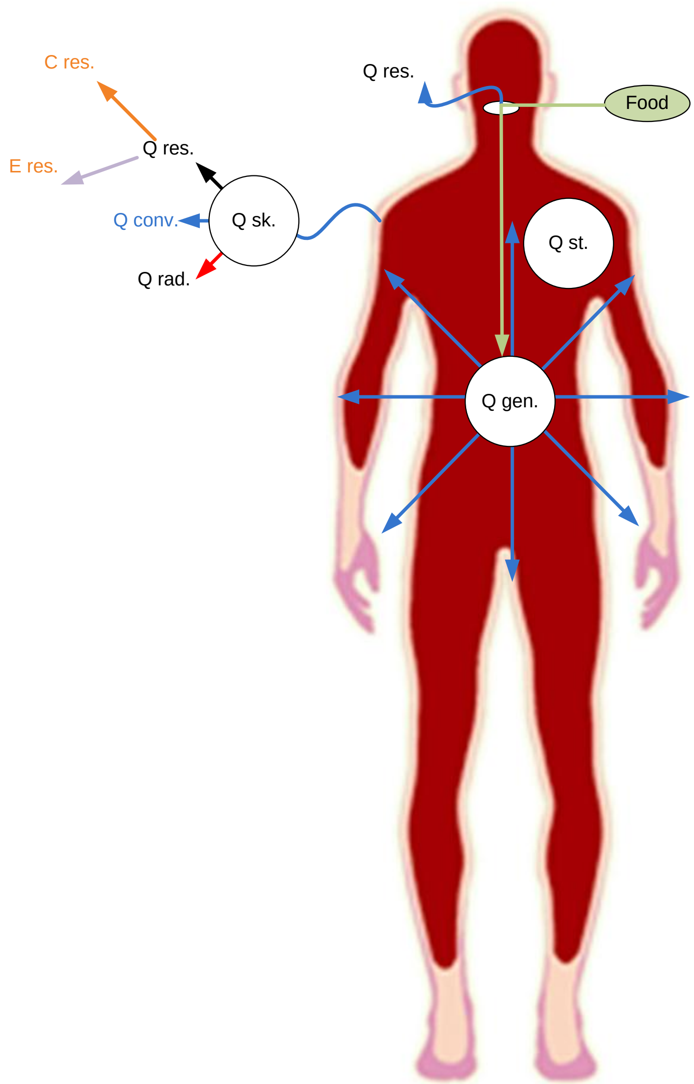
Where

Q_{gen} = Rate at which heat is generated inside the body

Q_{sk} = Total heat transfer rate from the skin

Q_{res} = Heat transfer rate due to respiration,

and Q_{st} = Rate at which heat is stored inside the body



The heat generation rate Q_{gen} is given by:

$$Q_{gen} = M(1-\eta) \approx M \quad (2)$$

where M = Metabolic rate, and

η = Thermal efficiency ≈ 0 for most of the activities

The metabolic rate depends on the activity. It is normally measured in the unit **"met".** A met is defined as the metabolic rate per unit area of a sedentary person and is found to be equal to about **58.2 W/m^2.** This is also known as "**basal metabolic rate**".

Studies show that the metabolic rate can be correlated to the rate of :

1. respiratory oxygen consumption and
2. carbon dioxide production.

Even though the metabolic rate and heat dissipation are not uniform throughout the body, for calculation purposes they are assumed to be uniform.

Table .1 shows typical metabolic rates for different activities:

Activity	Specifications	Metabolic rate
Resting	Sleeping	0.7 met
	Reclining	0.8 met
	Seated, quite	1.0 met
	Standing, relaxed	1.2 met
Walking	0.89 m/s	2.0 met
	1.79 m/s	3.8 met
Office activity	Typing	1.1 met
Driving	Car	1.0 to 2.0 met
	Heavy vehicles	3.2 met
Domestic activities	Cooking	1.6 to 2.0 met
	Washing dishes	1.6 met
	House cleaning	2.0 to 3.4 met
Dancing	-	2.4 to 4.4 met
Teaching	-	1.6 met
Games and sports	Tennis, singles	3.6 to 4.0 met
	Gymnastics	4.0 met
	Basket ball	5.0 to 7.6 met
	Wrestling	7.0 to 8.7 met

The human body is considered to be a cylinder with uniform heat generation and dissipation. The surface area over which the heat dissipation takes place is given by an empirical equation, called as **Du Bois Equation**. This equation expresses the surface area as a function of the mass and height of the human being. It is given by:

$A_{Du} = 0.202\ m^{0.425} h^{0.725}$ (3)

where

A_{Du} = Surface area of the naked body, m^2

m = Mass of the human being, kg

h = Height of the human being, m

Since the area given by Du Bois equation refers to a naked body, a correction factor must be applied to take the clothing into account. This correction factor, defined as the **"*ratio of surface area with clothes to surface area without clothes*"** has been determined for different types of clothing. These values are available in ASHRAE handbooks. Thus from the metabolic rate and the surface area, one can calculate the amount of heat generation, Q_{gen}.

The total heat transfer rate from the skin Q_{sk} is given by:

$Q_{sk} = \pm Q_{conv} \pm Q_{rad} + Q_{evp}$ (4)

where :

Q_{conv} = Heat transfer rate due to convection (sensible heat) can be positive or negative

Q_{rad} = Heat transfer rate due to radiation (sensible heat), can be positive or negative

and Q_{evp} = Heat transfer rate due to evaporation (latent heat) always positive

According to Belding and Hatch, the convective, radiative and evaporative heat transfer rates from the naked body of an average adult, Qconv, Qrad and Qevap, respectively, are given by:

$Qconv = 14.8V^{0.5}(t_b - t)$

$Q_{rad} = 11.603\,(t_b - t_s)$

$Q_{evap} = 181.76V^{0.4}(p_{s,b} - p_v)$ **5**

Where:

t_b : the average temperature of the surface of the human being ^oC

t : the indoor dry bulb temperature ^oC

$t_{s:}$ the mean radiant temperature ^oC

p_{sb} : saturated pressure of water vapour at surface temperature of the body kPa

$p_{v:}$ water vapour in air in kPa.

v: air velocity in m/s

In the above equation all the heat transfer rates are in watts, temperatures are in ^{o}C and velocity is in m/s.

From the above equations it is clear that the convective heat transfer from the skin can be increased:

- either by increasing the surrounding air velocity (V)
- and/or by reducing the surrounding air DBT (t).

The radiative heat transfer rate can be increased by reducing the temperature of the surrounding surfaces with which the body exchanges radiation.

The evaporative heat transfer rate can be increased by “

- increasing the surrounding air velocity

- and/or by reducing the moisture content of surrounding air.

The heat transfer rate due to respiration Q_{res} is given by:

$Q_{res} = C_{res} + E_{res}$ (5)

Where

C_{res} = Dry heat loss from respiration (sensible, positive or negative)

E_{res}= Evaporative heat loss from respiration (latent, always positive)

heat transfer rate due to respiration can be calculated from equation 5

$C_{res} + E_{res} = 0.0014M(34 – t_a) + 0.0173M\,(5.87 – p_a)$

Where:

t_a ambient temperature ^oC

p_a water vapor pressure in ambient air, kPa

For comfort, the rate of heat stored in the body Q_{st} should be zero, i.e.,

$Q_{st} = 0$ at neutral condition (6)

However, it is observed that a human body is rarely at steady state, as a result the rate of heat stored in the body is non-zero most of the time.

Depending upon the surroundings and factors such as activity level etc., the heat stored is either positive or negative.

All living beings have **in-built body regulatory processes against cold and heat,** which to some extent maintains the body temperatures when the external conditions are not favorable. For example, human beings consist of a thermoregulatory system, which tries to maintain the body temperature by initiating certain body regulatory processes against cold and heat.

When the **environment is colder** than the neutral zone, then body loses more heat than is generated. Then the regulatory processes occur in the following order.

1. Zone of vaso-motor regulation against cold (vaso-constriction): Blood vessels adjacent to the skin constrict, reducing flow of blood and transport of heat to the immediate outer surface. The outer skin tissues act as insulators.

2. Zone of metabolic regulation: If environmental temperature drops further, then vaso-motor regulation does not provide enough protection. Hence, through a spontaneous increase of activity and by shivering, body heat generation is increased to take care of the increased heat losses.

3. Zone of inevitable body cooling: If the environmental temperature drops further, then the body is not able to combat cooling of its tissues. Hence the body temperature drops, which could prove to be disastrous. This is called as zone of inevitable body cooling.

When the **environment is hotter** than the neutral zone, then body loses less heat than is generated. Then the regulatory processes occur in the following order.

1. Zone of vaso-motor regulation against heat (vaso-dilation): Here the blood vessels adjacent to the skin dilate, increasing the flow of blood and transport of heat to the immediate outer surface. The outer skin temperature increases providing a greater temperature for heat transfer by convection and radiation.

2. Zone of evaporative regulation: If environmental temperature increases further, the sweat glands become highly active drenching the body surface with perspiration. If the surrounding air humidity and air velocity permit, then increase in body temperature is prevented by increased evaporation from the skin.

3. Zone of inevitable body heating: If the environmental temperature increases further, then body temperature increases leading to the zone of inevitable body heating. The internal body temperature increases leading several ill effects such as heat exhaustion (with symptoms of fatigue, headache, dizziness, irritability etc.), heat cramps (resulting in loss of body salts due to increased perspiration) and finally heat stroke. Heat stroke could cause permanent damage to the brain or could even be lethal if the body temperature exceeds 43^oC.

A sedentary person at neutral condition loses about

- **40 % of heat by evaporation,**
- **about 30 % by convection and**
- **30 % by radiation.**

5. Factors affecting thermal comfort:

Thermal comfort is affected by several factors. These are:

1. Physiological factors such as age, activity, sex and health. These factors influence the metabolic rate. It is observed that of these factors, the most important is activity. Other factors are found to have negligible effect on thermal comfort.

2. Insulating factor due to clothing. The type of clothing has strong influence on the rate of heat transfer from the human body. The unit for measuring the resistance offered by clothes is called as **"clo".** 1 clo is equal to a resistance of about 0.155 $m^2.K/W$. Typical clo values for different types of clothing have been estimated and are available in the form of tables. For example, a typical business suit has a clo value of 1.0, while a pair of shorts has a clo value of about 0.05.

3. Environmental factors. Important factors are the dry bulb temperature, relative humidity, air motion and surrounding surface temperature. Of these the dry bulb temperature affects heat transfer by convection and evaporation, the relative humidity affects heat loss by evaporation, air velocity influences both convective and evaporative heat transfer and the surrounding surface temperature affects the radiative heat transfer.

6. Indices for thermal comfort:

These **Indices for thermal comfort** can be divided into direct and derived indices.

The direct indices are

- the dry bulb temperature,
- humidity ratio,
- air velocity
- and the mean radiant temperature (T_{mrt}).

The mean radiant temperature T_{mrt} affects the radiative heat transfer and is defined (in K) as:

$$T_{mrt}^4 = (T_g + 273)^4 + 0.247 \times 10^9 . V^{\frac{1}{2}} . \{(T_g - T_a\} \qquad 7$$

where:

T_{mrt} = The mean radiant temperature in K

T_g = Globe temperature measured at steady state by a thermocouple placed at the center of a black painted, hollow cylinder (6" dia) kept in the conditioned space ^oC,

Ta = Ambient DBT, ^oC

V = Air velocity in m/s, and

Air Temperatures and Mean Radiant Temperatures Necessary for Comfort (PMV = 0) of Sedentary Persons in Summer Clothing at 50% rh is shown in figure 3.

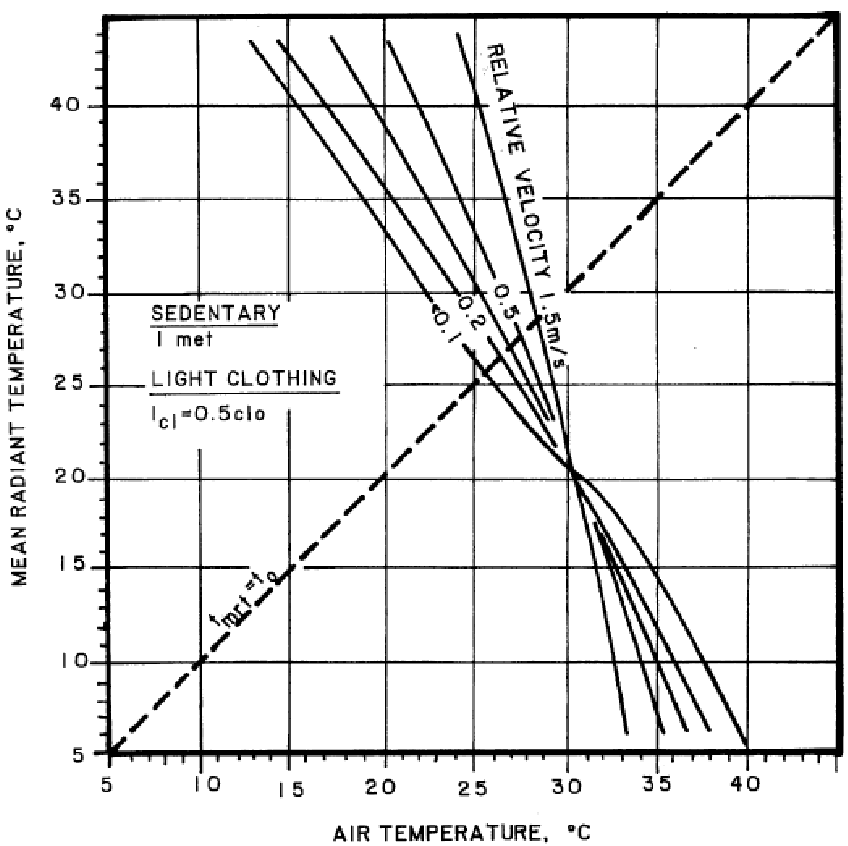


Figure 3 Air Temperatures and Mean Radiant Temperatures Necessary for Comfort (Predicted Mean Vote PMV = 0) of Sedentary Persons in Summer Clothing at 50% rh

Effective temperature (ET): This factor combines the effects of dry bulb temperature and air humidity into a single factor. ***It is defined as the temperature of the environment***

at 50% RH which results in the same total loss from the skin as in the actual environment. Since this value depends on other factors such as activity, clothing, air velocity and T_{mrt}, a Standard Effective Temperature (SET) is defined for the following conditions:

Clothing = 0.6 clo

Activity = 1.0 met

Air velocity = 0.1 m/s

Tmrt = DBT (in K)

The basic effective temperature can found from figure 4

Operative temperature (Top): This factor is a weighted average of air DBT and Tmrt into a single factor. It is given by:

$$T_{op} = \frac{h_r . T_{mrt} + h_c . t_a}{h_r + h_c} \approx \frac{T_{mrt} + t_a}{2} \qquad 8$$

where h_r and h_c are the radiative and convective heat transfer coefficients and t_a is the DBT of air.

ASHRAE has defined a comfort chart based on the effective and operative temperatures. Figure .5 shows the ASHRAE comfort chart with comfort zones for summer and winter conditions. It can be seen from the chart that the comfort zones are bounded by effective temperature lines, a constant RH line of 60% and dew point temperature of 2°C. The upper and lower limits of humidity (i.e. 60 % RH and 2^oC DPT, respectively) are based on the moisture content related considerations of dry skin, eye irritation, respiratory health and microbial growth. The comfort chart is based on statistical sampling of a large number of occupants with activity levels less than 1.2 met. On the chart, the region where summer and winter comfort zones overlap, people in winter clothing feel slightly warm and people in summer clothing feel slightly cool. Based on the chart ASHARE makes the following recommendations:

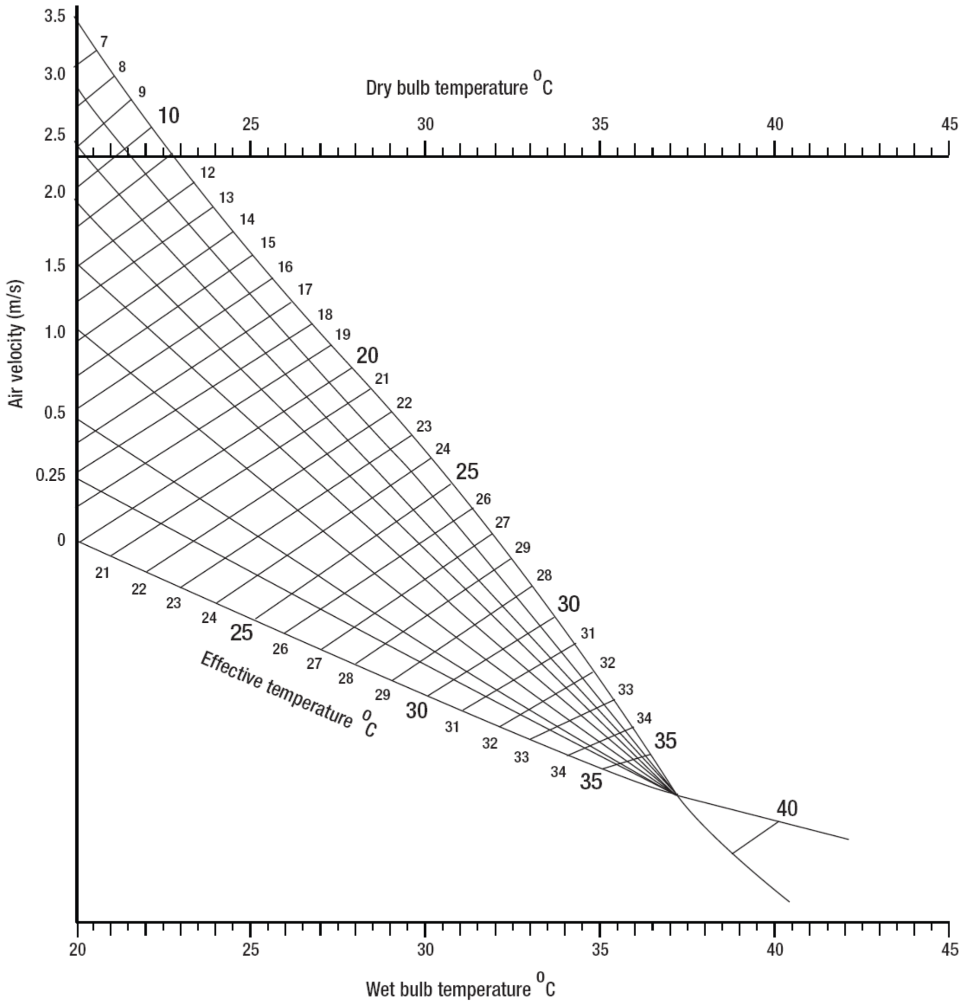


Figure 4 basic effective temperature

Inside design conditions for Winter:	Inside design conditions for Summer:
Top between 20.0 to 23.5°C at a RH of 60%	**Top between 22.5 to 26.0°C at a RH of 60%**
Top between 20.5 to 24.5°C at a DPT of 2°C	**Top between 23.5 to 27.0°C at a DPT of 2°C**

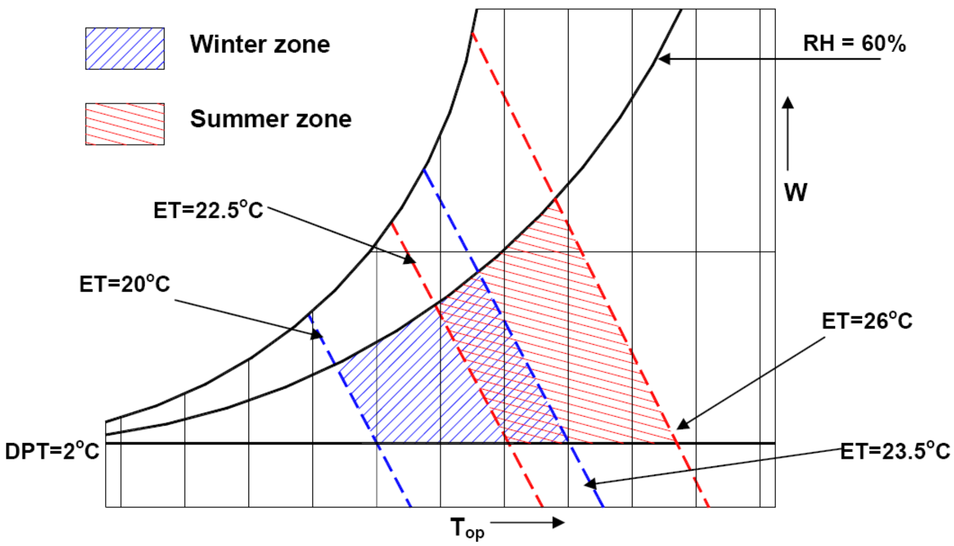


Fig. 5: ASHRAE comfort chart for a sedentary person (activity. ≈1.2 met)

Table 2 shows the recommended comfort conditions for different seasons and clothing suitable at 50 % RH, air velocity of 0.15 m/s and an activity level of ≤1.2 met

Table .2: Optimum and recommended operative temperatures for comfort

season	Clothing	I_{cl} clo	$T_{op\ opt.}$	**T_{op} range for 90% acceptance**
Winter	Heavy slacks, long sleeve shirt and sweater	0.9	**22°C**	**20 to 23.5 °C**
summer	Light slacks and short sleeve shirt	0.5	**24.5°C**	**23 to 26°C**
	Minimal (shorts)	0.05	**27°C**	**26 to °C**

The air conditioning systems should be operated at :

- as low a temperature as acceptable in winter
- and as high a temperature as acceptable in summer.

Use of suitable clothing and maintaining suitable air velocities in the conditioned space can lead to reduced cost of air conditioning.

These special considerations must be kept in mind while fixing the inside design conditions. **Prof. P.O. Fanger** of Denmark has carried out pioneering and detailed studies on thermal comfort and suggested comfort conditions for a wide variety of situations.

6.1. Predicted Mean Vote (PMV) and Percent People Dissatisfied (PPD):

Based on the studies of Fanger and subsequent sampling studies, ASHRAE has defined a thermal sensation scale, which considers the air temperature, humidity, sex of the occupants and length of exposure. The scale is based on empirical equations relating the above comfort factors. ***The scale varies from +3 (****hot****) to -3 (****cold****) with 0 being the neutral condition***.

+3	+2	+1	0	-1	-2	-3
Hot	warm	slightly warm	neutral	slightly cool	cool	cold

Then a Predicted Mean Vote (PMV) that predicts the mean response of a large number of occupants is defined based on the thermal sensation scale.

The PMV is defined by Fanger as:

$$\mathrm{PMV} = \left[0.303\ \mathrm{e}^{[-(0.036\mathrm{M}+0.028)]}\right].L \qquad 9$$

where M is the metabolic rate and L is the thermal load on the body that is the difference between the internal heat generation and heat loss to the actual environment of a person experiencing thermal comfort. The thermal load has to be obtained by solving the heat balance equation for the human body.

Fanger related the PMV to Percent of People Dissatisfied (PPD) by the following equation:

$$\mathrm{PPD} = 100 - 95\ \mathrm{e}^{[-(0.03353\mathrm{PMV}^4+0.2179\mathrm{PMV}^2)]} \qquad 10$$

where dissatisfied refers to anybody not voting for -1, 0 or +1. It can be seen from equation 10 and figure 4 that even when the PMV is zero (i.e., no thermal load on body) 5 % of the people are dissatisfied! When PMV is within ± 0.5, then PPD is less than 10 %.

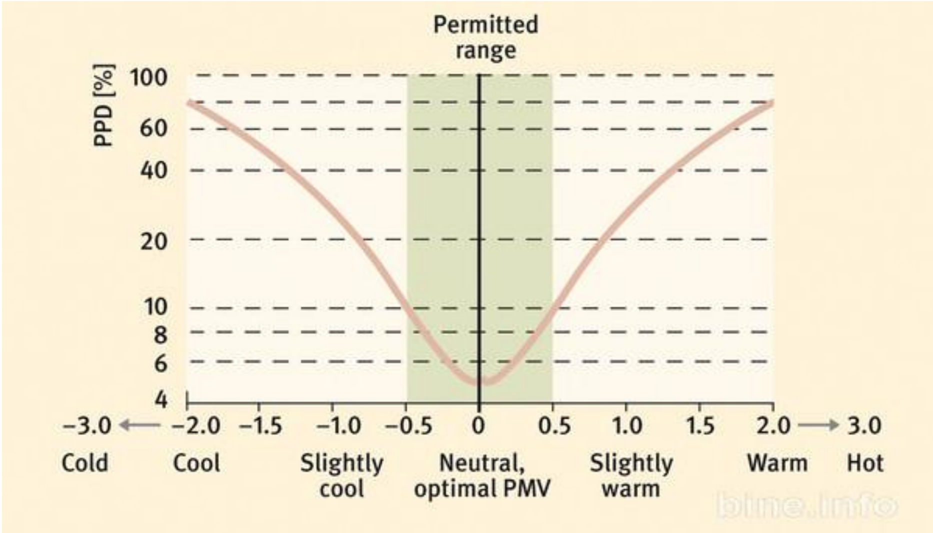


Figure 4 Percentage of room users that are dissatisfied with the existing indoor environment conditions (PPD index – predicted percentage dissatisfied) as a function of the user's average subjective evaluation of the environment (PMV index – predicted mean vote) as suggested by Fanger.

7. Selection of outside design conditions:

It is obvious that the selected design conditions may prevail only for a short a duration, and most of the time the actual outside conditions will be different from the design values. As a result, for most of the time the plant will be running at off-design conditions.

The design outside conditions also depend on the following factors:

a) Type of the structure, i.e., whether it is of heavy construction, medium or light
b) Insulation characteristics of the building
c) Area of glass or other transparent surfaces
d) Type of usage
e) Nature of occupancy
f) Daily range (difference between maximum and minimum temperatures in a given day).

7.1. Outdoor design conditions for summer:

Selection of maximum dry and wet bulb temperatures at a particular location leads to excessively large cooling capacities as the maximum temperature generally persists for only a few hours in a year. Hence it is recommended that the outdoor design conditions for summer be chosen based on the values of dry bulb and mean coincident wet bulb temperature that is equaled or exceeded 0.4, 1.0 or 2.0 % of total hours in an year. These values for major locations in the world are available in data books, such as AHRAE handbooks. Whether to choose the 0.4 % value or 1.0 % value or 2.0 % value depends on specific requirements. In the absence of any special requirements, the 1.0% or 2% value may be considered for summer outdoor design conditions.

7.2. Outdoor design conditions for winter:

Similar to summer, it is not economical to design a winter air conditioning for the worst condition on record as this would give rise to very high heating capacities. Hence it is recommended that the outdoor design conditions for winter be chosen based on the values of **dry bulb temperature** that is **equaled or exceeded 99.6 or 99.0 % of total hours in a year.** Similar to summer design conditions, these values for major locations in the world are available in data books, such as AHRAE handbooks. Generally the 99.0% value is adequate, but if the building is made of light-weight materials, poorly insulated or has considerable glass or space temperature is critical, then the 99.6% value is recommended.

Table .3 shows the ASHRAE recommended summer design conditions for some Iraqi cities.

Table 3 Outdoor design Conditions For Baghdad and Mosel

Governor	Lat. o	Long. o	Winter		Summer						Daily rang ^oC
			DBT ^oC		DBT ^oC			WBT ^oC			
			97.5%	99%	1%	2.5%	5%	1%	2.5%	5%	
Baghdad	$33^o20^/N$	$44^o24^/E$	1.5	0	42	44	45	22	22	23	19
Mosel	$36^o19^/N$	$43^o09^/E$	0	-1.5	43.5	44.5	45.5	22	22	23	22

Human race lives in a hostile environment. The degree of hostility varies from season to season and with the geographical area .For this reason mankind has to provide a

8- Weather and climate

8-1 How does climate differ from weather?

8-1-1 Weather is the current atmospheric conditions, including temperature, rainfall, wind, and humidity at a given place. If you stand outside, you can see that it's raining or windy, or sunny or cloudy. You can tell how hot it is by taking a temperature reading. Weather is what's happening right now or is likely to happen tomorrow or in the very near future.

8-1-2 Climate, on the other hand, is the general weather conditions over a long period of time. For example, on any given day in January, we expect it to be rainy in Portland, Oregon and sunny and mild in Phoenix, Arizona. And in Buffalo, New York, we're not surprised to see January newscasts about sub-zero temperatures and huge snow drifts.

Some meteorologists say that "climate is what you expect and weather is what you get." According to one middle school student, "climate tells you what clothes to buy, but weather tells you what clothes to wear."

Climate is sometimes referred to as "average" weather for a given area. The National Weather Service uses data such as temperature highs and lows and precipitation rates for the past thirty years to compile an area's "average" weather. However, some atmospheric scientists think that you need more than "average" weather to accurately portray an area's climatic character - variations, patterns, and extremes must also be included. Thus, climate is the sum of all statistical weather information that helps describe a place or region. The term also applies to large-scale weather patterns in time or space such as an 'Ice Age' climate or a 'tropical' climate.

8-2 Climate Variability

Although an area's climate is always changing, the changes do not usually occur on a time scale that's immediately obvious to us. While we know how the weather changes from day to day, subtle climate changes are not as readily detectable. Weather patterns and climate types take similar elements into account, the most important of which are:

- The temperature of the air
- The humidity of the air
- The type and amount of cloudiness
- The type and amount of precipitation

- Air pressure
- Wind speed and direction

8-3 What is the greenhouse effect?

The greenhouse effect is a naturally occurring phenomenon that blankets the earth and warms it, maintaining the temperature that living things need to survive. Surprisingly, the atmosphere's most abundant gases — nitrogen, oxygen, and argon — do not influence climate. Instead, it's the molecules of trace gases, especially water vapor (H2O), carbon dioxide (CO_2), methane (CH_4), nitrous oxide (N_2O) and ozone (O_3) that strongly absorb infra-red radiation contained in sunlight, or emitted by land and water as they cool Just as greenhouses keep the air inside them warm, so water vapor and trace gases keep Earth about 12º C warmer than it would be without them. This retention of heat is called the greenhouse effect and the gases that cause it are known as greenhouse gases.

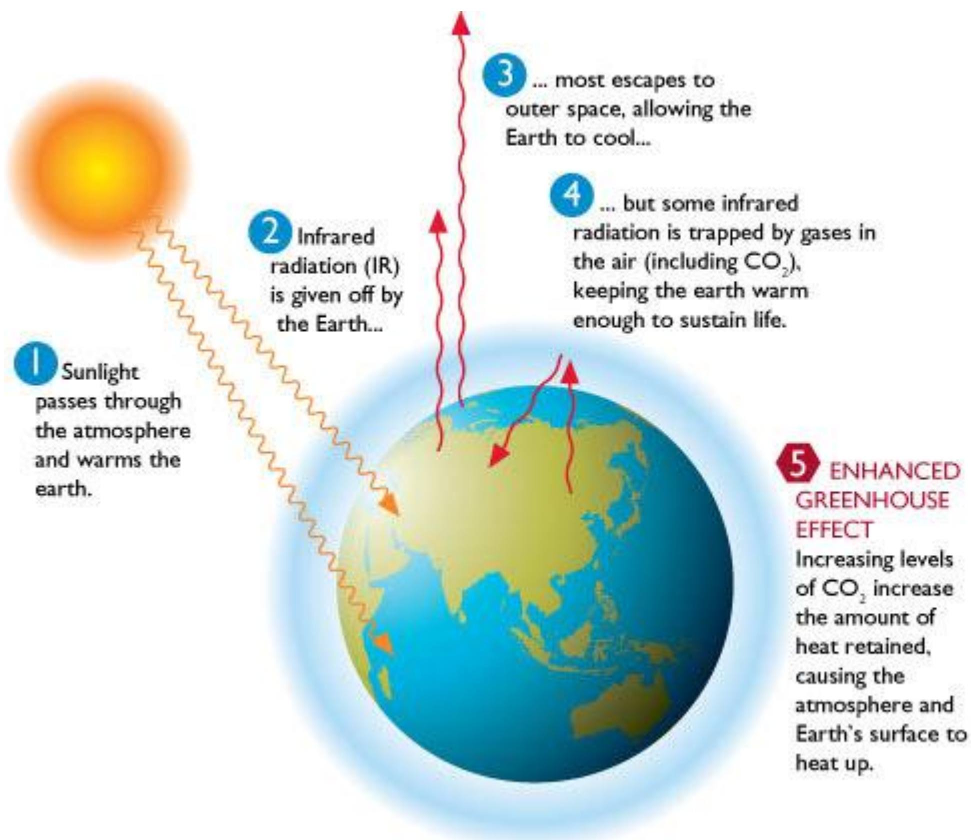


Figure greenhouse effect

8-4 What is the Ozone depletion?

Three forms (or allotropes) of oxygen are involved in the ozone-oxygen cycle: oxygen atoms (O or atomic oxygen), oxygen gas (O_2 or diatomic oxygen), and ozone gas (O_3 or triatomic oxygen). Ozone is formed in the stratosphere when oxygen molecules photodissociate after absorbing an ultraviolet photon whose wavelength is shorter than 240 nm. This converts a single O_2 into two atomic oxygen ions. The atomic oxygen ions then combine with separate O_2 molecules to create two O_3 molecules. These ozone molecules absorb UV light between 310 and 200 nm, following which ozone splits into a molecule of O_2 and an oxygen atom. The oxygen atom then joins up with an oxygen molecule to regenerate ozone. This is a continuing process which terminates when an oxygen atom "recombines" with an ozone molecule to make two O_2 molecules.

$O + O_3 \rightarrow 2\ O_2$

The overall amount of ozone in the stratosphere is determined by a balance between photochemical production and recombination. Ozone can be destroyed by a number of free radical catalysts, the most important of which are the hydroxyl radical (OH·), the nitric oxide radical (NO·), the atomic chlorine ion (Cl·) and the atomic bromine ion (Br·). All of these have both natural and man-made sources; at the present time, most of the OH· and NO· in the stratosphere is of natural origin, but human activity has dramatically increased the levels of chlorine and bromine. These elements are found in certain stable organic compounds, especially chlorofluorocarbons (CFCs), which may find their way to the stratosphere without being destroyed in the troposphere due to their low reactivity. Once in the stratosphere, the Cl and Br atoms are liberated from the parent compounds by the action of ultraviolet light, e.g.

$CFCl_3$ + electromagnetic radiation $\rightarrow CFCl_2 + Cl$

$Cl + O_3 \rightarrow ClO + O_2$ – The chlorine atom changes an ozone molecule to ordinary oxygen

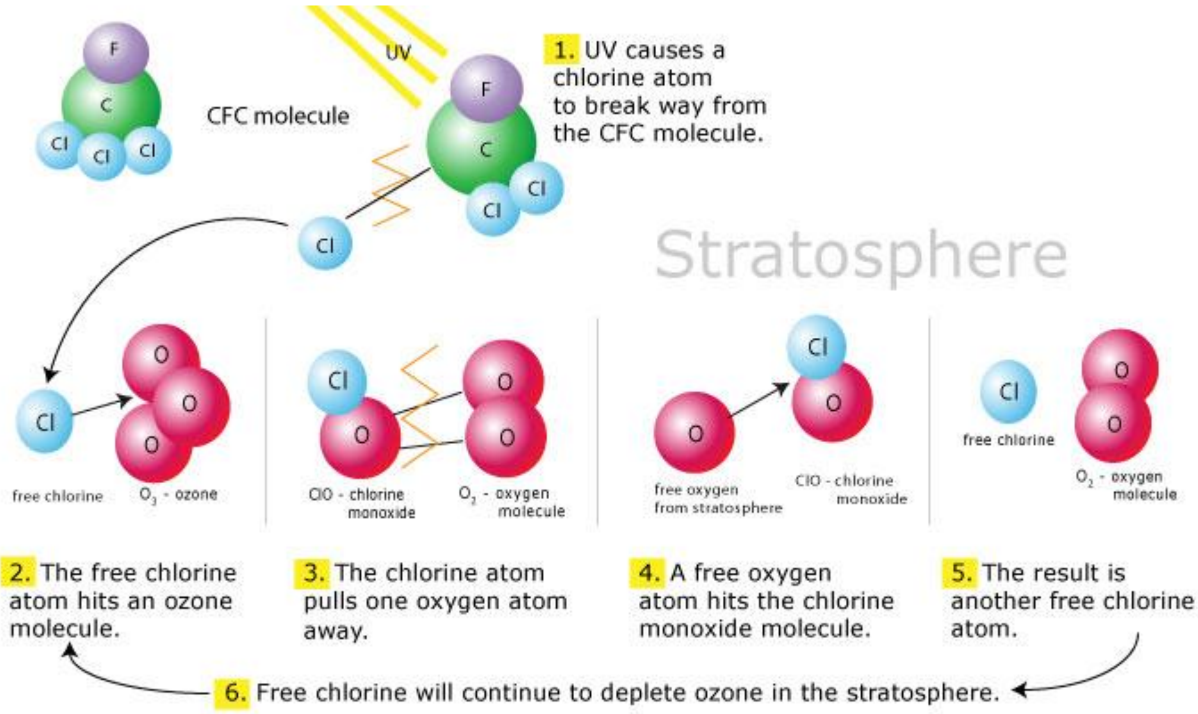
$ClO + O_3 \rightarrow Cl + 2\ O_2$

The ClO from the previous reaction destroys a second ozone molecule and recreates the original chlorine atom, which can repeat the first reaction and continue to destroy ozone

The overall effect is a decrease in the amount of ozone.

Furthermore, a single chlorine atom is able to react with 100,000 ozone molecules. This fact plus the amount of chlorine released into the atmosphere by chlorofluorocarbons

(CFCs) yearly demonstrates how dangerous CFCs are to the environment.



8-5 Classification of climates

The most classification of climate is the Koppen climate classification, that shown in figure 5 which classifies the world into six major groups of world climate, these groups are: 1- Cold and cool climates, 2- Warm temperature rain climate, 3- Hot dry climates (hot desert and steppe climates), 4- Composite climates, 5- Warm humid climates, and 6- Tropical upland climate.

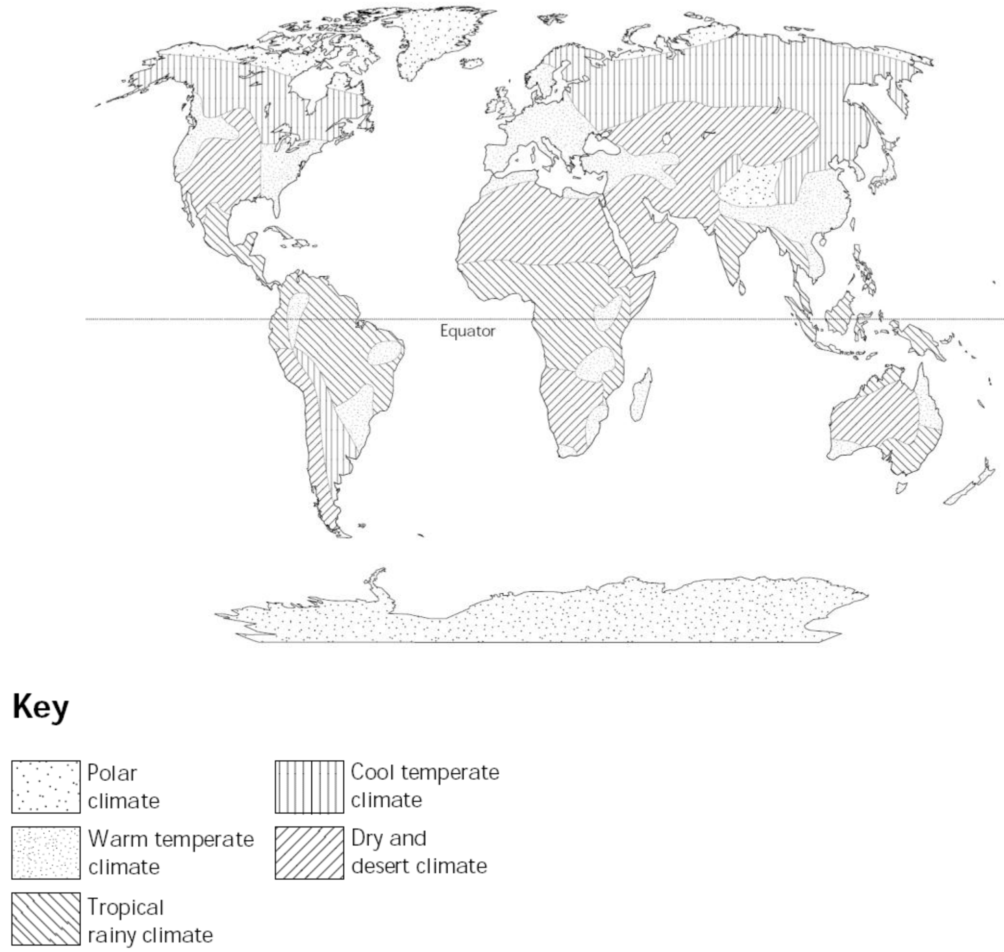


Figure 5 Koppen climate classification

9-Climate of Iraq:

9-1 Temperature;

The annual temperature rang in the arid desert is larger than in any other type of climate within tropic.

- The mean annual temperature of Baghdad is 23^{o}C,
- the mean temperature in August is 34 ^{o}C, and January is 10 ^{o}C.
- The mean temperature for summer season is 33 ^{o}C,
- while the maximum in July is 42 ^{o}C.
- The mean monthly number of day with maximum temperature equal or exceeding to 40^{o}C for Baghdad in August is equal to 30 day, and they are equal to

8 days when the maximums temperature equal to or exceeding 45^{o}C. Figure 6 and 7 show the temperature distribution for different locations in Iraq.

9-2 Sky Conditions and sun shine

Iraqi climate is characterized by right sunshine, high solar radiation with very little could cover and radiation from the ground. Figures 8 and 9 show the mean monthly number of clear day in July, and the sun shine duration.

9-3 Relative humidity

It can be seen from figure 10 and 11 that Baghdad has mean relative humidity of 30% in summer, and 65% in winter.

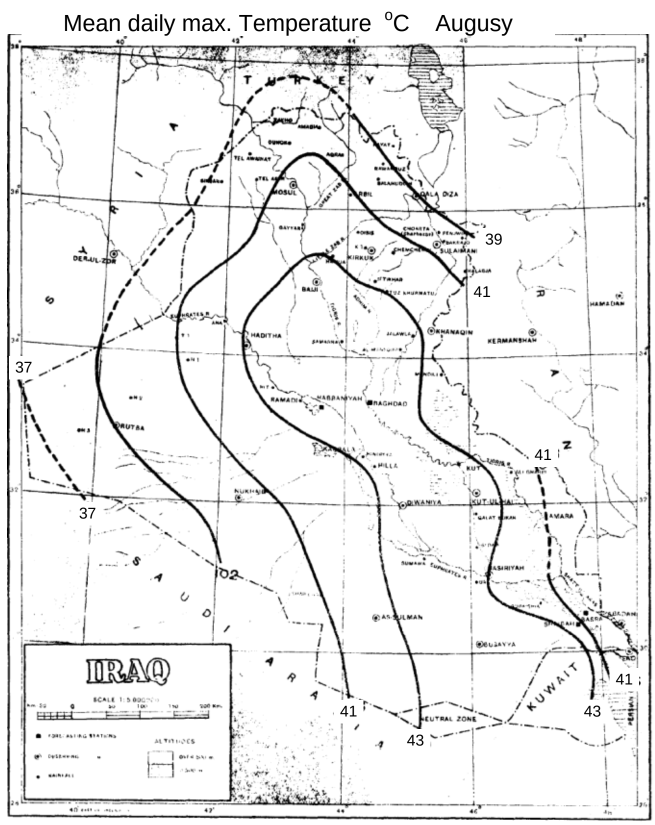


Figure 6 Mean daily maximum temperature ^{o}C August

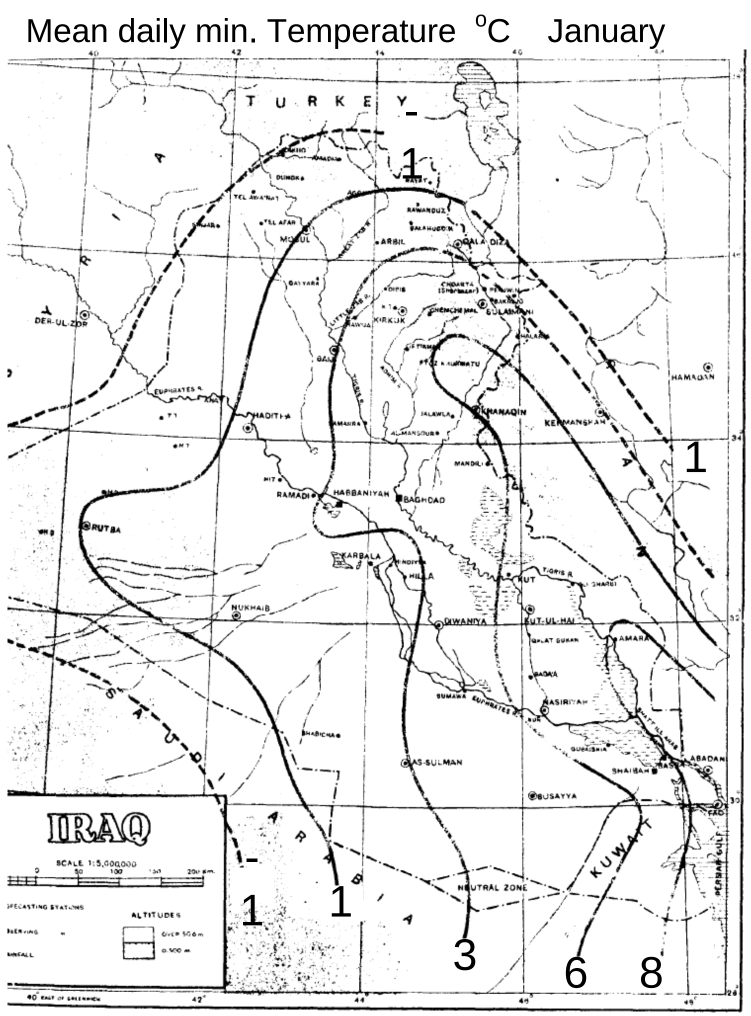


Figure 7 Mean daily maximum temperature °C January

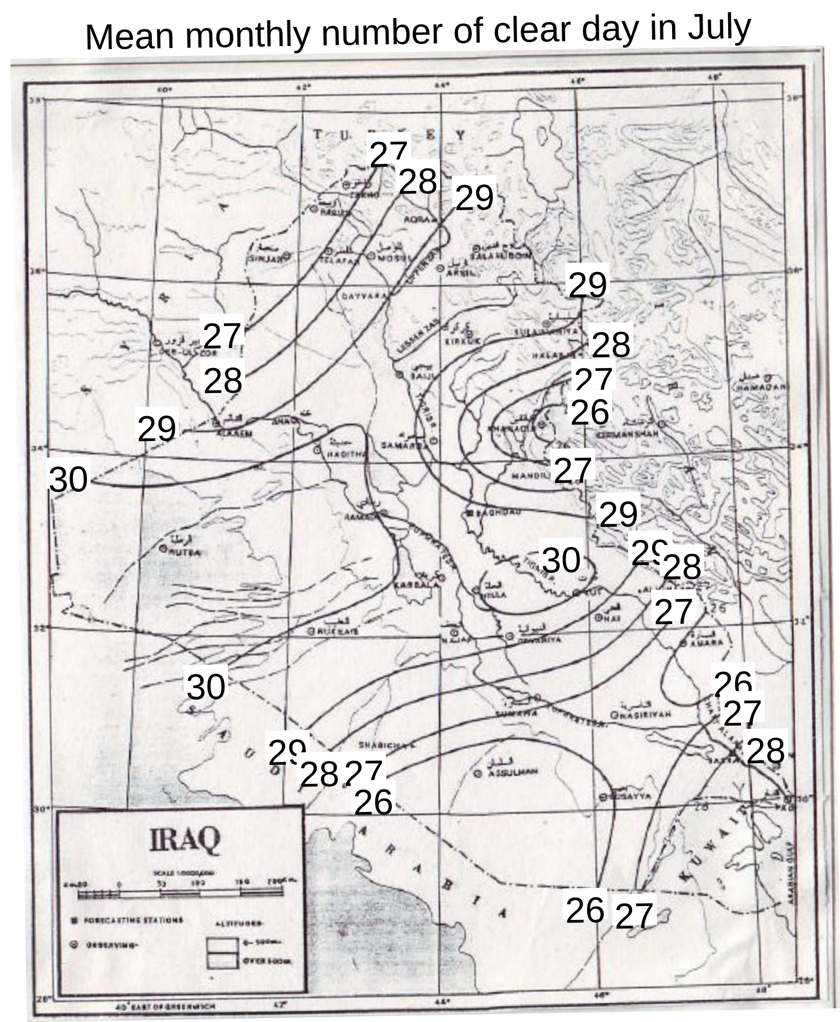


Figure 8 Mean monthly number of clear day in July

Sun shine Duration Scale 1mm= 1 hr

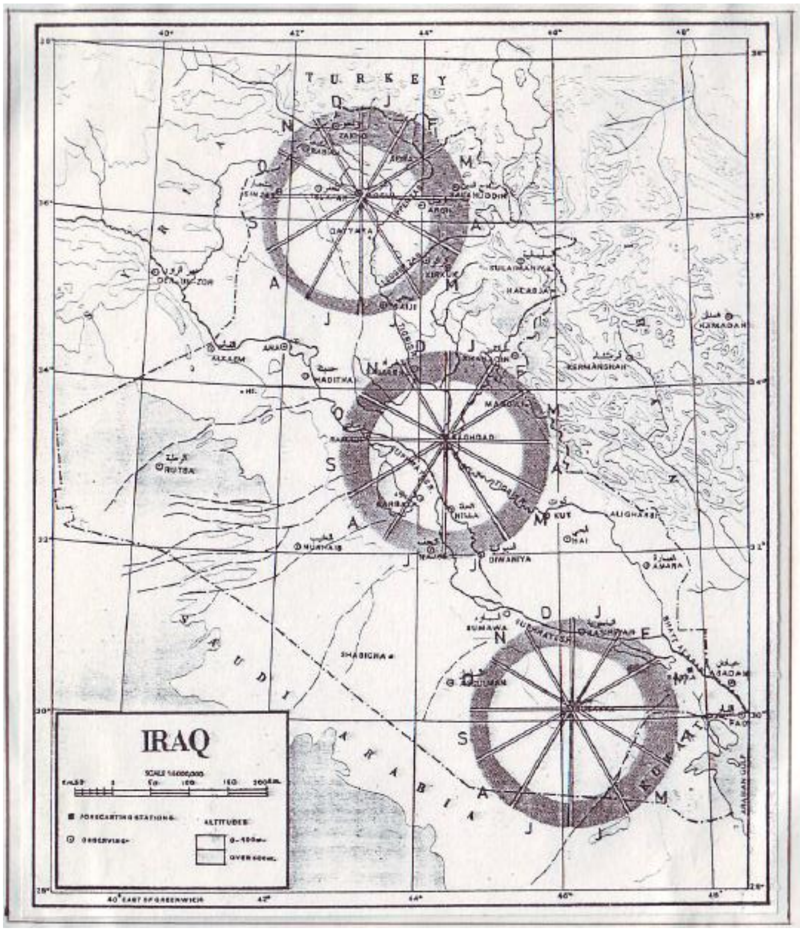


Figure 9 Sun shine duration not to scale

Mean relative humidity in Winter

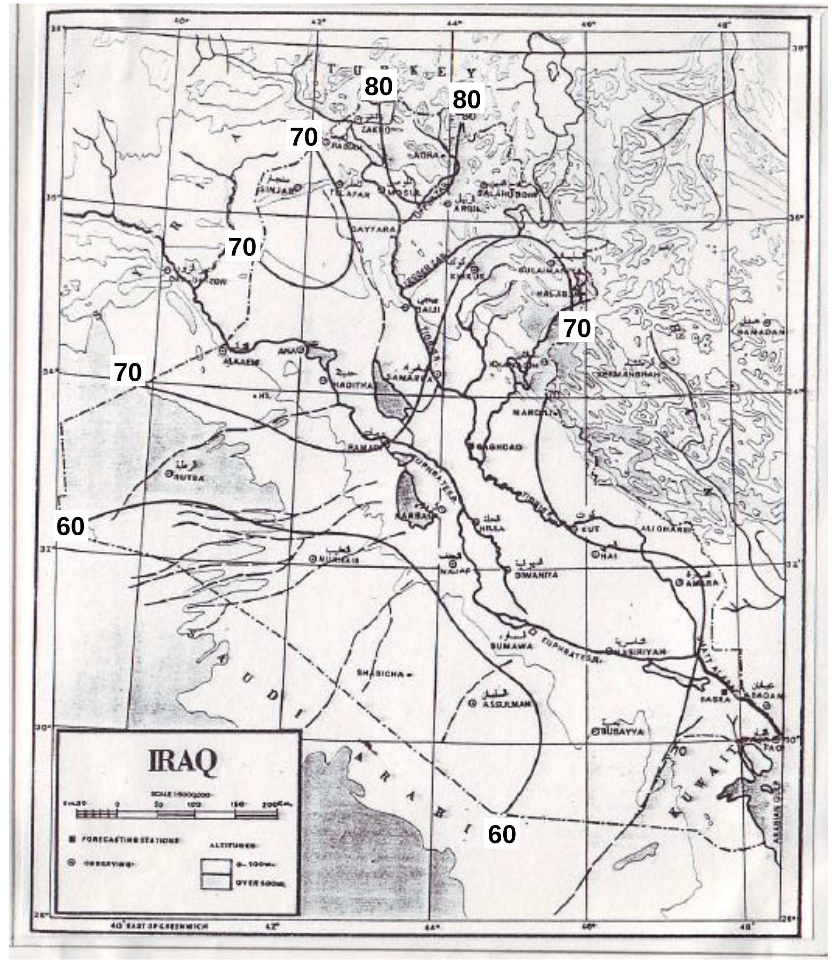


Figure 10 mean relative humidity in winter

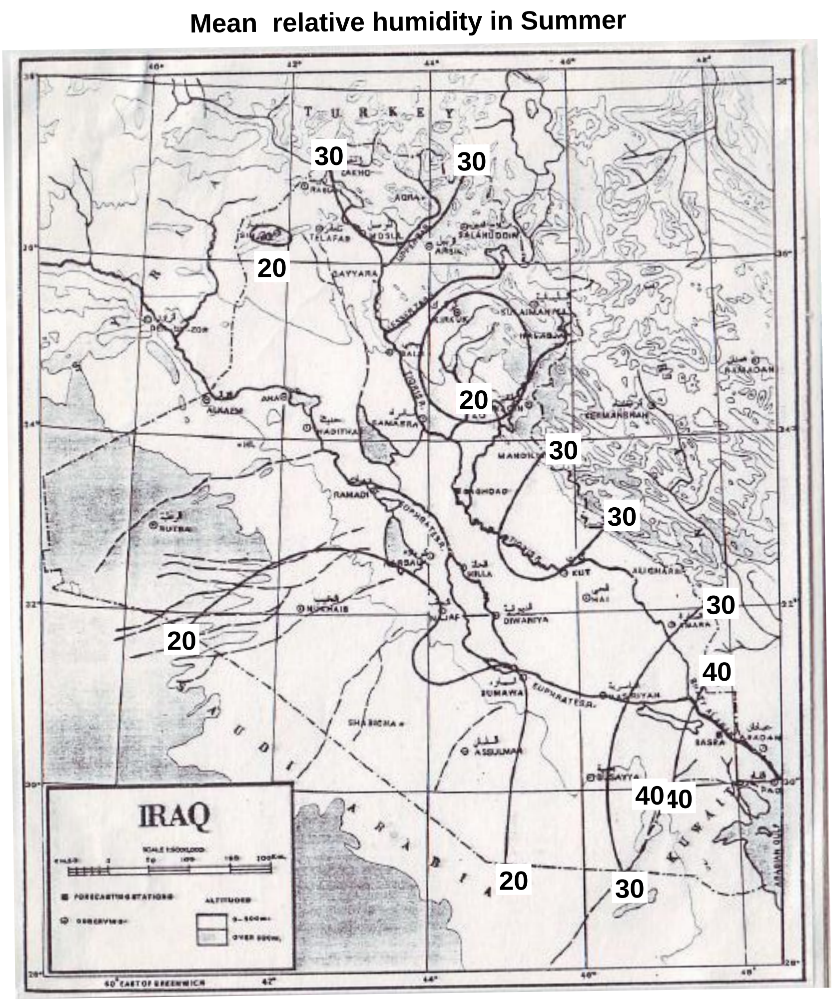


Figure 11 mean relative humidity in Summer

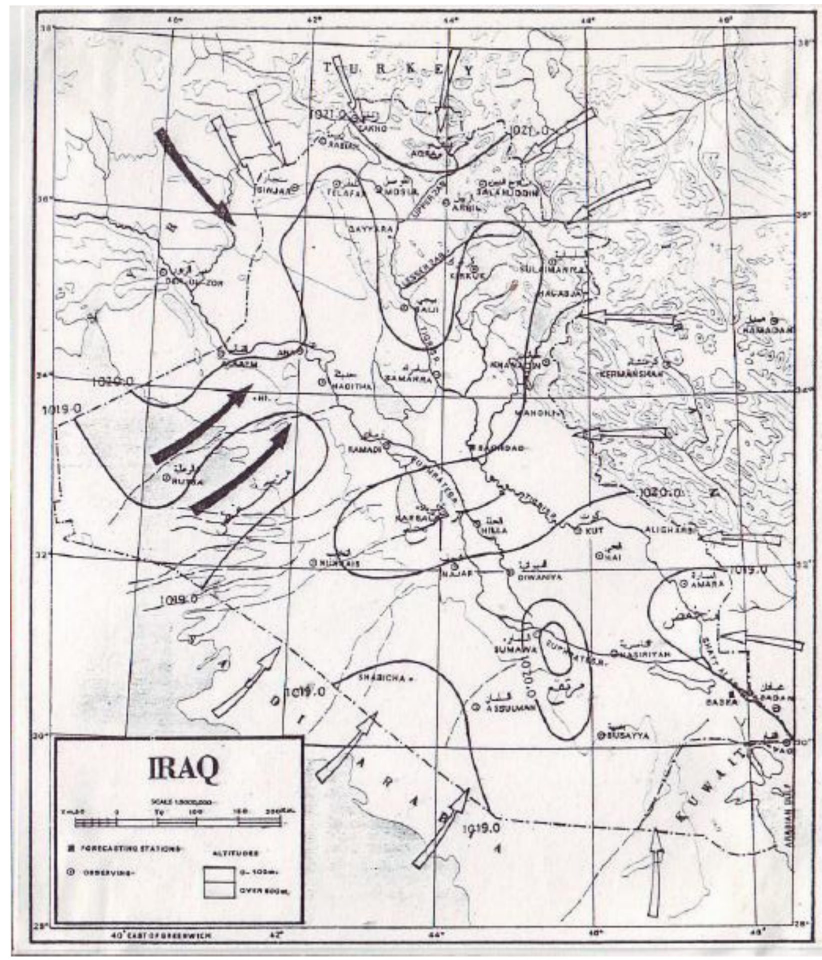


Figure 12 the general distribution of atmospheric pressure and (mbar) and wind flow (January)

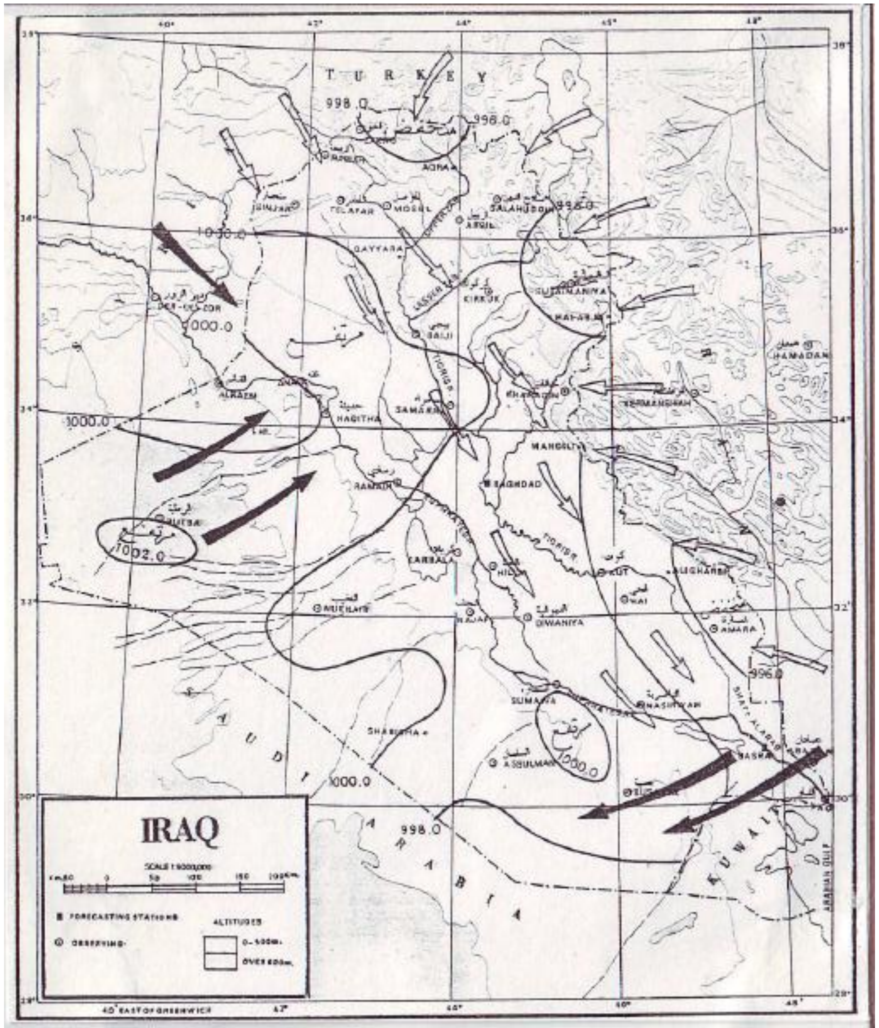


Figure 13 the general distribution of atmospheric pressure and (mbar) and wind flow (July)

Sheet No. Three

1. **Which of the following statements are TRUE?**

a) The metabolic rate depends mainly on age of the human being

b) The metabolic rate depends mainly on the activity level of the human being

c) The metabolic rate depends mainly on the sex of the human being

d) All of the above

Ans.: b)

2. **Which of the following statements are TRUE?**

a) To maintain thermal comfort, the DBT of air should be increased as its moisture content increases

b) To maintain thermal comfort, the DBT of air should be decreased as air velocity increases

c) To maintain thermal comfort, the DBT of air should be increased as the temperature of the surrounding surfaces decrease

d) All of the above

Ans.: c)

3. **Which of the following statements are TRUE?**

a) Surrounding air velocity affects convective heat transfer from the body only

b) Surrounding air velocity affects evaporative heat transfer from the body only

c) Surrounding air velocity affects both convective and evaporative heat transfers from the body

d) Moisture content of the air affects both convective and evaporative heat transfers from the body

Ans.: c)

4. **Which of the following statements are TRUE?**

a) As the amount of clothing increases, the surrounding DBT should be increased to maintain thermal comfort

b) As the amount of clothing increases, the surrounding DBT should be decreased to maintain thermal comfort

c) As the activity level increases, DBT of air should be increased to maintain thermal comfort

d) As the activity level increases, DBT of air should be decreased to maintain thermal comfort

Ans.: b) and d)

5. **Which of the following statements are TRUE?**

a) Effective temperature combines the effects of dry bulb temperature and air velocity into a single index

b) Effective temperature combines the effects of dry bulb temperature and wet bulb temperature into a single index

c) Mean radiant temperature combines the effects of dry bulb temperature and surrounding surface temperature into a single index

d) Operative temperature combines the effects of dry bulb temperature and mean radiant temperature into a single index

Ans.: b) and d)

6. **From ASHRAE comfort chart it is observed that:**

a) Lower dry bulb temperatures and higher moisture content are recommended for winter

b) Lower dry bulb temperatures and lower moisture content are recommended for winter

c) Lower dry bulb temperatures and higher moisture content are recommended for summer

d) Higher dry bulb temperatures and higher moisture content are recommended for summer

Ans.: b) and d)

7. **Which of the following statements are TRUE?**

a) For the same metabolic rate, as the thermal load on human body increases, the PMV value increases

b) For the same metabolic rate, as the thermal load on human body increases, the PMV value decreases

c) As the absolute value of PMV increases, the percent of people dissatisfied (PPD) increases

d) As the absolute value of PMV increases, the percent of people dissatisfied (PPD) decreases

Ans.: a) and c)

8. **Which of the following statements are TRUE?**

a) When a human body is at neutral equilibrium, the PMV value is 1.0

b) When a human body is at neutral equilibrium, the PMV value is 0.0

c) When a human body is at neutral equilibrium, the PPD value is 0.0

d) When a human body is at neutral equilibrium, the PPD value is 5.0

Ans.: b) and d)

9. **Which of the following statements are TRUE?**

a) The air conditioning load on a building increases, if 0.4% design value is used for outside conditions instead of 1.0% value for summer

b) The air conditioning load on a building decreases, if 0.4% design value is used for outside conditions instead of 1.0% value for summer

c) For winter air conditioning, a conservative approach is to select 99.6% value for the outside design conditions instead of 99% value

d) For winter air conditioning, a conservative approach is to select 99% value for the outside design conditions instead of 99.6% value

Ans.: a) and c)

10. A 1.8 meter tall human being with a body mass of 60 kg performs light work (activity = 1.2 met) in an indoor environment. The indoor conditions are: DBT of 30^oC, mean radiant temperature of 32^oC, air velocity of 0.2 m/s. Assuming an average surface temperature of 34^oC for the surface of the human being and light clothing, find the amount of evaporative heat transfer required so that the human being is at neutral equilibrium.

Combined psychrometric Process

The specific objectives of this lecture are to:

1. Purpose of psychrometric calculations.
2. Analysis of a simple, summer air conditioning system with 100% re-circulated air
3. Analysis of a summer air conditioning system with outdoor air for ventilation and with zero by-pass factor
4. Analysis of a simple, summer air conditioning system with outdoor air for ventilation and with non-zero by-pass factor
5. Analysis of a summer air conditioning system with re-heat for high latent cooling load applications
6. Selection guidelines for supply air conditions

At the end of the lesson, the student should be able to:

1. Estimate the load on the cooling coil and fix the supply conditions for various summer conditioning systems, namely:

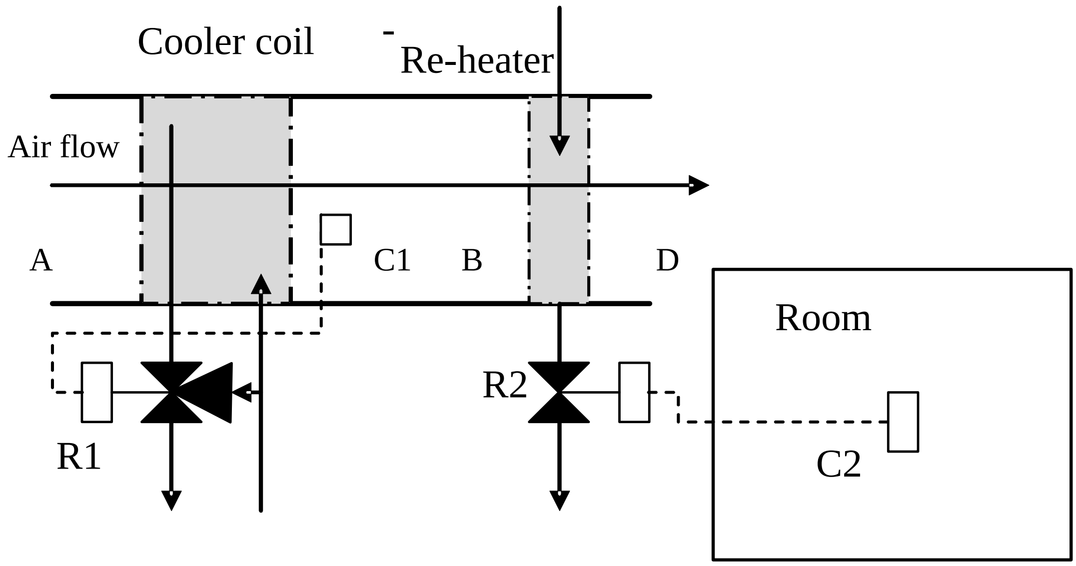
a) Systems with 100% re-circulation

b) Systems with outdoor air for ventilation with zero by-pass factor

c) Systems with outdoor air for ventilation with non-zero by-pass factor

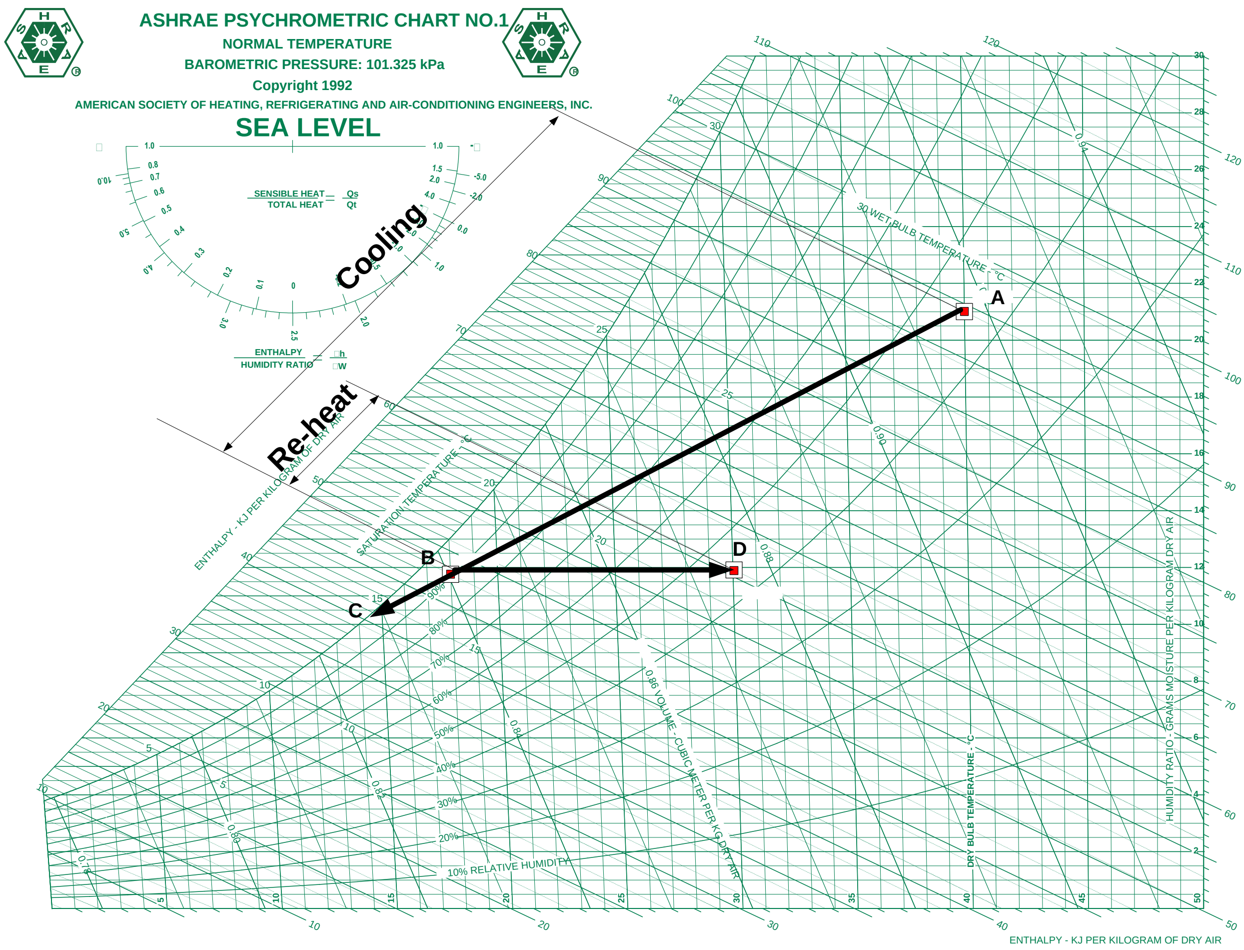
d) Systems with reheat for high latent cooling load applications

Cooling and Dehumidification with re-heat



Moist air at state **A** passes over finned tube of a cooler coil through which chilled water is flowing. The amount of dehumidification carried out is controlled by a dew point thermostat, **C1**, positioned after the coil. This thermostat regulate regulates the amount of chilled water flowing through the coil by means of three way mixing valve **R1**. Air leaves the coil at state **B**, with moisture content suitable for the proper removal of latent heat gains occurring in the room being conditioned. The moisture content has been reduced from $\mathbf{g_a}$ to $\mathbf{g_b}$ and cooler coil has a mean surface temperature of $\mathbf{t_c}$.

If the sensible heat gains then required a temperature of **td**, greater than $\mathbf{t_b}$, the air is passed over the tubes of a heater, through which a low pressure hot water may be flowing. The flow rate of water is regulate by means of two port modeling valve **R2** controlled from thermostat **C2** positioned to this room at state **D**, with the correct temperature and moisture content. Re-heat is usually only permitted to waste cooling capacity under partial load condition, that is, the design should be such that the state B can adequately deal both maximum sensible and maximum latent loads.



Example 9

Moist air at 28°C DBT and 21 °C WBT and 101.325 kPa pressure flows over a cooler coil and leaves it at a state 10°C DBT and 7.046 g/kga moisture content.

a- if the air required to offset a sensible heat gain of 2.35 kW and latent heat gain of 0.31 kW in a space being air-conditioned, calculate the mass of dry air which must be supplied to the room in order to maintain a dry bulb temperature of 21°C therein.

b- What will be the relative humidity.

c- If the sensible heat gain diminishes by 1.75 kW but the latent heat remains unchanged, at what temperature and moisture content must be supplied to the room.

$$Qs = ma.cp.(Tr - Tc)$$
$$2.35 = ma.1.005(21 - 10)$$
$$ma = 0.211 kg/s$$

$$Ql = mw.hfg$$
$$0.31 = mw(2454)$$
$$mw = 0.0001262 kgw/s$$

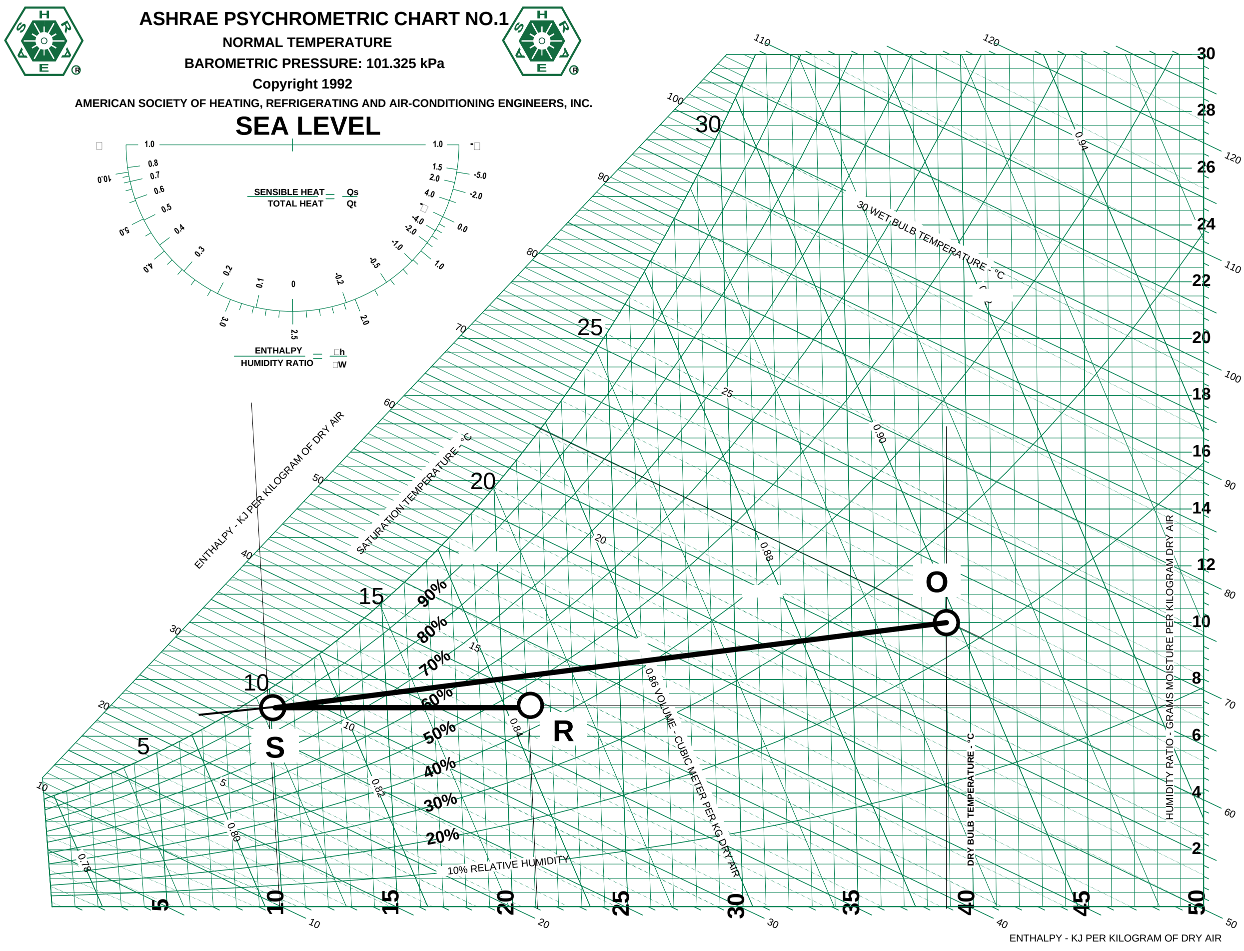
the moisture content with 0.211kga/s is

$$g = \frac{0.1262}{0.211} = 0.599 gw/kga$$

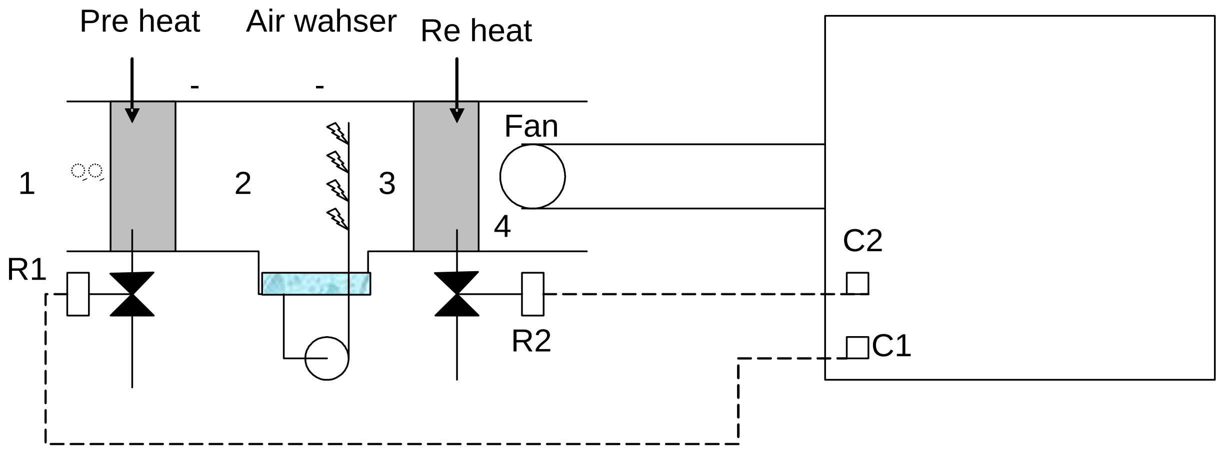
then the moisture content in the room is

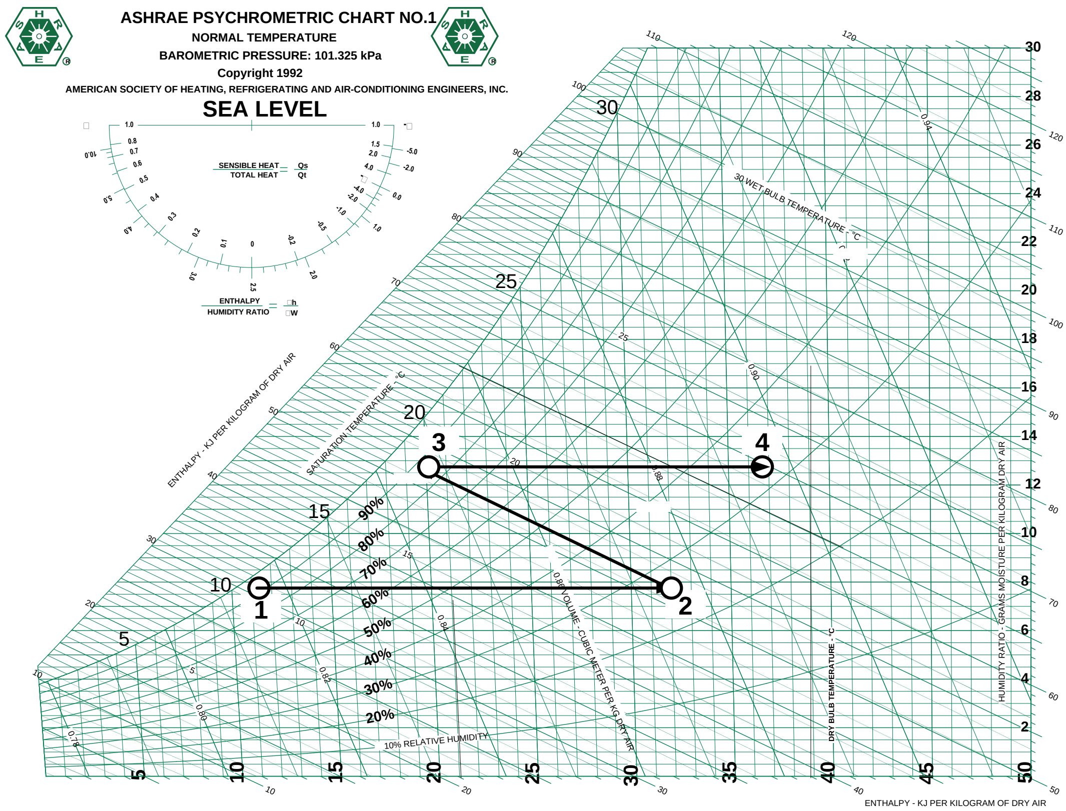
$$g_{room} = 7.046 + 0.599 = 7.645 gw/kga$$

from the psychrometric the relative humidity is 51%



Pre-heating and humidification with re-heat



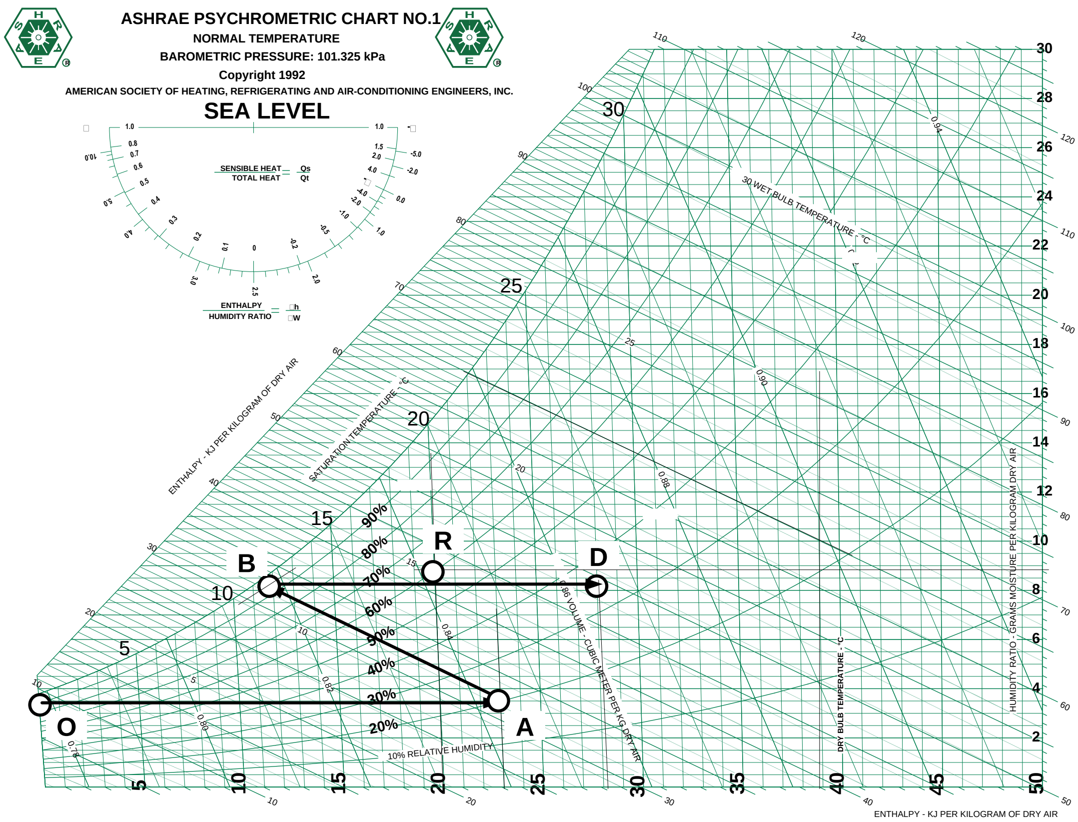


Air-conditioning plants which handle fresh air only are faced in winter with the task of increasing both the moisture content and the temperature of they supply to the conditioned space. Humidification is needed because the outside air in winter may have a very low moisture content, and if this air were to be introduced directly to the room there would be a low moisture content there as well, and when the air is heated to higher temperature its relative humidity may become very low. Therefore the room condition will be far from human comfort. A popular and effective approach is to pre-heat the air(1-2), pass it through air washer(2-3), where it undergoes adiabatic saturation, and then to reheat it to the temperature at which it must be supplied to the room. The figure above shows a typical plant. Opening the valve R1 in the return pipeline from the pre-heater increases the heating output of the battery. Similarly, opining the control valve R2, associated with the re-heater, allows air at a higher temperature to be delivered to the room being conditioned. C1 and C2 are room humidistat and room thermostat, respectively.

Exapmle10:

Air is pre-heat from (0^oC) DBT and 86% RH to 23^oC DBT. It is then passed through air washer having humidifying efficiency of 85% and using re-circulated spray water. Calculate the following:

1. the relative humidity of the air leaving the washer.
2. the cold water makeup to the washer in lit/s given that $2.5m^3/s$ leaves the air washer.
3. the duty of the pre-heater in kW
4. the temperature of the air supplied to the conditioned space if the sensible heat loss from it are 24kW and 20^oC DBT is maintained there
5. the duty of the re-heater in kW
6. the relative humidity maintained in the room if the latent heat gain are 5 kW



the moisture content of point O is 0.0038 kgw/kga, and the WBT of point A is 11.5^{o}C

$$E=\frac{TA-TB}{TA-TC}$$

$$TB=23-0.85(23-10)=12^{o}C$$

1. g_B=0.0082 from psychrometric chart, $\phi=94\%$

cold water makeup= $\rho_a.V_a(g_B-g_A)$

2. $mw=1.284\times2.5\times1.005(0.0082-0.0038)=0.0141kgw/s$

load of pre-heater= $\rho_a.V_a(h_A-h_O)$

3. $Q=1.284\times2.5\times1.005(23.18-8.11)=48.37kW$

4. $$Qs=ma.cp(T_D-T_R)$$
$$T_D=\frac{24}{1.22\times2.5\times1.005}+20=27.8\approx28^{o}C$$

5. $Q_{re-heater}=ma.cp.(T_D-T_B)=1.22\times2.5\times1.005(28-12)=49.04kW$

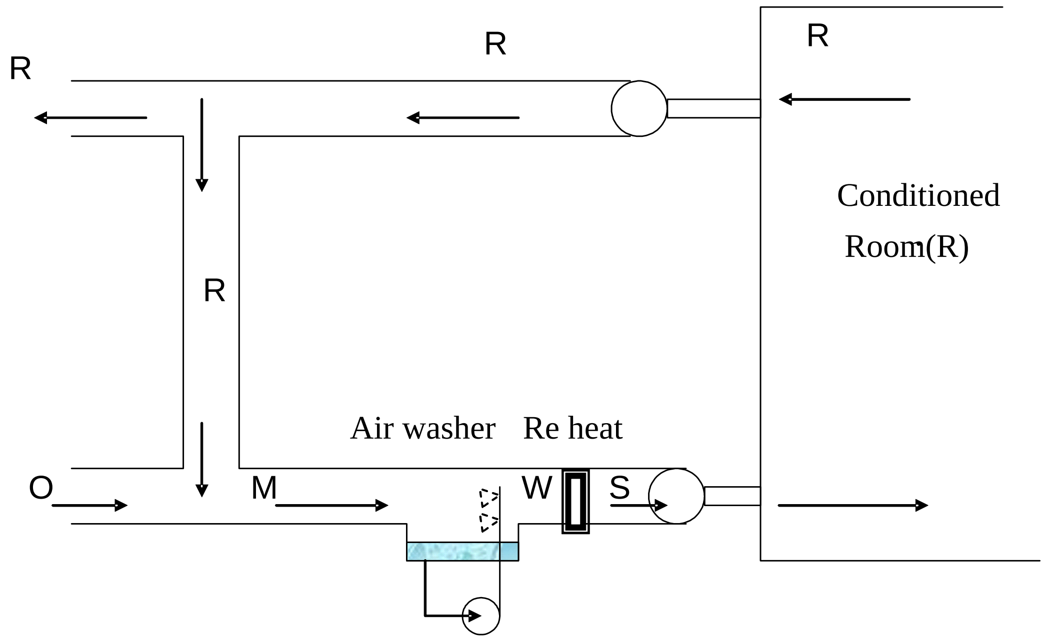
moisture liberated in the room can be found as follow

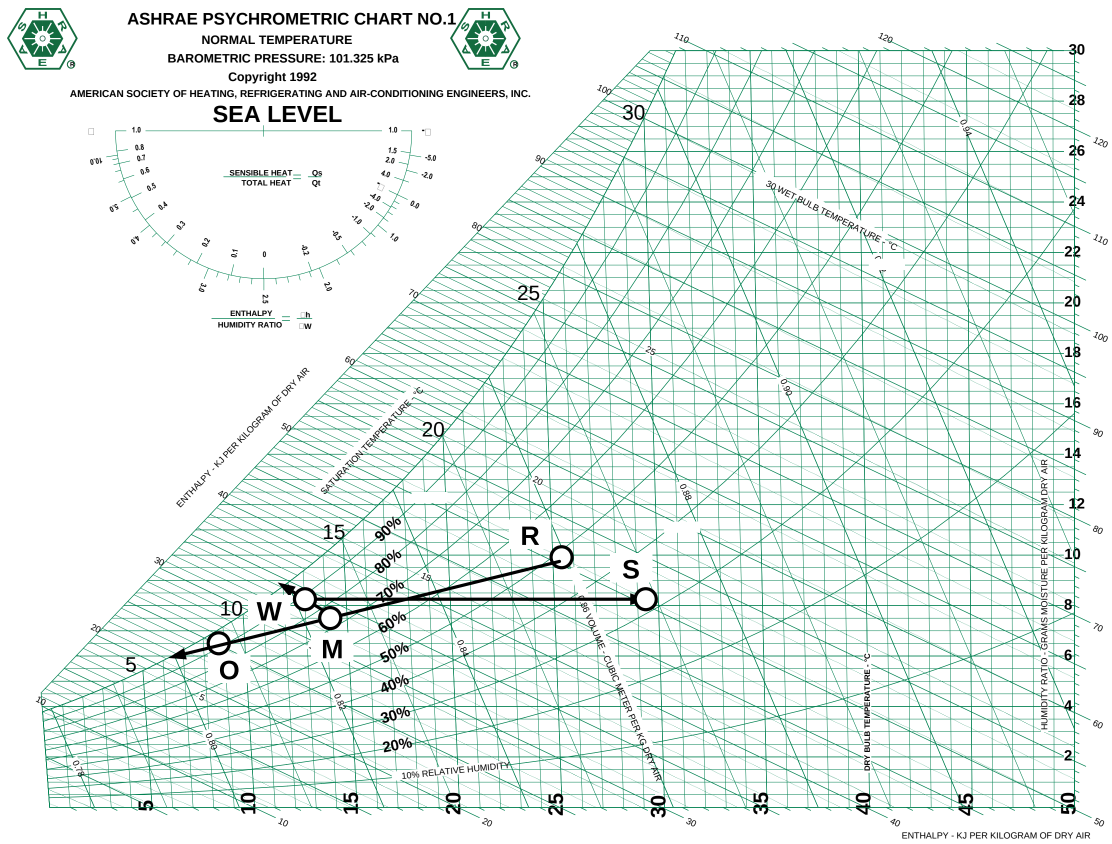
$$Ql = ma.(g_D - g_R)h_{fg}$$

$$g_R = \frac{5}{(1.22 \times 2.5 \times 1.005)2454} + 0.0082 = 0.008864 kgw/kga$$

6. Therefore room condition can be found from psychrometric chart using 20°C DBT & 0.008864 moisture content: then the relative humidity is about 61%

Mixing and Adiabatic Saturation with Re-heat

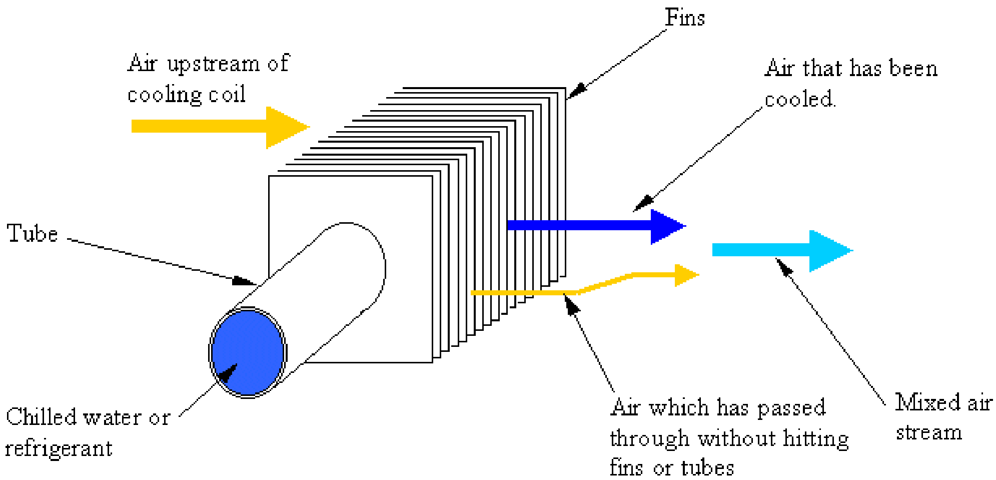




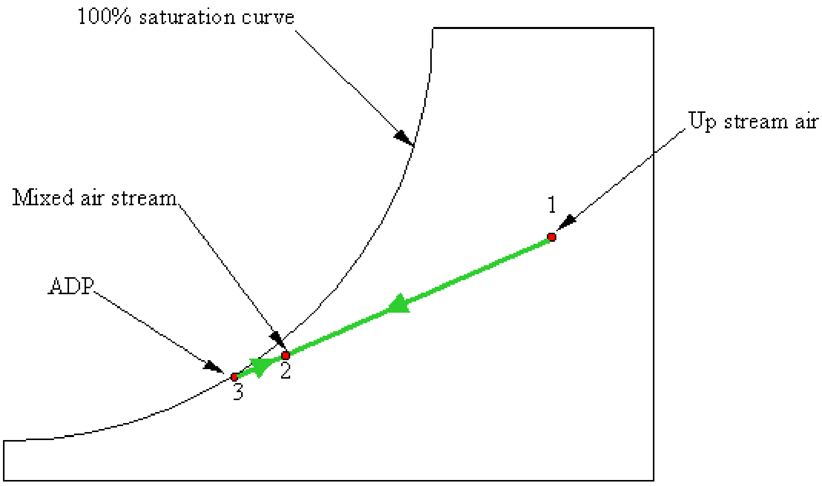
The figure above shows a very common type of plant, air at state ***R*** is extracted from the conditioned room and partly re-circulated, the reminder being discharge to the atmosphere. The portion of the extracted air returned to the air conditioned plant mixes with out side air at state ***O***, and forms a mixture at state ***M***. The mixed air is the passed through an air-washer and adiabatic saturation occur, the state of air change from ***M*** to ***W*** at constant WBT, the extension of line ***MW*** cut the saturation curve at ***C*** (the apparatus dew point), the humidified air is then re-heated by a re-heater battery and leave at supply condition ***S***.

Cooling Coil Contact Factor

Some of the air going through a cooling coil does not come into contact with the tubes or fins of the cooling coil and is therefore not cooled to the ADP temperature. A mixing process therefore takes place as two air streams mix downstream of the cooling coil as shown below. One air stream is cooled down to the ADP and the other air stream by-passes the coil surfaces to give an off-coil air temperature (mixed air stream) a little higher than the ADP. This may be looked upon as an inefficiency of the coil and is usually given as the cooling coil contact factor. The process is shown on the psychrometric chart below



A SECTION OF COOLING COIL SHOWING AIR STREAMS



PSYCHROMETRIC CHART SHOWING COOLING COIL CONTACT FACTOR

$$\beta = \frac{Line\ \ 1-2}{Line\ 1-3}$$

While the by-pass factor is:

$$B.F. = \frac{line\ 2-3}{line\ 1-3}$$

The contact factor can be calculated as:

$\beta = 1 - BF$

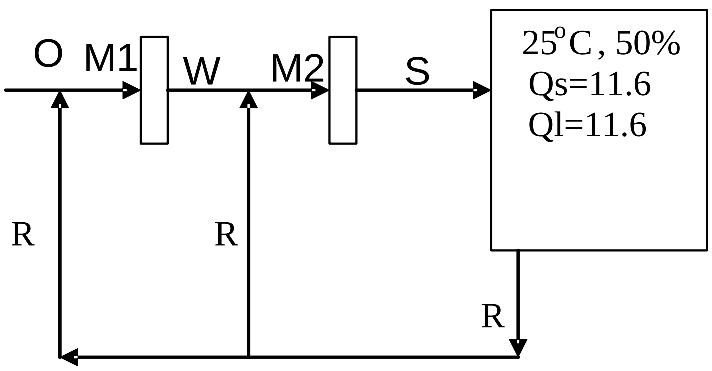
$$\beta = \frac{DBT_m - DBT_s}{DBT_m - ADP} = \frac{g_m - g_s}{g_m - g_{ADP}} = \frac{h_m - h_s}{h_m - hg_{ADP}}$$

While the bypass factor can be calculated as:

$$BF = \frac{DBT_s - ADP}{DBT_m - ADP} = \frac{g_s - g_{ADP}}{g_m - g_{ADP}} = \frac{h_s - h_{ADP}}{h_m - hg_{ADP}}$$

Example 11

The following data apply to an A/C unit. Room sensible heat and latent heat are 11.6kW and 11.6 kW respectively, inside condition is 25°C DBT, 50% RH. Outside design condition is 35 °C DBT and 27.8 °C, return air from the room is mixed with outside air in the ratio of 4:1 before entering the cooler coil. Return air from the room is mixed with the air leaving the coil in the ratio of 1:4. cooling coil bypass factor 0.1, the air may reheated if necessary before supply to the room. Coil temperature is 10 °C. find a-supply air condition b- refrigeration load due to reheat c- total refrigeration load capacity d- the quantity of fresh air supplied.



$$T_{M1} = \frac{4\times T_R + 1\times T_O}{5} = \frac{4\times 25 + 1\times 35}{4} = 27^o C$$

$$\beta = \frac{T_C - T_W}{T_C - T_{M1}}$$

$$0.1 = \frac{10 - T_W}{10 - 27} \Rightarrow\Rightarrow T_W = 11.7^o C$$

$$T_{M2} = \frac{4\times T_W + 1\times T_R}{5} = \frac{4\times 11.7 + 1\times 25}{5} = 14.36^o C$$

$$RSH = \frac{Qs}{QS + Ql} = \frac{Qs}{Qt} = \frac{11.6}{11.6 + 11.6} = 0.5$$

Draw the sensible heat line with 0.5slope, and translate the line to the psychrometric chart beginning from room condition R and. The room condition should on the line of RSH. There for we should reheated the air at M2. since the condition is heating, there for we should draw a horizontal line from M2 till this line is cut the line of RSH, the intersection is at S which is represent the supply condition to the room.

Then the supply condition S is

T_s=21^{o}C and 54% RH.

$$Qs = m.c_p.(T_R - T_S)$$

$$11.6 = \dot{m}.1.005(25-21)$$

$$\dot{m} = 2.885 kg/s$$

$$Q_{reheater} = \dot{m}.cp(T_S - T_{M2})$$

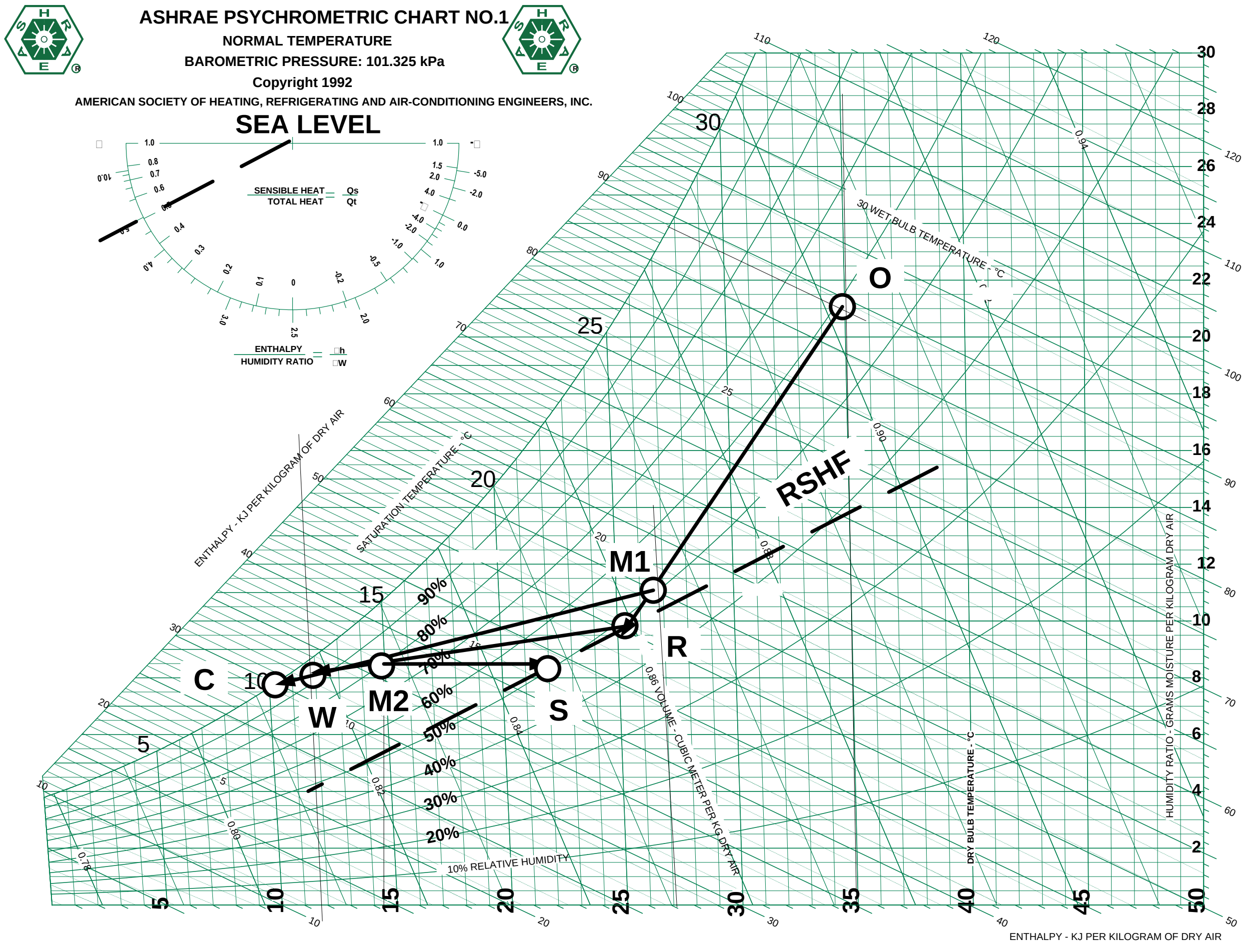
$$Q_{reheater} = 2.885\times 1.005\times (21.14.36) = 19.2 kW$$

$$\dot{m}_W = \frac{4}{5}\dot{m}_S = 0.8\times 2.885 = 2.304 kg/s$$

$$Q_{cooler} = \dot{m}_W.(h_{M1} - h_{W1}) = 2.304(58.2-32)$$

$$Q_{cooler} = 60.5 kW$$

$$\dot{m}_O = \frac{1}{5}\dot{m} = 0.2\times 2.304 = 0.4615 kg/s$$



Air Conditioning Cycles on Psychrometric Chart

Air Washers:

An air washer is a device for conditioning air. As shown in Fig.1, in an air washer air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapour) between air and water. The outlet condition of air depends upon the temperature of water sprayed in the air washer. Hence, by controlling the water temperature externally, it is possible to control the outlet conditions of air, which then can be used for air conditioning purposes

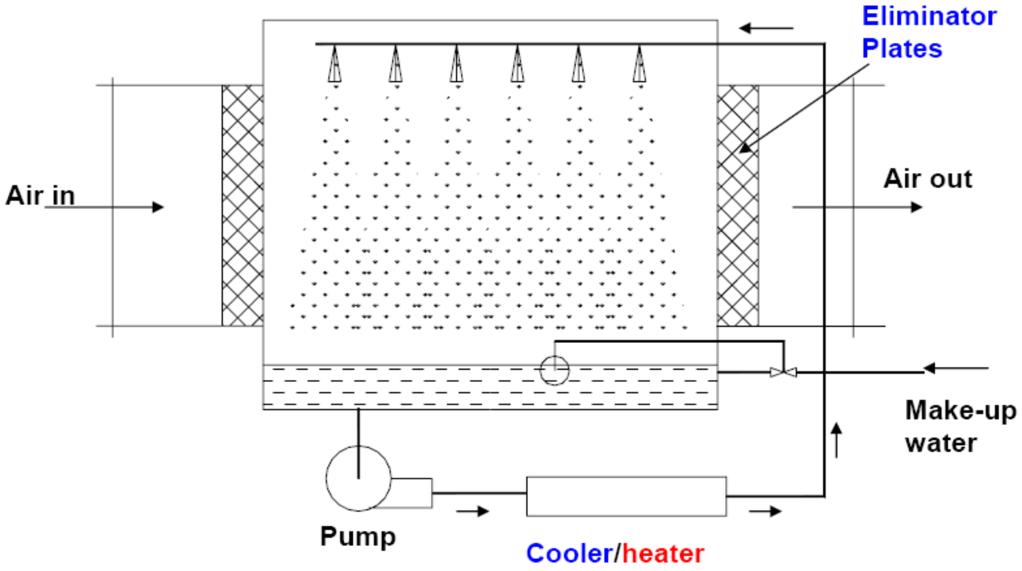


Figure 1 Air Washer

In the air washer, the mean temperature of water droplets in contact with air decides the direction of heat and mass transfer. As a consequence of the 2nd law, the heat transfer between air and water droplets will be in the direction of decreasing temperature gradient. Similarly, the mass transfer will be in the direction of decreasing vapor pressure gradient. For example,

a) Cooling and dehumidification: **tw < DPT**. Since the exit enthalpy of air is less than its inlet value, from energy balance it can be shown that there is a transfer of total energy from air to water. Hence to continue the process, water has to be externally cooled.

Here both latent and sensible heat transfers are from air to water. This is shown by Process O-A in Fig.1.

b) Adiabatic saturation**: tw = WBT**. Here the sensible heat transfer from air to water is exactly equal to latent heat transfer from water to air. Hence, no external cooling or heating of water is required. That is this is a case of pure water recirculation. This is shown by Process O-B in Fig.1. This the process that takes place in a perfectly insulated evaporative cooler.

c) Cooling and humidification: **DPT < tw <tWBT**. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer

is from air to water, hence, water has to be cooled externally. This is shown by Process O-C in Fig.1

d) Cooling and humidification: **<u>WBT < tw < DBT</u>**. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from water to air, hence, water has to be heated externally. This is shown by Process O-D in Fig.1. This is the process that takes place in a cooling tower. The air stream extracts heat from the hot water coming from the condenser, and the cooled water is sent back to the condenser.

e) Heating and humidification: **<u>tw > DBT</u>**. Here both sensible and latent heat transfers are from water to air, hence, water has to be heated externally. This is shown by Process O-E in Fig.1.

Thus, it can be seen that an air washer works as a year-round air conditioning system. Though air washer is a and extremely useful simple device, it is not commonly used for comfort air conditioning applications due to concerns about health resulting from bacterial or fungal growth on the wetted surfaces. However, it can be used in industrial applications.

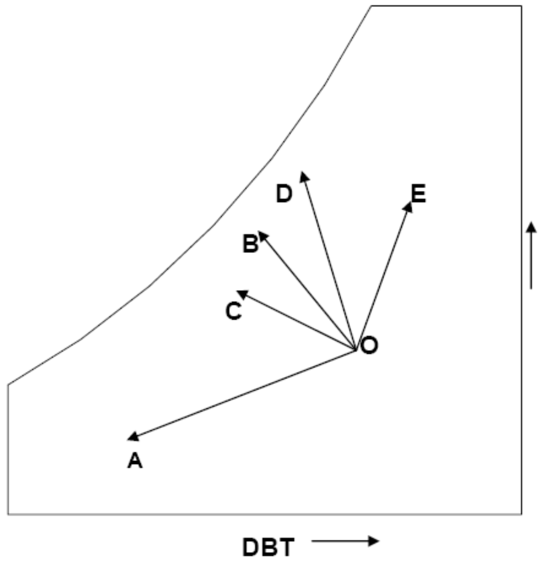


Fig.1: psychrometric processes that can take place in an air washer

Introduction:

Generally from the building specifications, inside and outside design conditions; the latent and sensible cooling or heating loads on a building can be estimated. Normally, depending on the ventilation requirements of the building, the required outdoor air (fresh air) is specified. The topic of load estimation will be discussed in a later chapter. From known loads on the building and design inside and outside conditions, psychrometric calculations are performed to find:

1. Supply air conditions (air flow rate, DBT, humidity ratio & enthalpy)
2. Coil specifications (Latent and sensible loads on coil, coil ADP & BPF)

In this chapter fixing of supply air conditions and coil specifications for summer air conditioning systems are discussed. Since the procedure is similar for winter air conditioning system, the winter air conditioning systems are not discussed here.

Summer air conditioning systems

Simple system with 100 % re-circulated air:

In this simple system, there is no outside air and the same air is recirculated as shown in Figure 2 also shows the process on a psychrometric chart. It can be seen that cold and dry air is supplied to the room and the air that leaves the condition space is assumed to be at the same conditions as that of the conditioned space. The supply air condition should be such that as it flows through the conditioned space it can counteract the sensible and latent heat transfers taking place from the outside to the conditioned space, so that the space can be maintained at required low temperature and humidity. Assuming no heat gains in the supply and

return ducts and no energy addition due to fans, and applying energy balance across the room; the Room Sensible Cooling load (Qs,r), Room Latent Cooling Load (Ql,r) and Room Total Cooling load (Qt,r) are given by:

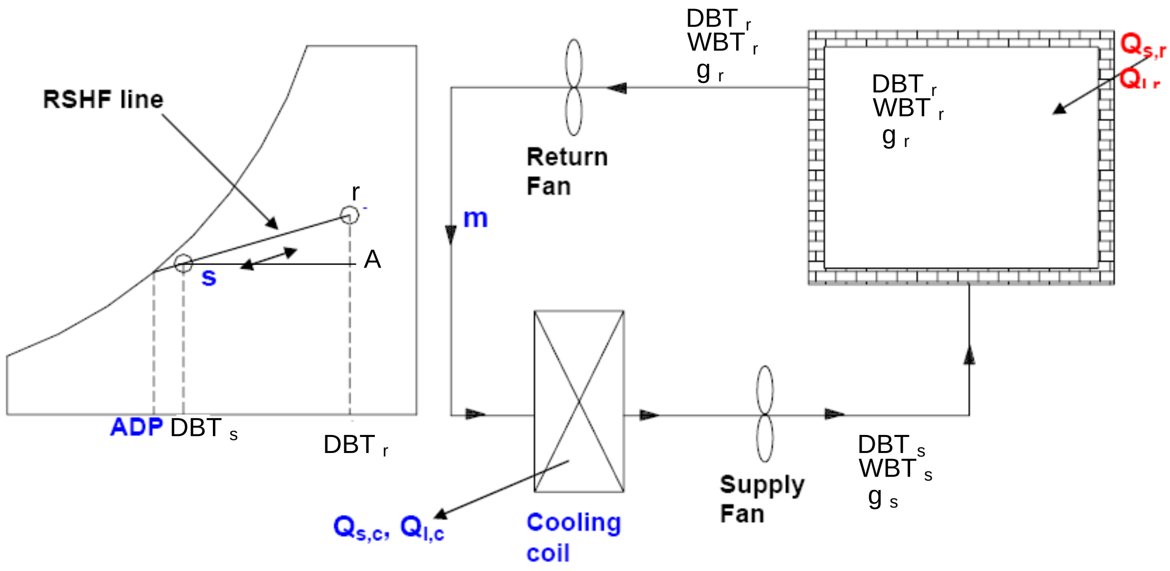


Figure 2 the psychrometery of Simple system with 100 % re-circulated air

$$Q_{sr} = \dot{m}_a . c_{pa} . (DBT_r - DBT_s) = \dot{m}_a (h_A - h_s)$$

$$Q_{lr} = \dot{m}_a . h_{fg} . (g_r - g_s) = \dot{m}_a (h_r - h_A)$$

$$Q_{Tr} = Q_{sr} + Q_{lr} = \dot{m}_a (h_r - h_s)$$

From cooling load calculations, the sensible, latent and total cooling loads on the room are obtained. Hence one can find the Room Sensible Heat Factor (RSHF) from the equation

$$RSHF = \frac{Q_{sr}}{Q_{sr} + Q_{lr}} = \frac{Q_{sr}}{Q_{Tr}}$$

The intersection of this line with the saturation curve gives the ADP of the cooling coil as shown in Fig.2. It should be noted that for the given room sensible and latent cooling loads, **the supply condition must always lie on this line so that the it can extract the sensible and latent loads on the conditioned space in the required proportions.**

System with outdoor air for ventilation:

In actual air conditioning systems, some amount of outdoor (fresh) air is added to take care of the ventilation requirements. Normally, the required outdoor air for ventilation purposes is known from the occupancy data and the type of the building (e.g. operation theatres require 100% outdoor air). Normally either the quantity of outdoor air required is specified in absolute values or it is specified as a fraction of the re-circulated air.

Case i) By-pass factor of the cooling coil is zero:

Figure 3 shows the schematic of the summer air conditioning system with outdoor air and the corresponding process on psychrometric chart, when the by-pass factor B.F. is zero. Since the sensible and latent cooling loads on the conditioned space are assumed to be known from cooling load calculations, similar to the earlier case, one can draw the process line s-R, from the RSHF and state i. The intersection of this line with the saturation curve gives the room ADP. As shown on the psychrometric chart, when the by-pass factor is zero, the room ADP is equal to coil ADP, which in turn is equal to the temperature of the supply air. Hence from the supply temperature one can calculate the required supply air mass flow rate (which is the minimum required as B.F. is zero) using the equation:

$$\dot{m}_a = \frac{Q_{s,r}}{c_{pm}.(DBT_R - DBT_S)} = \frac{Q_{sr}}{(h_A - h_S)}$$

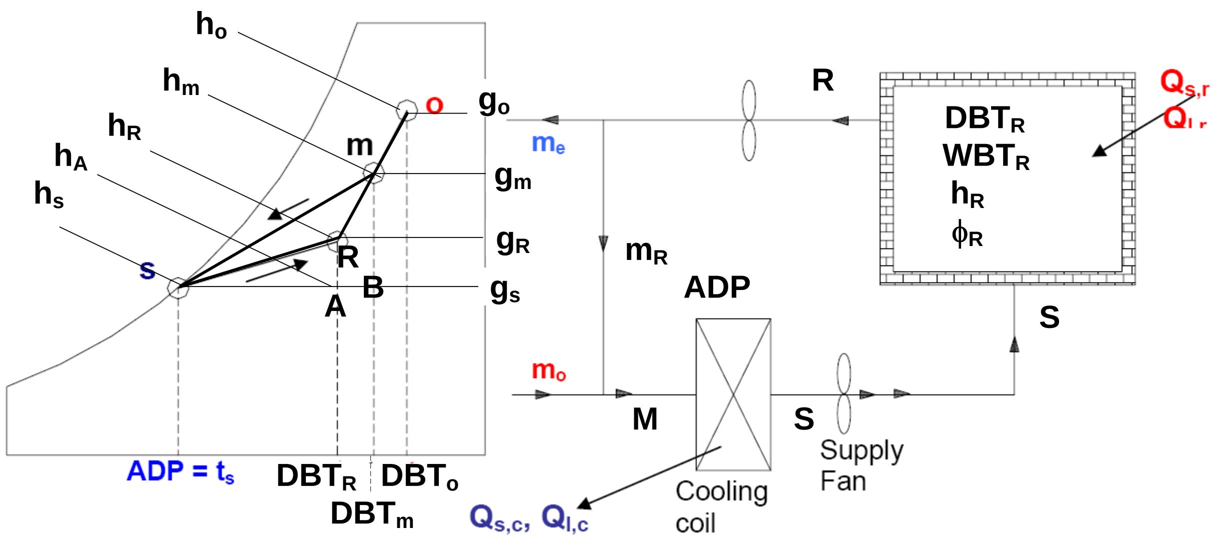


Figure 3 schematic of the summer air conditioning system with outdoor air

Room Load:	**Cooling Coil Load:**
Sensible load $Q_{sr} = \dot{m}_a.c_{pm}.(DBT_R - DBT_S)$ $= \dot{m}_a.(h_A - h_S)$ Room Total load $Q_{Tr} = \dot{m}_a.(h_R - h_S)$ Latent load $Q_{lr} = 2501.\dot{m}_a.(g_R - g_S)$ $= (Q_{Tr} - Q_{Sr})$	Sensible load: $Q_{sc} = \dot{m}_a.c_{pm}.(DBT_M - DBT_S)$ $= \dot{m}_a.(h_B - h_S)$ Total Load: $Q_{Tc} = \dot{m}_a.(h_M - h_S)$ Latent load: $Q_{lc} = 2501.\dot{m}_a.(g_M - g_S)$ $= (Q_{Tc} - Q_{Sc})$

The line joining the mixed condition (M) with the coil (ADP) is the process line undergoes by the air as it flows the cooling coil. The slope of this line depends on the Coil Sensible Heat Factor (CSHF) given by :

$$CSHF = \frac{Q_{sc}}{Q_{sc} + Q_{lC}} = \frac{Q_{sc}}{Q_{Tc}}$$

Mixing conditions

Mass balanc**e**

$$\dot{m}_M = \dot{m}_R + \dot{m}_o$$

$$\dot{m}_M.h_M = \dot{m}_{R.}h_R + \dot{m}_{o.}h_o$$

Or

$$\dot{m}_M.DBT_M = \dot{m}_{R.}DBT_R + D\dot{B}T_{o.}h_o$$

The load on the cooling coil is greater than on the conditioned space. This is due to the fact that during mixing, some amount of hot and humid air is added and the same amount of relative cool and dry air is exhausted.

Case ii: Coil by-pass factor, BF > 0:

For actual cooling coils, the bypass factor will be greater than zero, as a result the air temperature at the exit of the cooling coil will be higher than the coil ADP. This shown in figure 4 along with the process on psychrometric chart, it can be seen from the figure that when BF > 0, the room ADP will be different from the coil ADP. The system shown in Fig.4 is adequate when the RSHF is high (> 0.75).

Calculate the supply conditions using Contact factor:

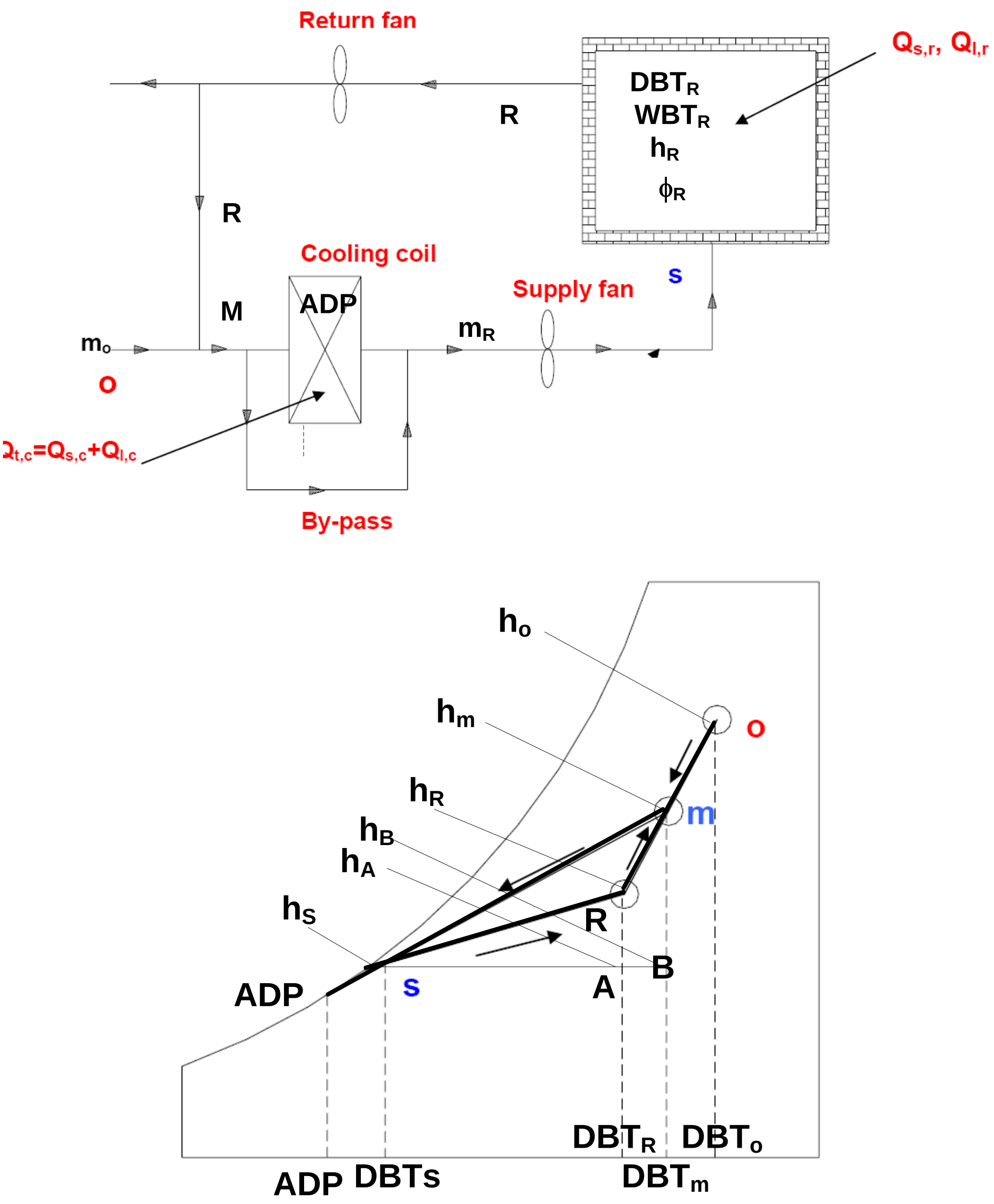
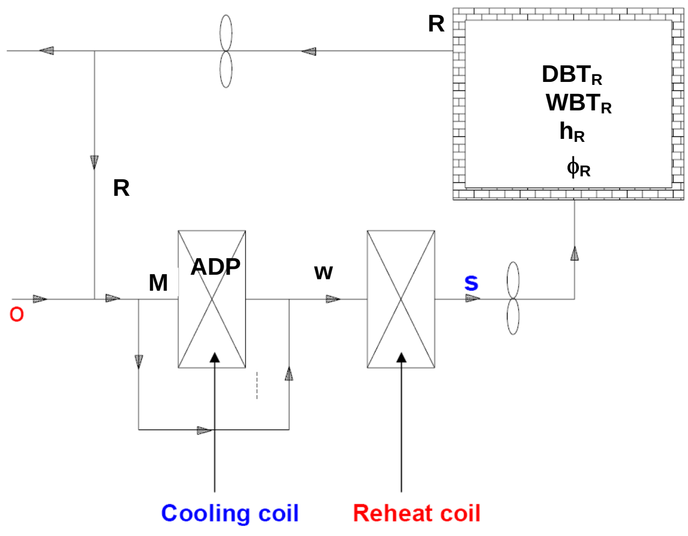


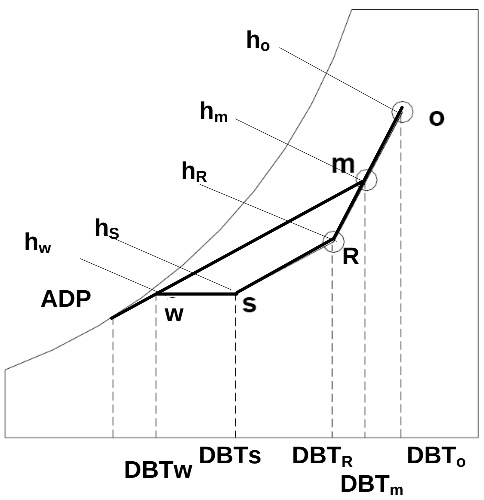
Figure 4 schematic of the summer air conditioning system with outdoor air and by pass factor

3. High latent cooling load applications (low RSHF):

When the latent load on the building is high due either to high outside humidity or due to large ventilation requirements (e.g. hospitals) or due to high internal latent loads (e.g. presence of kitchen or laundry), then the simple system discussed above leads to very low coil ADP. A low coil ADP indicates operation of the refrigeration system at low evaporator temperatures. Operating the system at low evaporator temperatures decreases the COP of the refrigeration system leading to higher costs. Hence a reheat coil is sometimes used so that the cooling coil can be operated at relatively high ADP, and at the same time the high latent load can also be taken care of. Figure 4 shows an air conditioning system with reheat coil along with the psychrometric representation of the process. As shown in the figure, in a system with reheat coil,

air is first cooled and dehumidified from point 'm' to point 'c' in the cooling coil and is then reheated sensibly to the required supply temperature ts using the reheat coil. If the supply temperature is specified, then the mass flow rate and state of the supply air and condition of the air after mixing can be obtained using equations given above. Since the heating process in the reheat coil is sensible, the process line c-s will be horizontal. Thus if the coil ADP is known, then one can draw the coil condition line and the intersection of this line with the horizontal line drawn from supply state 's' gives the condition of the air at the exit of the cooling coil. From this condition, one can calculate the load on the cooling coil using the supply mass flow rate and state of air after mixing. The capacity of the reheat coil is then obtained from energy balance across it.





Sheet No. Four

1. State which of the following statements are TRUE?

a) The purpose of psychrometric calculations is to fix the supply air conditions

b) The purpose of psychrometric calculations is to find the load on the building

c) In a 100% re-circulation system, the coil ADP is equal to room ADP

d) In a 100% re-circulation system, the coil ADP is less than room ADP

Ans.: a) and c)

2. State which of the following statements are TRUE?

a) In a 100% re-circulation system, the load on coil is equal to the load on building

b) In a system with outdoor air for ventilation, the load on building is greater than the load on coil

c) In a system with outdoor air for ventilation, the load on building is less than the load on coil

d) In a system with outdoor air for ventilation, the Coil ADP is less than room ADP

Ans.: a), c) and d)

3. Which of the following statements are TRUE?

a) Systems with reheat are used when the Room Sensible Heat Factor is low

b) Systems with reheat are used when the Room Sensible Heat Factor is high

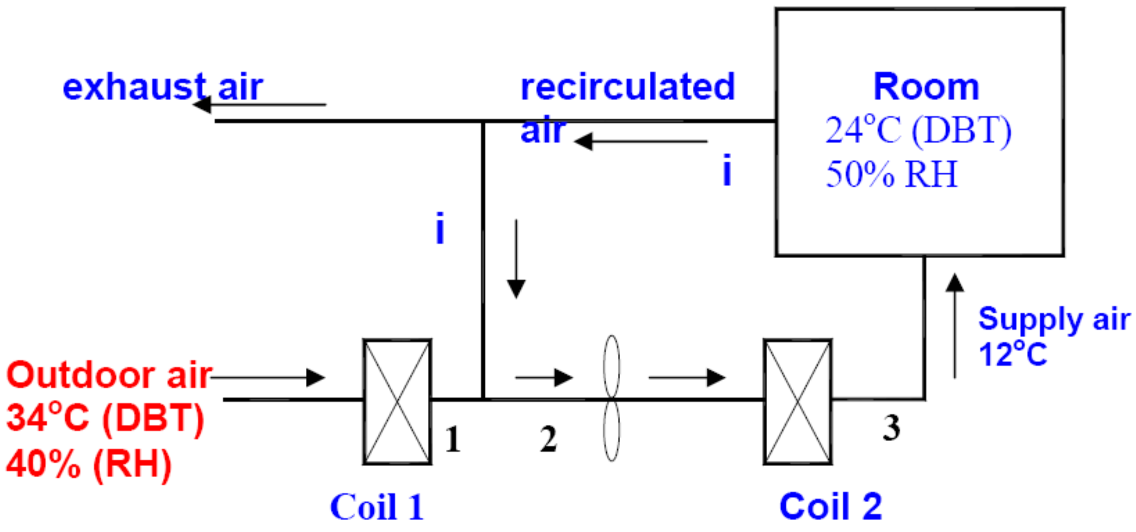
c) When reheat coils are used, the required coil ADP can be increased

d) When reheat coils are used, the required supply airflow rate increases

Ans.: a), c) and d)

4. A 100% outdoor summer air conditioning system has a room sensible heat load of 400 kW and a room latent heat load of 100 kW. The required inside conditions are 24oC and 50% RH, and the outdoor design conditions are 34oC and 40% RH. The air is supplied to the room at a dry bulb temperature of 14oC. Find a) the required mass flow rate of air b) moisture content of supply air, c) Sensible, latent heat loads on the coil, and d) The required cooling capacity of the coil, Coil Sensible Heat Factor and coil ADP if the by-pass factor of the coil is 0.2. Barometric pressure = 1 atm. Comment on the results.

5. A room is air conditioned by a system that maintains 25oC dry bulb and 50 % RH inside, when the outside conditions are 34oC dry bulb and 40% RH. The room sensible and latent heat gains are 60 kW and 12 kW respectively. As shown in the figure below, The outside fresh air first flows over a first cooler coil and is reduced to state 1 of 10oC dry bulb and a relative humidity of 85%. It is then mixed with recirculated air, the mixture (state 2) being handled by a fan, passed over a second cooler coil and sensibly cooled to 12oC dry bulb (state 3). The air is then delivered to the room. If the outside fresh air is used for dealing with the whole of the room latent heat gain and if the effects of fan power and duct heat gains are ignored, find: a) mass flow rates of outside fresh air and supply air; b) DBT and enthalpy of the air handled by the fan (state 2); and c) required cooling capacity of first cooler coil and second sensible cooler coil.



6. An air conditioned building has a sensible cooling load of 60 kW and latent load of 40 kW. The room is maintained at 24oC (DBT) and 50% RH, while the outside design conditions are: 34oC (DBT) and 40% RH. To satisfy the ventilation requirement, outdoor air is mixed with re-circulated air in the ratio of 1:3 (by mass). Since the latent load on the building is high, a reheat coil is used along with a cooling and dehumidifying coil. Air is supplied to the conditioned space at 14oC (DBT). If the bypass factor of the cooling coil is 0.15 and the barometric pressure is 101.325 kPa, find: a) Mass flow rate of supply air, b) Required cooling capacity of the cooling coil and heating capacity of the reheat coil
7. In an air-conditioned plant 3.5m^3/s of air at 27^{o}C DBT, 50%RH. The leaving condition of the air is 13 ^{o}C DBT and 90% RH. Calculate a-the refrigeration capacity in kW b-the rate of water removal from the air. (88kW, 0.0113kg/s)
8. A stream of out door air is mixed with a stream of return air in an air-conditioning system that operates at 101 kPa pressure. The flow rate of outdoor air is 2 kg/s and its condition is 35 ^{o}C DBT and 25 ^{o}C WBT. The flow rate of return air is 3 kg/s, and its condition is 24 ^{o}C and 50% RH determine a-the enthalpy of mixture b-humidity ratio of the mixture .(59.1kJ/kg, 0.01198kg/kg)
9. A winter air-conditioning system adds for humidification 0.00225 kg/s of saturated steam at 101kPa pressure to an airflow of 0.36 kg/s. the air is initially at 15 ^{o}C DBT and 20%RH. What are the DBT, WBT of the air leaving the humidifier. (16and 13.8^{o}C)

10. Outdoor air at 4.5 ^oC and 60% RH is heated and humidified by steam at 110 ^oC. the airflow rate is 14000l/s, and heat is added to the air at the rate of 440 kW while absorb 0.01 kg/s of steam. Determine the DBT& WBT at exit

11. Cold air at 10 ^oC DBT and 5 ^oC WBT is mixed with warm air at 25 ^oC DBT and 20 ^oC WBT in the ratio of 1:2 respectively. Find the mixture condition.

12. An evaporative cooler is able to cool air by 85% of the difference between the entering air DBT and WBT. If the inlet air is at 38 ^oC DBT and 20% RH. What is the outlet condition. How much water is evaporated if the air flow is 4700 l/s

13. Air is cooled from 27 ^oC DBT to the saturation at 13 ^oC outlet of cooling coil. How much water is removed, how much latent and sensible heat are removed, what is the SHR.

14. An office is occupied by 30persons who each produce 58 kW of sensible heat and 0.1 kg/h of moisture. The office is to be maintained at 22 ^oC and 50% RH. Conditioned of supply air at 15 ^oC DBT to meet the sensible and laten loads. What is the SHR to meet the load, what must the humidity ratio and mass flow rate of supply air.

15. An economizer must deliver 4700 l/s at 13.5 ^oC DBT. The return air is at 24 DBT and 50% RH, and the outside condition is 35 ^oC DBT and 40%RH. How much outside air is need and what is the mixed air humidity ratio.

16. What is the inlet humidity ratio to an ideal evaporative cooler? That produce saturated outlet air at 17 ^oC from 29 ^oC DBT inlet air. an adiabatic saturator with entering ait of 26 ^oC DBT has a leaving air temperature of 18 ^oC DBT. Compute the entering air humidity ratio and relative humidity.

17. Outdoor air is at 0^oC and 100% RH is heated with a low pressure hot water battery to 30 ^oC, identify the condition of the air for exit condition.

18. Air at 12 ^oC and 20% RH is heated through an increase of 25 ^oC and then adabatically humidified to 90% RH. Find the condition of end points.

19. Outdoor air at 28 ^oC DBT and 22 ^oC WBT passes a direct expansion cooler coil when the refrigerant evaporates at 5 ^oC. Assuming the air-side dew point of the coil is 5 ^oC and 100% contact factor is maintained. Find all the cndition data for the coil air-on and air off state.

20. an air handling unit receives re-circulated room air at 23 ^oC DBT and 50% RH and flow at 2 m^3/s, and fresh air at 5 ^oC and 30% RH with a flow rate of 0.05m^3/s, the mixed air is then heated to 35 ^oC DBT and the adiabatically saturated to 24 ^oC. calculate each point condition and the load of heater battery in kW.

21. an air handling unit mixes 0.8m3/s of fresh air at 32 ^oC DBT and 23 ^oC WBT, with 4m3/s of re-circulated room air at 22 ^oC DBT and 55% RH. The mixed air passes through a chilled water coiling coil whose dew point is 6 ^oC, incomplete contact between the air and coil surface causes 10% of mixed air to bypass the cooling effect. Calculate the capacity of cooling coil in kW and the rate of moisture removed.

Evaporative, winter and All Year Air Conditioning Systems

The specific objectives of this lecture are to:

1. Introduce evaporative cooling systems.
2. Classify evaporative cooling systems.
3. Discuss the characteristics of direct evaporative cooling systems.
4. Discuss the characteristics of indirect evaporative cooling systems.
5. Discuss the characteristics of multi-stage evaporative cooling systems.
6. Discuss advantages and disadvantages of evaporative cooling systems.
7. Discuss the applicability of evaporative cooling systems.
8. Describe winter air conditioning systems.
9. Describe all year air conditioning systems.

At the end of the lecture, the student should be able to:

1. Explain the working principle of direct, indirect and multi-stage evaporative cooling systems
2. Perform psychrometric calculations on evaporative cooling systems
3. List the advantages and disadvantages of evaporative cooling systems
4. Evaluate the applicability of evaporative cooling systems based on climatic conditions
5. Describe winter air conditioning systems and perform psychrometric calculations on these systems
6. Describe all year air conditioning systems.

Introduction to evaporative air conditioning systems

Evaporative cooling has been in use for many centuries in countries such as Iraq for cooling water and for providing thermal comfort in hot and dry regions. This system is based on the principle that when moist but unsaturated air comes in contact with a wetted surface whose temperature is higher than the dew point temperature of air, some water from the wetted surface evaporates into air. The latent heat of evaporation is taken from water, air or both of them. In this process, the air loses sensible heat but gains latent heat due to transfer of water vapour. Thus the air gets cooled and humidified. The cooled and humidified air can be used for providing thermal comfort.

Classification of evaporative cooling systems:

The principle of evaporative cooling can be used in several ways.

1. Direct evaporation process, or
2. Indirect evaporation process, or

Direct evaporative cooling systems:

In direct evaporative cooling, the process or conditioned air comes in direct contact with the wetted surface, and gets cooled and humidified. Figure 1 shows the schematic of an elementary direct, evaporative cooling system and the process on a psychrometric chart. As shown in the figure, hot and dry outdoor air is first filtered and then is brought in contact with the wetted surface or spray of water droplets in the air washer. The air gets cooled and dehumidified due to simultaneous transfer of sensible and latent heats between air and water (process o-s). The cooled and humidified air is supplied to the conditioned space, where it extracts the sensible and latent heat from the conditioned space (process s-R). Finally the air is exhausted at state i. In an ideal case when the air washer is perfectly insulated and an infinite amount of contact area is available between air and the wetted surface, then the cooling and humidification process follows the constant wet bulb temperature line and the temperature at the exit of the air washer is equal to the wet bulb temperature of the entering air (WBT_O), i.e., the process becomes an adiabatic saturation process. However, in an actual system the temperature at the exit of the air washer will be higher than the inlet wet bulb temperature due to heat leaks from the surroundings and also due to finite contact area. One can define the saturation.

$$E = \frac{DBT_0 - DBT_s}{DBT_0 - WBT_o}$$

Depending upon the design aspects of the evaporative cooling system, the effectiveness may vary from 50% (for simple drip type) to about 90% (for efficient spray pads or air washers).

The amount of supply air required $\dot{m}_a$ can be obtained by writing energy balance equation for the conditioned space, i.e.,

$$\dot{m}_a = \frac{Q_T}{h_R - h_S}$$

Compared to the conventional refrigeration based air conditioning systems, the amount of airflow rate required for a given amount of cooling is much larger in case of evaporative cooling systems. As shown by the above equation and also from Fig.1, it is clear that for a given outdoor dry bulb temperature, as the moisture content of outdoor air increases, the required amount of supply air flow rate increases rapidly. And at a threshold moisture content value, the evaporative coolers cannot provide comfort as the cooling and humidification line lies above the conditioned space condition 'i'. Thus evaporative coolers are very useful essentially in dry climates, whereas the conventional refrigeration based air conditioning systems can be used in any type of climate

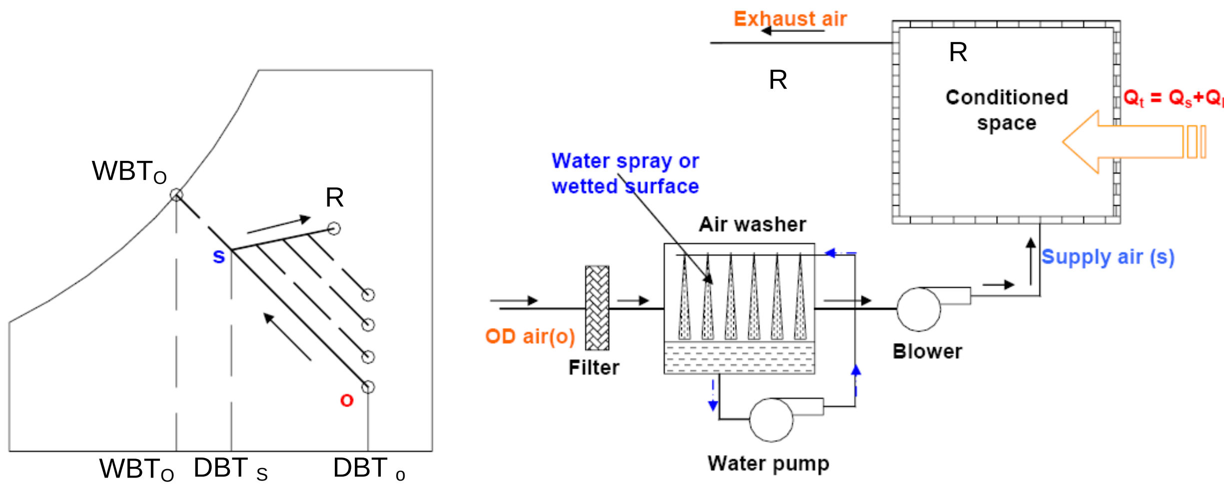


Fig.1: A direct, evaporative cooling system

Indirect evaporative cooling system:

Figure 2 shows the schematic of a basic, indirect evaporative cooling system and the process on a psychrometric chart. As shown in the figure, in an indirect evaporative cooling process, two streams of air – primary and secondary are used. The primary air stream becomes cooled and humidified by coming in direct contact with the wetted surface (o-P), while the secondary stream which is used as supply air to the conditioned space, decreases its temperature by exchanging only sensible heat with the cooled and humidified air stream (o-s). Thus the moisture content of the supply air remains constant in an indirect evaporative cooling system, while its temperature drops. However, since the moisture content of supply air remains constant in an indirect evaporation process, this may provide greater degree of comfort in regions with higher humidity ratio. The commercially available indirect evaporative coolers have saturation efficiency as high as 80%.

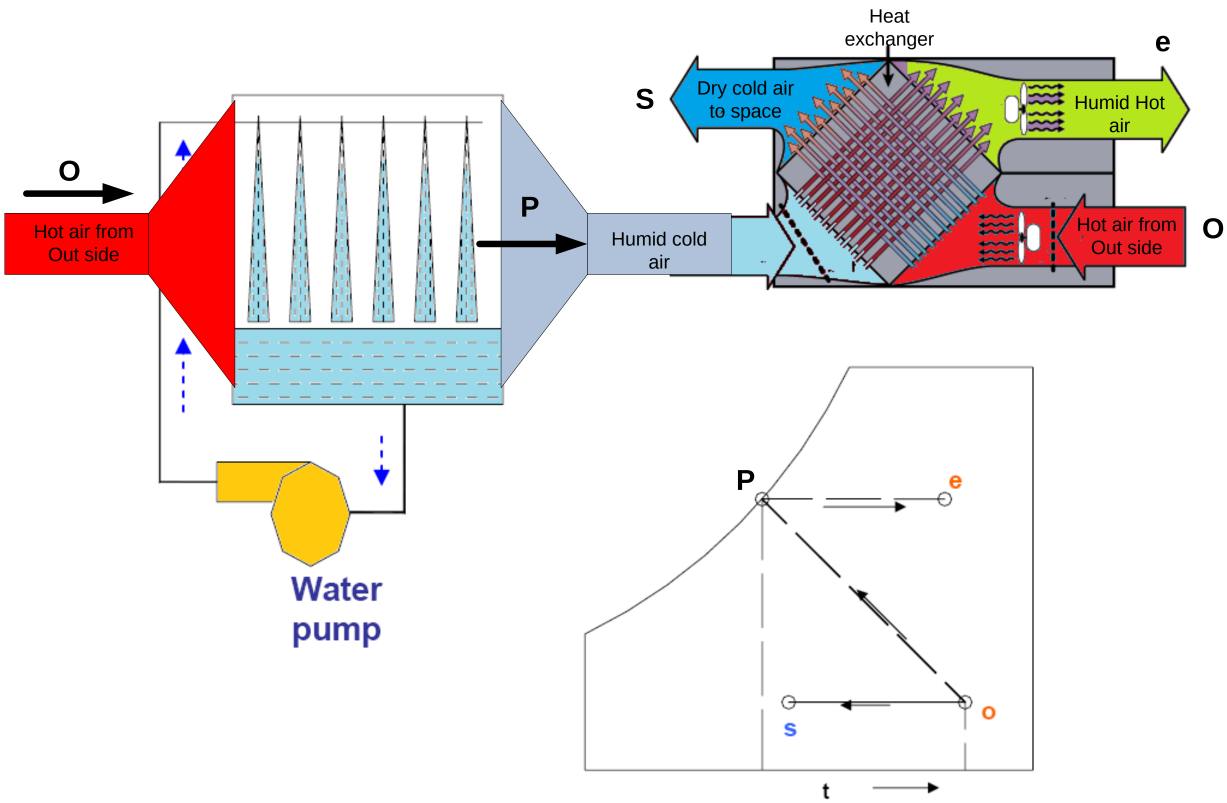


Figure 2 indirect evaporative cooling system

Winter Air Conditioning Systems

In winter the outside conditions are cold and dry. As a result, there will be a continuous transfer of sensible heat as well as moisture (latent heat) from the buildings to the outside. Hence, in order to maintain required comfort conditions in the occupied space an air conditioning system is required which can offset the sensible and latent heat losses from the building. Air supplied to the conditioned space is heated and humidified in the winter air conditioning system to the required level of temperature and moisture content depending upon the sensible and latent heat losses from the building. In winter the heat losses from the conditioned space are partially offset by solar and internal heat gains. Thus in a conservative design of winter A/C systems, the effects of solar radiation and internal heat gain are not considered. Heating and humidification of air can be achieved by different schemes. Figure 3 shows one such scheme along with the cycle on psychrometric chart. As shown in the figure, the mixed air (mixture of return and outdoor air) is first pre-heated (m-w1) in the pre-heater, then humidified using a humidifier or an air washer (w1-w2) and then finally reheated in the reheater (w2-s). The reheated air at state 's' is supplied to the conditioned space. The flow rate of supply air should be such that when released into the conditioned space at state 's', it should be able to maintain the conditioned space.

The humidification of air can be achieved in several ways, e.g. by bringing the air in contact with a wetted surface, or with droplets of water as in an air washer, by adding aerosol sized water droplets directly to air or by direct addition of dry saturated or superheated steam. Humidification by direct contact with a wetted surface or by using an air washer are not recommended for comfort applications or for other applications where people are present in the conditioned space due to potential health hazards by the presence of micro-organisms in water. The most common method of humidifying air for these applications is by direct addition of dry steam to air. When air is humidified by contact with wetted surface as in an air washer, then temperature of air decreases as its humidity increases due to simultaneous transfer of sensible and latent heat. If the air washer functions as an adiabatic saturator, then humidification proceeds along the constant wet bulb temperature line.

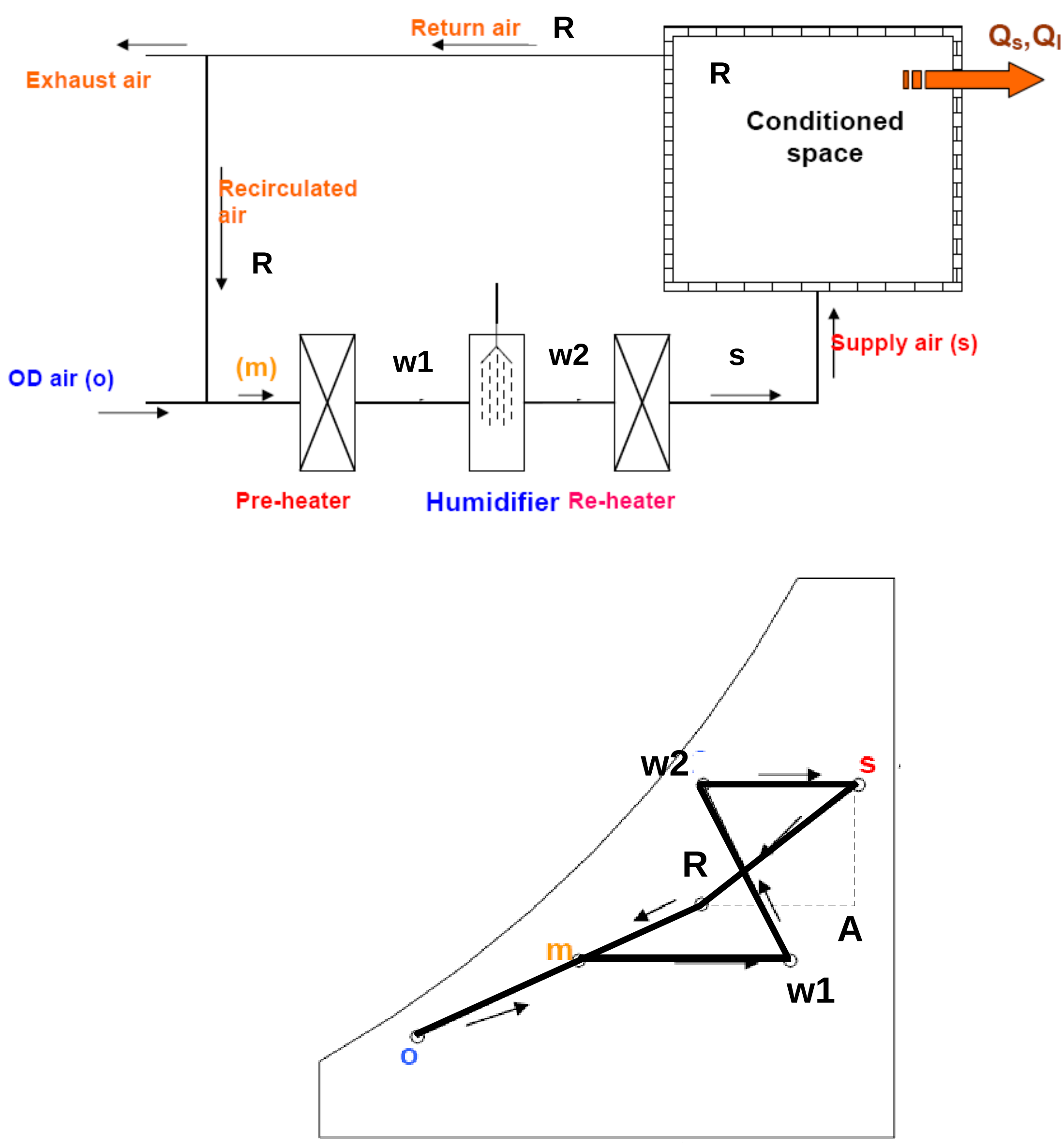


Figure *A winter air conditioning system with a pre-heater*

By applying energy balance across the conditioned space, at steady state, the sensible and latent heat losses from the building can be written as:

Pre- heater load: **<u>Room load</u>**

$$Q_{sp} = \dot{m}_a . c_p (DBT_{w1} - DBT_m)$$
$$= \dot{m}_a . (h_{w1} - h_m)$$

Re- heater load:

$$Q_{sre} = \dot{m}_a . c_p (DBT_S - DBT_{w2})$$
$$= \dot{m}_a . (h_S - h_{w1})$$

Sensible load

$$Q_{sr} = \dot{m}_a . c_p (DBT_S - DBT_R)$$
$$= \dot{m}_a . (h_A - h_R)$$

Total load

$$Q_{Tr} = \dot{m}_a . (h_S - h_R)$$

Latent load

$$Q_{sr} = 2501 . \dot{m}_a . (g_S - g_R)$$
$$= \dot{m}_a . (h_S - h_A)$$

Figure 4 shows another scheme that can also be used for heating and humidification of air as required in a winter air conditioning system. As shown in the figure, this system does not consist of a pre-heater. The mixed air is directly humidified using an air washer (**m-w**) and is then reheated (w-**s**) before supplying it to the conditioned space. Though this system is simpler compared to the previous one, it suffers from disadvantages such as possibility of water freezing in the air washer when large amount of cold outdoor air is used and also from health hazards to the occupants if the water used in the air washer is not clean. Hence this system is not recommended for comfort conditioning but can be used in applications where the air temperatures at the inlet to the air washer are above 0^oC and the conditioned space is used for products or processes, but not for providing personnel comfort.

Actual winter air conditioning systems, in addition to the basic components shown below, consist of fans or blowers for air circulation and filters for purifying air. The fan or blower introduces sensible heat into the air stream as all the electrical power input to the fan is finally dissipated in the form of heat.

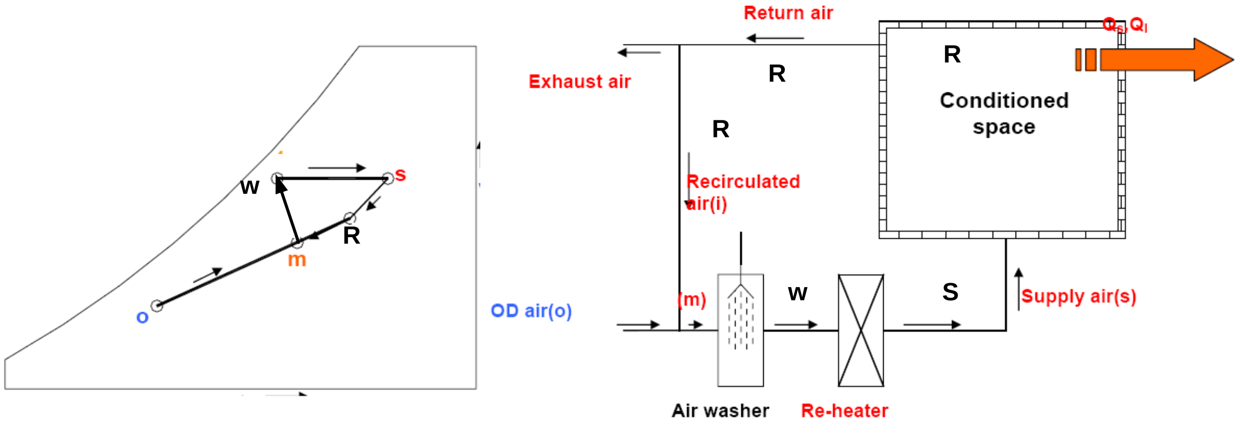


Figure 4 another scheme that can also be used for heating and humidification of air.

Sheet No. Five

1. Which of the following statements are TRUE?

a) Evaporative cooling systems are attractive for hot and humid climates

b) Evaporative cooling systems are attractive for hot and dry climates

c) Evaporative cooling systems are ideal for comfort applications

d) Evaporative cooling systems are ideal for several industrial applications

Ans.: b) and d)

2. Which of the following statements are TRUE?

a) In a direct evaporative cooling system, the lowest possible temperature is the wet bulb temperature corresponding to the outdoor air

b) In a direct evaporative cooling system, the lowest possible temperature is the dew point temperature corresponding to the outdoor air

c) In a direct evaporative cooling system, cooled and humidified air is supplied to the conditioned space

d) In a direct evaporative cooling system, cooled and dehumidified air is supplied to the conditioned space

Ans.: a) and c)

3. Which of the following statements are TRUE?

a) In an indirect evaporative cooling system, the air supplied to the conditioned space is at a lower temperature, but higher humidity ratio

b) In an indirect evaporative cooling system, the air supplied to the conditioned space is at a lower temperature and at a humidity ratio corresponding to the outdoor air

c) Compared to direct evaporative cooling systems, it is possible to achieve lower supply air temperatures in simple indirect evaporative coolers

d) In multi-stage evaporative cooling systems, it is possible to cool the air to a temperature lower than the entering air WBT

Ans.: b) and d)

4. Which of the following statements are TRUE?

a) Evaporative cooling systems are environment friendly

b) Evaporative cooling systems offer lower initial and lower running costs

c) Evaporative cooling systems are easier to maintain and fabricate

d) Evaporative systems provide better control on indoor climate

Ans.: a), b) and c)

5. Which of the following statements are TRUE?

a) Direct evaporative cooling systems are attractive in places where the summer design WBT is greater than 24oC

b) Direct evaporative cooling systems are attractive in places where the summer design WBT is less than 24oC

c) Indirect evaporative cooling systems can be used over an extended range of climatic conditions

d) A combination of evaporative cooling system with conventional air conditioning system can offer better overall performance

Ans.: b), c) and d)

6. Which of the following statements are TRUE?

a) In winter air conditioning systems, heated and dehumidified air is supplied to the conditioned space

b) In winter air conditioning systems, heated and humidified air is supplied to the conditioned space

c) A pre-heater is recommended in winter air conditioning systems to improve overall efficiency of the system

d) A pre-heater is recommended in winter air conditioning systems to prevent freezing of water in the humidifier and for better control

Ans.: b) and d)

7. Which of the following statements are TRUE?

a) When humidification is done using an air washer, the temperature of air drops during humidification

b) When humidification is done using an air washer, the temperature of air rises during humidification

c) When humidification is carried out by adding dry steam, the temperature of air remains close to the WBT of entering air

d) When humidification is carried out by adding dry steam, the temperature of air remains close to the DBT of entering air

Ans.: a) and d)

8. Which of the following statements are TRUE?

a) An all year air conditioning system can be used either as a summer air conditioning system or as a winter air conditioning system

b) When an all year air conditioning system is used during summer, the heaters are always switched-off

c) When an all year air conditioning system is used during winter, the cooling and dehumidification coils are switched-off

d) In an all year air conditioning systems, the blowers are always on

Ans.: a), c) and d)

9. A large warehouse located at an altitude of 1500 m has to be maintained at a DBT of 27oC and a relative humidity of 50% using a direct evaporative cooling system. The outdoor conditions are 33oC (DBT) and 15oC (WBT). The cooling load on the warehouse is 352 kW. A supply fan located in the downstream of the evaporative cooler adds 15 kW of heat. Find the required mass flow rate of air. Assume the process in evaporative cooler to follow a constant WBT.

10. A winter air conditioning system maintains a building at 21oC and 40% RH. The outdoor conditions are 0oC (DBT) and 100% RH. The sensible load load on the building is 100 kW, while the latent heating load is 25 kW. In the air conditioning system, 50% of the outdoor air (by mass) is mixed with 50% of the room air. The mixed air is heated in a pre-heater to 25oC and then required amount of dry saturated steam at 1 atm. pressure is added to the pre-heated air in a humidifier. The humidified air is then heated to supply temperature of 45oC and is then supplied to the room. Find a) The required mass flow rate of supply air, b) Required amount of steam to be added, and c) Required heat input in pre-heater and re-heater. Barometric pressure = 1atm..

General Examples

The aim of this section of the notes is to allow students to size air conditioning plant such as; cooling coil, heater battery and humidifier.

The notes are divided into several sections as follows:

<u>EXAMPLES OF PSYCHROMETRIC PROPERTIES</u>

<u>AIR CONDITIONING PLANT FOR SUMMER & WINTER</u>

<u>TYPICAL AIR CONDITIONING PROCESSES</u>

<u>ANNOTATION AND ROOM RATIO</u>

<u>SUMMER AND WINTER CYCLES</u>

<u>EXAMPLES</u>

A diagram of an air conditioning system is shown in schematic form in the section entitled AIR CONDITIONING PLANT FOR SUMMER & WINTER.

Before sizing takes place the student should also understand the processes that take place in air conditioning systems.

There are four basic processes for summer and winter air conditioning systems.

The following basic processes are explained:

1. Mixing
2. Sensible Cooling and Heating
3. Cooling with Dehumidification
4. Humidification

The section on Typical Air Conditioning Processes shows winter and summer schematic diagrams and psychrometric charts.

There are some more details that may be useful to the designer of air conditioning systems.

Further information is as follows: Annotation, Room ratio

When the processes have been superimposed onto a psychrometric chart then calculations may commence.

These are as detailed in the following sections of the notes.

Summer and Winter Cycles

1. Summer cycle psychrometrics
2. Summer cycle calculations

3. Winter cycle psychrometric
4. Winter cycle calculations
5. Duct and Fan gains.

The final section is seven examples of plant sizing using psychrometric charts.

Example 1. Summer Cycle

A room is to be maintained at 22^{o}C dry-bulb temperature, 50% saturation, when the sensible heat gain is 10.8 kW in summer. The latent heat gain is 7.2 kW. Determine the cooling coil and reheater outputs required by using a psychrometric chart if the plant schematic is as shown below.

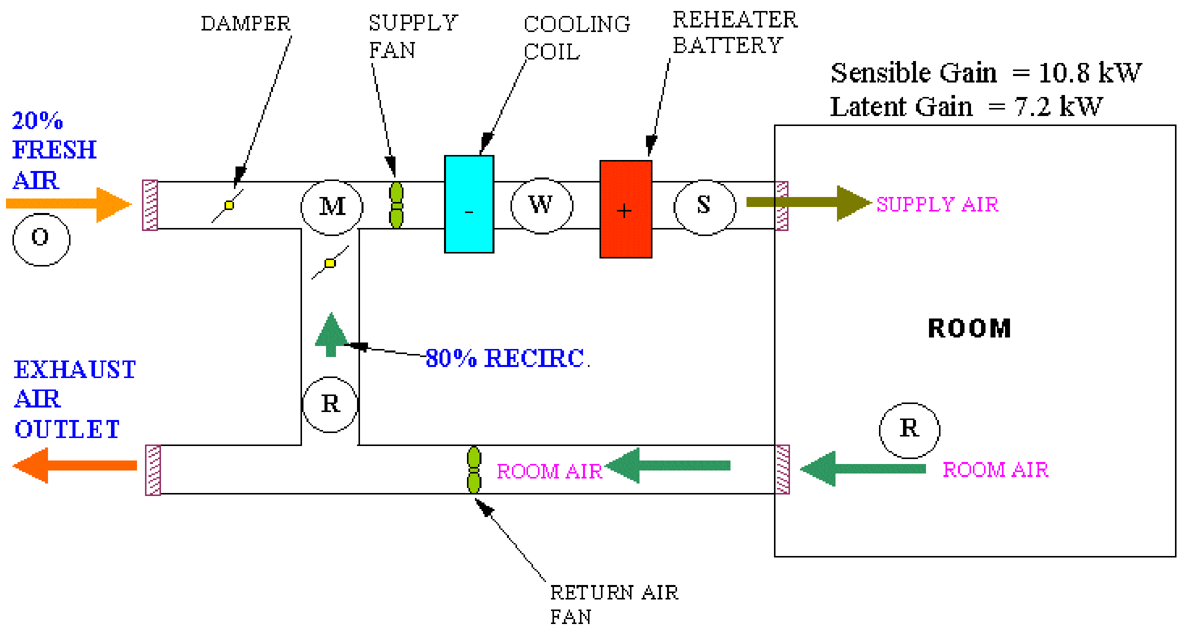
DATA:

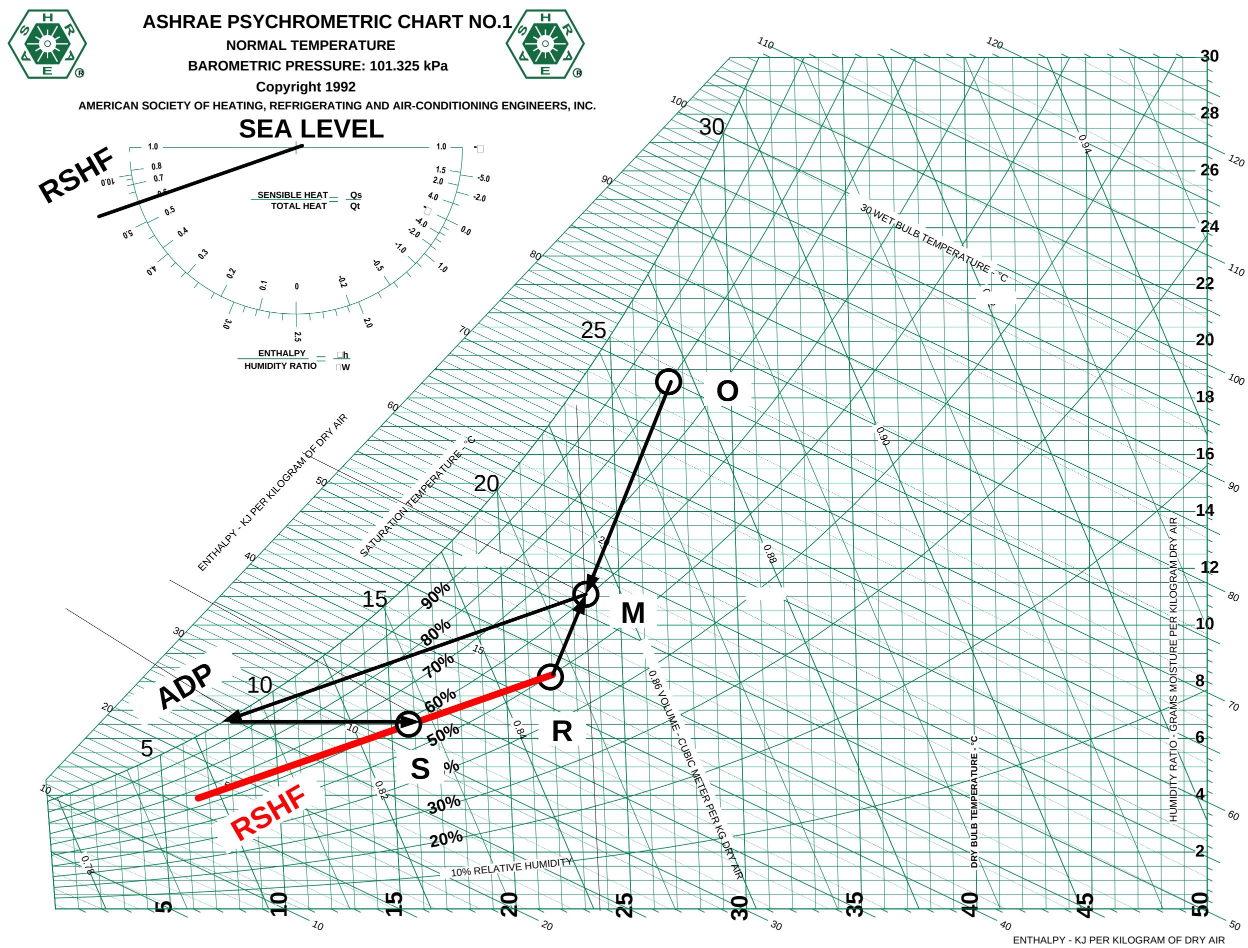
Outdoor condition is 28^{o}C, 80% saturation.

The outdoor air and recirculated air ratio is 20%/80%.

The Apparatus Dew Point ADP is 8^{o}C

Neglect the cooling coil contact factor.





Mixing conditions

$$DBT_M = \frac{\dot{m}_R . DBT_R + \dot{m}_o . DBT_o}{\dot{m}_R + \dot{m}_o} = \frac{0.8 \times 22 + 0.2 \times 28}{0.8 + 0.2} = 23.2\ ^oC$$

Locate point M by plot a vertical line from DBT of 23.2, the intersection of the vertical line with the line OR located point M.

From M draw a line to the ADP of 8 ^{o}C.

Find RSHF as follow:

$$RSHF = \frac{Q_{sR}}{Q_{sR} + Q_{lR}} = \frac{10.8}{10.8 + 7.2} = 0.6$$

Draw line parallel to the line of RSHF from R, The condition S must be lie on the line of RSHF.

Thus the supply conditions are DBT= 16^{o}C, R.H= 58%, g= 6.5 gw/kgair

The mass flow rate of air can be calculated as:

$$\dot{m}_a = \frac{Q_S}{c_p.(DBT_R - DBT_S)} = \frac{10.8}{1.005.(22-16)} = 1.79\ kg/s$$

Cooling coil load

$$Q_T = \dot{m}_a.(h_M - h_w) = 1.79.(51 - 25.5) = 45.6\ kW$$

Heater load

$Q_T = \dot{m}_a.(h_M - h_w)$=1.79(33.5- 25.5)= 14.32 kW.

Example 2. Winter Cycle

A room has a 18.0 kW sensible heat loss in winter and a 4.5 kW latent heat gain from the occupants. Determine the supply air temperature and heater battery load using the following information.

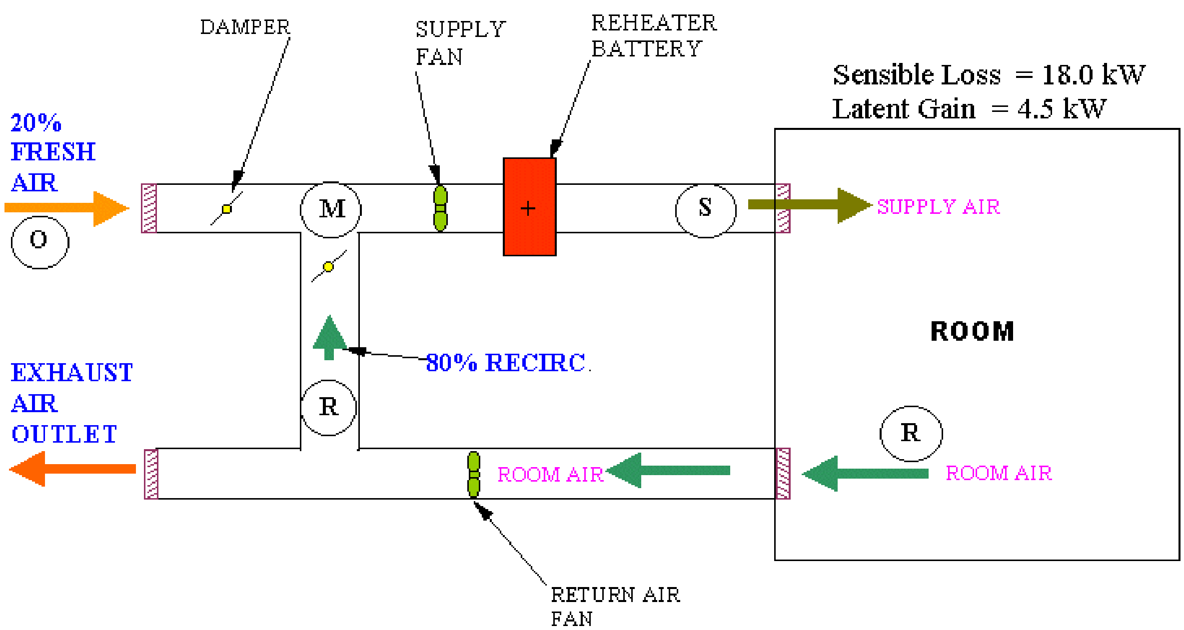
DATA:

Indoor condition: 21^{o}C dry-bulb temperature, 50% saturation.

Outdoor condition: 0^{o}C d.b., 80% saturation.

The outdoor air and recirculated air ratio is 20%/80%.

No preheating or humidification takes place in this simplified example



Mixing conditions

$$DBT_M = \frac{\dot{m}_R . DBT_R + \dot{m}_o . DBT_o}{\dot{m}_R + \dot{m}_o} = \frac{0.8 \times 21 + 0.2 \times 0}{0.8 + 0.2} = 16.8\ ^oC$$

Locate point M by plot a vertical line from DBT of 16.8, the intersection of the vertical line with the line OR located point M.

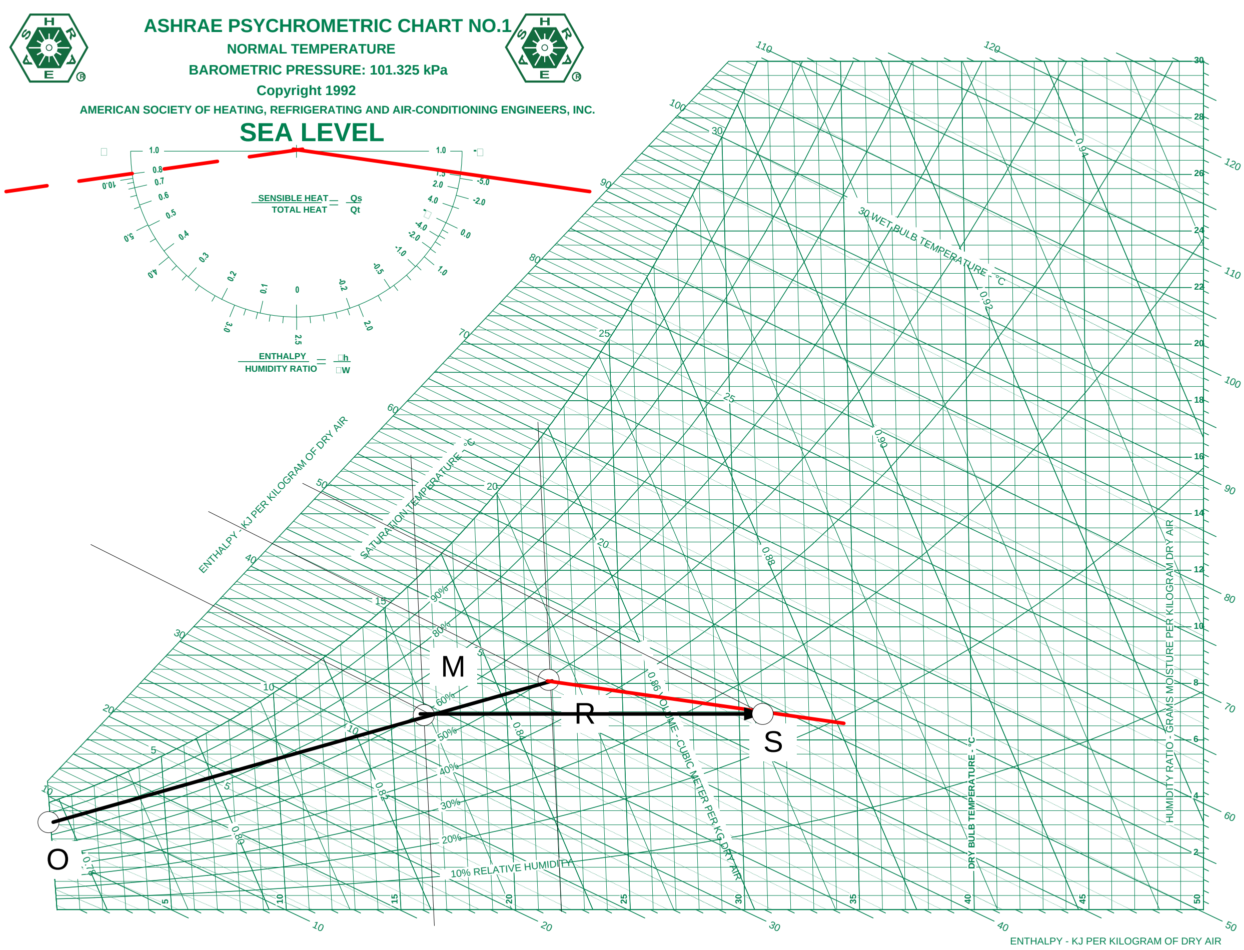
From M draw a line to the ADP of 8 ^{o}C.

Find RSHF as follow:

$$RSHF = \frac{Q_{sR}}{Q_{sR} + Q_{lR}} = \frac{18}{18 + 4.5} = 0.8$$

Draw line parallel to the line of RSHF from R, The condition S must be lie on the line of RSHF.

Thus the supply conditions are DBT= 30 ^{o}C, R.H= 25%, g= 6.1 gw/kgair



The mass flow rate of air can be calculated as:

$$\dot{m}_a = \frac{Q_S}{c_p.(DBT_s - DBT_R)} = \frac{18}{1.005.(30-21)} = 1.99\ kg/s$$

Heating coil load

$$Q_T = \dot{m}_a.(h_M - h_S) = 1.99.(49.5 - 41) = 16.91\ kW$$

Example 3. Summer Cycle (Cooling Coil contact factor)

An office is to be maintained at 22^{o}C dry-bulb temperature, 50% saturation in summer.

The sensible heat gain is 8.0 kW.

The latent heat gain is 2.0 kW.

Determine the cooling coil and reheater outputs required by using a psychrometric chart if the plant schematic is as shown below.

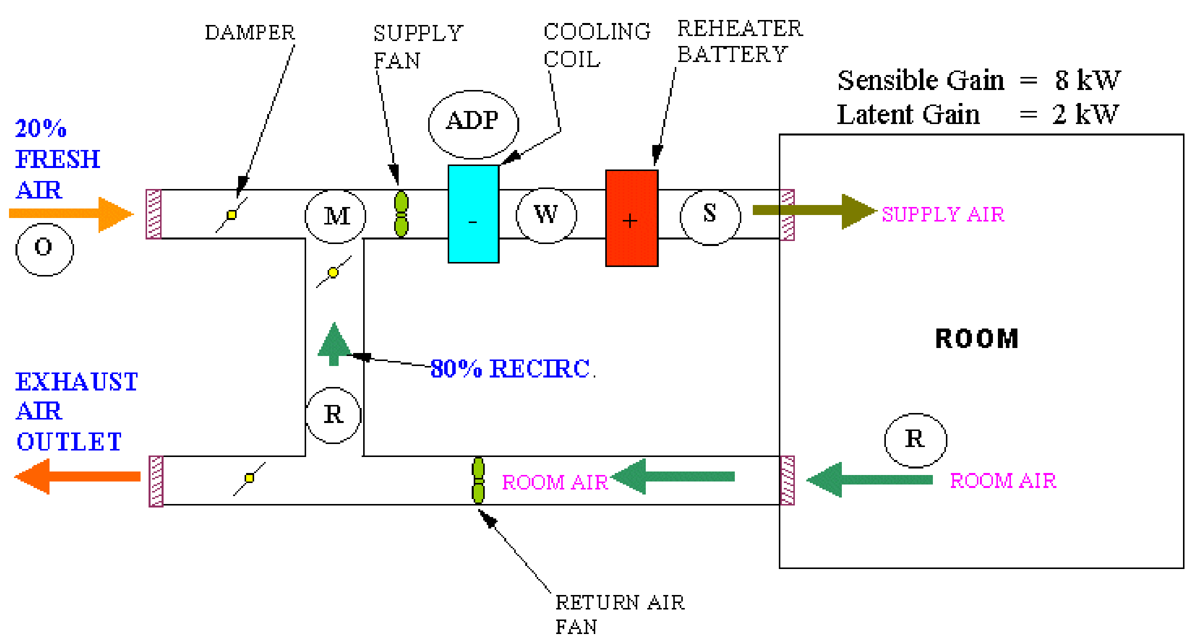
DATA:

Outdoor condition is 28^{o}C, 80% saturation.

The outdoor air and recirculated air ratio is; 20% / 80%.

The Apparatus Dew Point ADP is 8^{o}C

The cooling coil contact factor is 0.8.



Mixing conditions

$$DBT_M = \frac{\dot{m}_R.DBT_R + \dot{m}_o.DBT_o}{\dot{m}_R + \dot{m}_o} = \frac{0.8 \times 22 + 0.2 \times 28}{0.8 + 0.2} = 23.2\ ^oC$$

Locate point M by plot a vertical line from DBT of 23.2, the intersection of the vertical line with the line OR located point M.

From M draw a line to the ADP of 8 ^{o}C.

Since the contact factor of the coil is 0.8 therefor the condition of exit air from coil is far away from ADP, the condition of leaving air from coil is calculated as follow:

$$\beta = \frac{DBT_M - DBT_W}{DBT_M - ADP}$$

$$0.8 = \frac{23.2 - DBT_w}{23.2 - 8}$$

$$DBT_w = 11.04\ \ ^oC$$

Find RSHF as follow:

$$RSHF = \frac{Q_{sR}}{Q_{sR} + Q_{lR}} = \frac{8}{8+2} = 0.8$$

Draw the RSHF from R, the intersection of the horizontal line drawn from W with RSHF line represent the supply condition S.

Supply conditions S are, DBT =15 ^{o}C, RH=50%

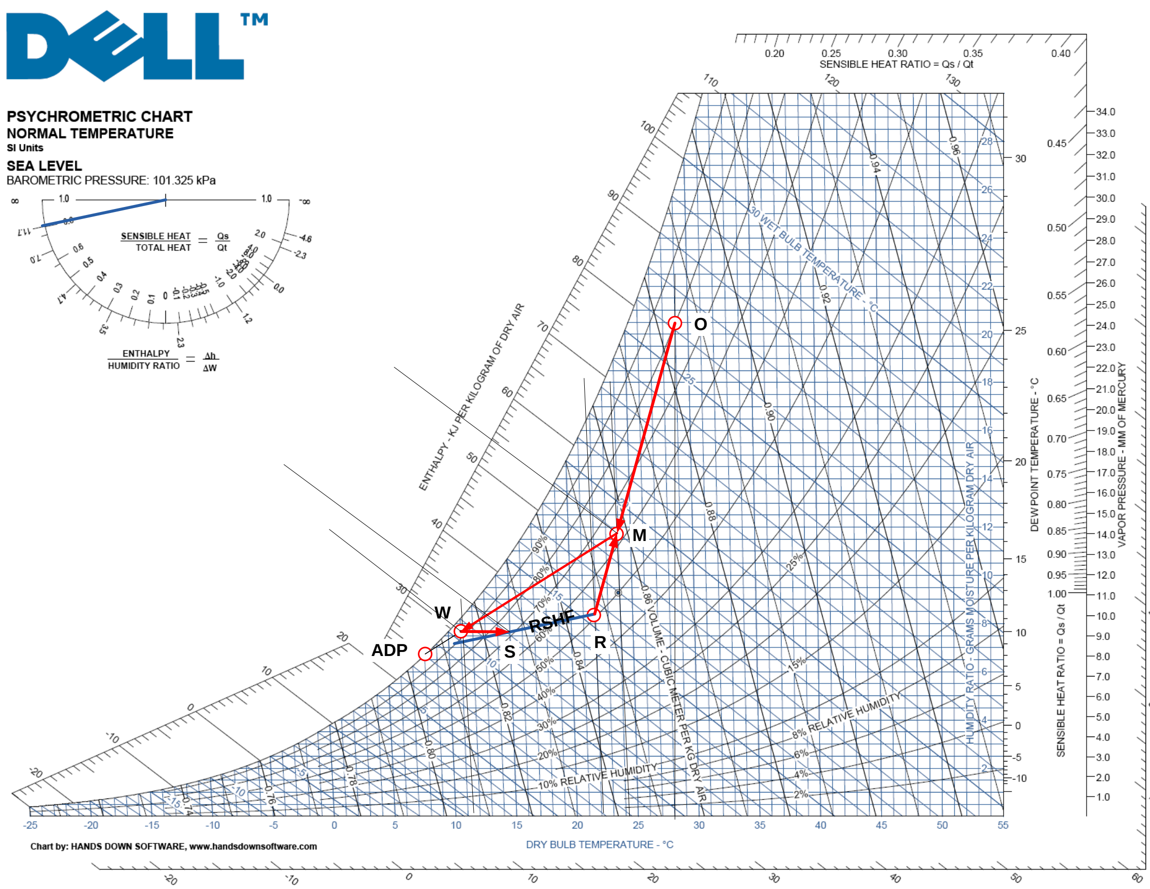
$$\dot{m}_a = \frac{Q_S}{c_p.(DBT_s - DBT_R)} = \frac{8}{1.005.(28-22)} = 1.32\ kg/s$$

Cooling coil load

$$Q_T = \dot{m}_a.(h_M - h_W) = 1.32.(54 - 30.5) = 31.2\ kW$$

Heating coil load

$$Q_T = \dot{m}_a.(h_W - h_S) = 1.32.(30.5 - 34.5) = 5.28\ kW$$



Example 4. Winter Cycle with Humidifier

An conference room is to be maintained at 21°C dry-bulb temperature, 50% saturation in winter. The sensible heat loss for the room is 17.0 kW. The latent heat gain is 40 Watts per person Determine the preheater and reheater outputs required and the amount of moisture to be added at the humidifier in litre/hour, by using a psychrometric chart if the plant schematic is as shown below.

DATA:

Outdoor condition is -2°C, 80% saturation.

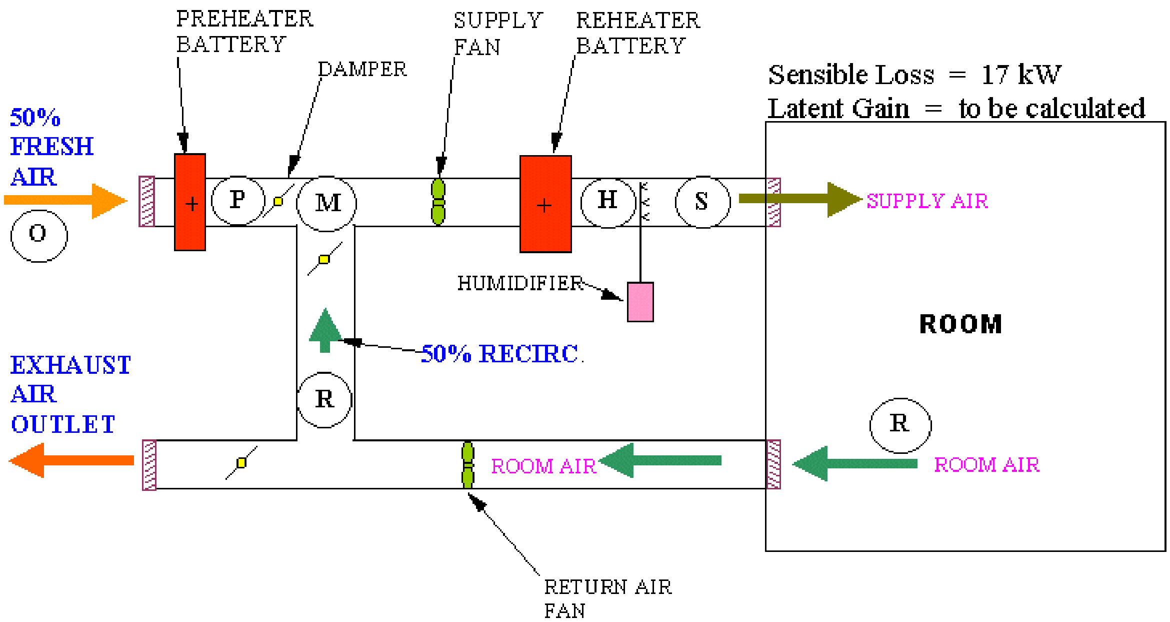
The outdoor air and recirculated air ratio is 50%/50%.

Maximum occupancy is 250 people.

The preheater off coil temperature is 5°C.

Supply air quantity is 8 air changes per hour, at room condition.

Room volume is 20 x 12 x 4m high = 960 m^3.



Latent heat gain = 40 x 250 = 10000 W = 10 kW

Volume flow rate of supply air:

Room Volume = 20 x 12 x 4m high = 960 m^3

Volume flow rate = 8 x 960 = 7680 m^3/hr = 2.13 lit./s

$$DBT_M = \frac{\dot{m}_R . DBT_R + \dot{m}_P . DBT_P}{\dot{m}_R + \dot{m}_P} = \frac{0.5 \times 21 + 0.5 \times 5}{0.8 + 0.2} = 13\ ^oC$$

Locate point M by plot a vertical line from DBT of 13^{o}C, the intersection of the vertical line with the line OR located point M.

Room sensible heat factor

$$RSHF = \frac{Q_{sR}}{Q_{sR} + Q_{lR}} = \frac{17}{17 + 10} = 0.63$$

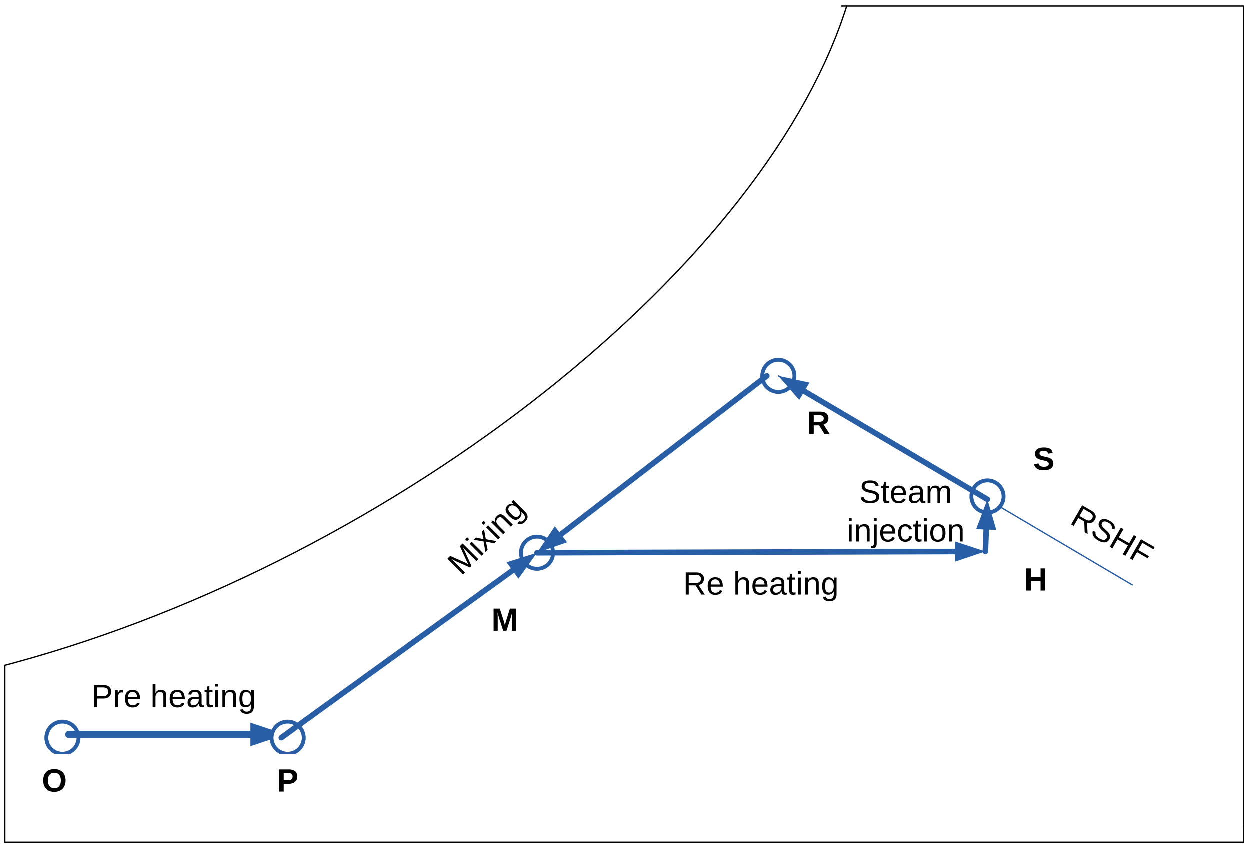
Draw the RSHF at 0.63 at flip the line, the move the RSHF line to room condition R.

Mas flow rate of air

$$\dot{m}_a = \frac{\dot{V}_a}{v_R} = \frac{2.13}{0.844} = 2.52\ kg/s$$

$$Q_s = \dot{m}_a . c_p (DBT_s - DBT_R)$$

$$17 = 2.52 \times 1.005 (DBT_s - 21)$$



$DBT_s = 27.7\ ^oC$

Pe-heater load:

$$Q_T = \dot{m}_a . c_p (DBT_P - DBTh_O) = \frac{2.52}{2} \times 1.005. (5 - (-2)) = 8.86\ kW$$

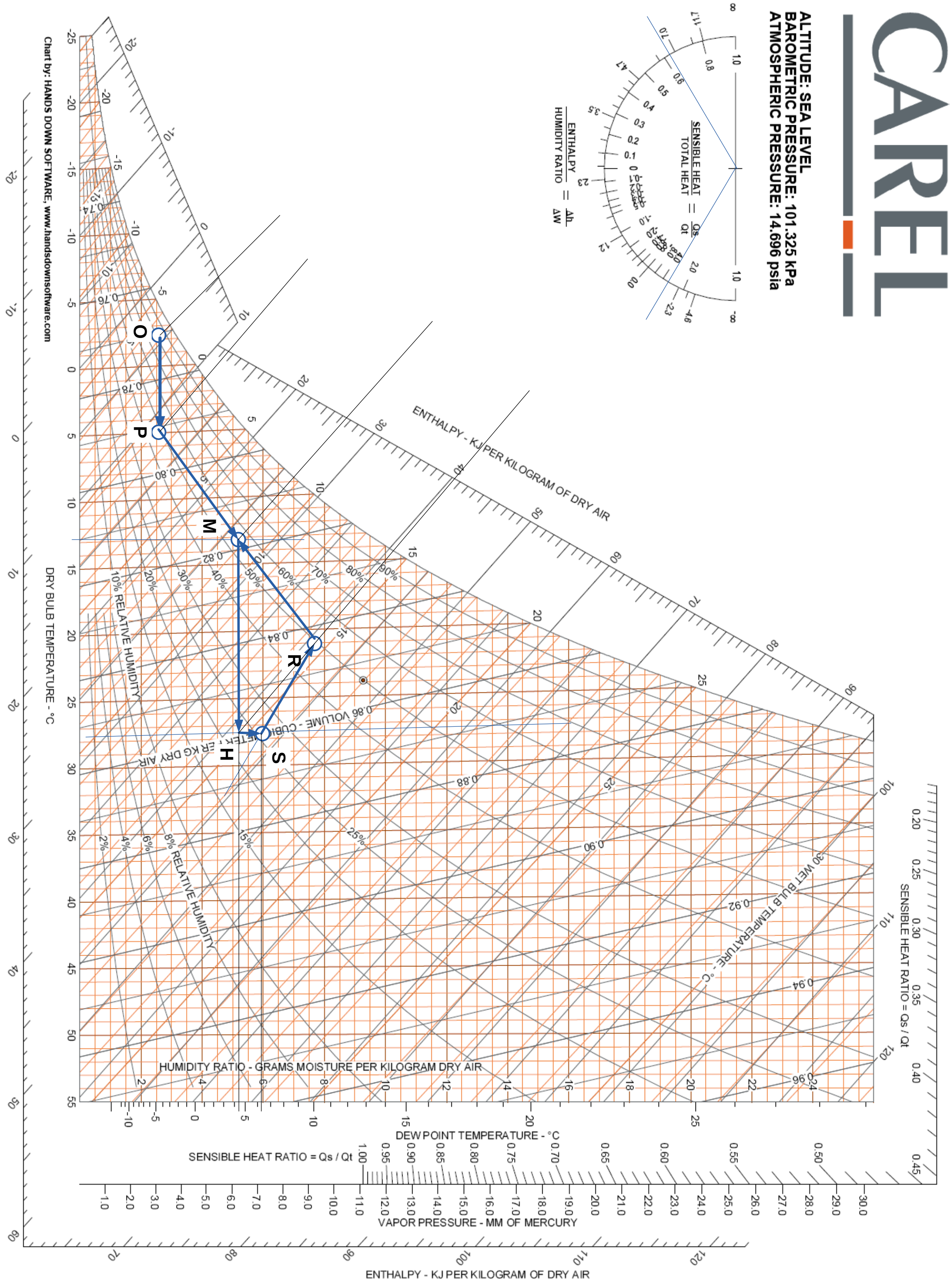
Re- heater load

$$Q_T = \dot{m}_a . c_p (DBT_H - DBTh_M) = 2.52 \times 1.005. (27.7 - 13)) = 27.3\ kW$$

Moisture added

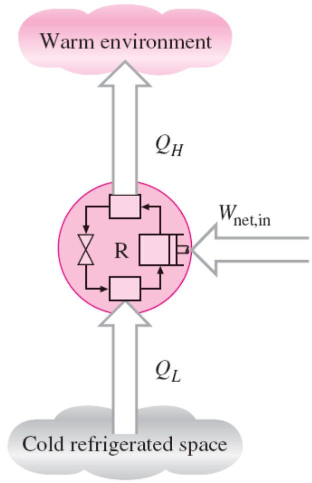
$$m_w = \dot{m}_a . (g_s - g_H) = 2.52. (6.1 - 4.1)) = 5.04\ g$$

$$m_w = \frac{5.04\ g}{1000} \times 3600 = 18.44\ lit/hr$$

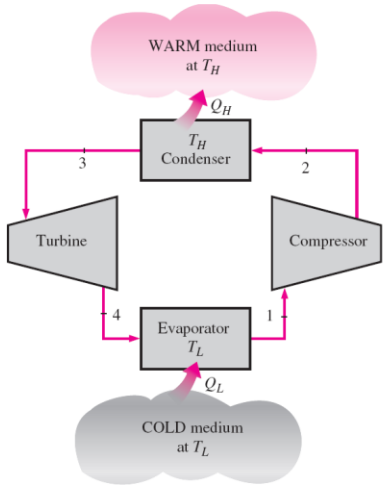

CAREL
ALTITUDE: SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa
ATMOSPHERIC PRESSURE: 14.696 psia
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY - KJ PER KILOGRAM OF DRY AIR
DRY BULB TEMPERATURE - °C
10% RELATIVE HUMIDITY
8% RELATIVE HUMIDITY
WET BULB TEMPERATURE - °C
0.86 VOLUME - CUBIC METER PER KG DRY AIR
HUMIDITY RATIO - GRAMS MOISTURE PER KILOGRAM DRY AIR
DEW POINT TEMPERATURE - °C
SENSIBLE HEAT RATIO = Qs / Qt
VAPOR PRESSURE - MM OF MERCURY
Chart by: HANDS DOWN SOFTWARE, www.handsdownsoftware.com
O
P
M
R
H
S

Vapour compression cycle

Refrigeration in the engineering sense, means maintaining a system at a temperature less than the temperature of the surroundings. This will not occur naturally, so a device must be developed that will maintain this condition. A reversed **Carnot** engine will removed heat from a low temperature reservoir and deliver this energy, plus work necessary to transfer the heat, to high temperature reservoir. The refrigerated system in this case is the low temperature reservoir.



Reversed Carnot Cycle There are two types of reversed **Carnot** cycle ; the first is:
1-Refrigeration **Carnot** cycle: in which the heat is absorbed from the cold reservoir and rejected to the hot reservoir.



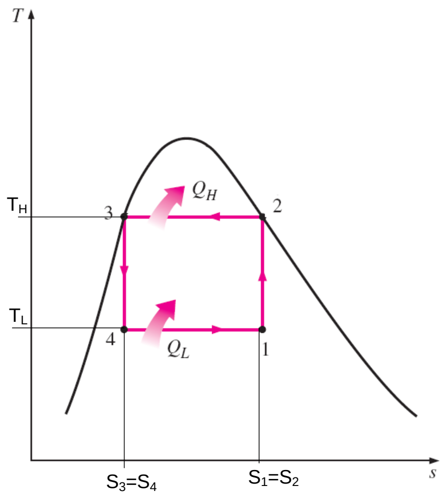
Processes of Refrigeration Carnot cycle

1-2 heat input to the compressor to compress dry saturated vapour, rising its pressure and temperature to dry or superheated condition.

2-3 heat rejected from the vapour to the ambient, changing the dry, or superheated vapour to saturated liquid.

3-4 throttling the saturated liquid (expansion) changing it to wet vapour.

1-4 heat absorbed from cold reservoir, changing the wet vapour of low quality to wet steam of high quality.



Work input (process 1-2)=Q_H -Q_L
Heat rejected (process 2-3) $Q_H= T_H(S_2-S_3)$
Heat absorbed (process2-1) $Q_L= T_L(S_1-S_4)$
Since S3=S4 & S2=S1 $Q_L = T_L(S_2-S$

Coefficient Of Performance (COP)

The performance ratio of refrigeration system is not the efficiency, but rather the **C**oefficient **O**f **P**erformance, and define as the refrigeration effect (heat absorbed) divided by the net work done on the cycle(work input):

$$COP=\frac{QL}{W}=\frac{T_L(S_2-S_3)}{T_H(S_2-S_3)-T_L(S_2-S_3)}=\frac{T_L}{T_H-T_L}$$

$$COP=\frac{T_L}{T_H-T_L}$$

It is more suitable to change the names of the processes of the reversed Carnot cycle to:

Heat absorbed	to	**Refrigeration effect**	Q_L	=	$T_L(S_2\text{-}S_3)$
Heat rejected	to	**Heat rejected from the condenser**	Q_H	=	$T_H(S_2\text{-}S_3)$
Work input	to	**Work input to compressor**	**W**	=	$Q_H\text{-}Q_L$

Example 1

A refrigerator has working temperature in the evaporator and condenser of -30°C and 32°C respectively, what is the maximum COP possible?, if the actual COP of 0.75 of the maximum COP, calculate the refrigeration effect in kW per kW of power input.

$$COP=\frac{TL}{TH-TL}=\frac{-30+273}{32-(-30)}=3.91$$

$$actual\ COP_r=0.75\times 3.91=2.939$$

$$COP_r=\frac{QL}{W}$$

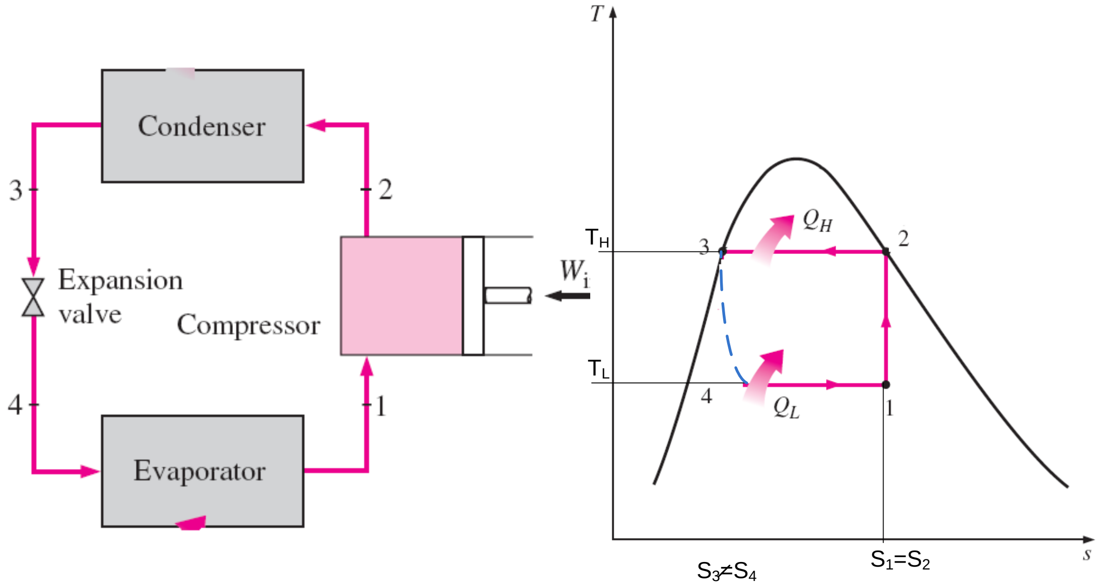
$$2.939=\frac{QL}{1}$$

$$QL=2.939\ kW_{of\ refrigeratin}/kW_{of\ work\ input}$$

Modifications to the ideal refrigeration cycle

a- Replacement of the expansion engine by a throttle valve.

The plant is simplified by placing the expansion engine by a simple expansion valve. The process is highly irreversible so that the whole cycle becomes irreversible. The process is representing by the dotted line 3-4 on the figure. The refrigeration effect $\boldsymbol{Q_L=T_1(S_1-S_2)}$ is reduced by using the expansion valve.



b- Condition at the compressor inlet

To make complete use of the latent heat of the refrigerant in the evaporator it is desirable to continue the process until the vapour is dry saturated.

In practical unit this process is extended to give the vapour a define amount of superheat as it leaves the evaporator. This really undesirable, since the work to be done by the compressor is increased. It is a practical necessity to allow the refrigerant to become superheated in this way in order to prevent the carry-over of liquid refrigerant into the compressor, where it interferes with the lubrication.

c- under-cooling of the condensed vapour

The condensed vapour can be cooled at constant pressure to a temperature below that of the saturation temperature corresponding to the condenser pressure. The effect of sub-cooling can increase the refrigeration effect.

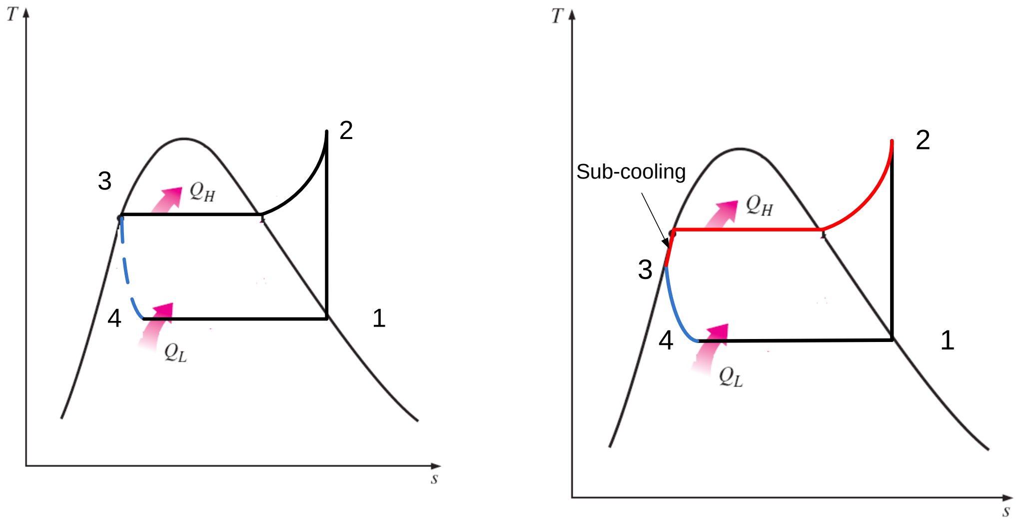


figure b Evaporation process   Figure c sub-cooling process

Ideal cycle

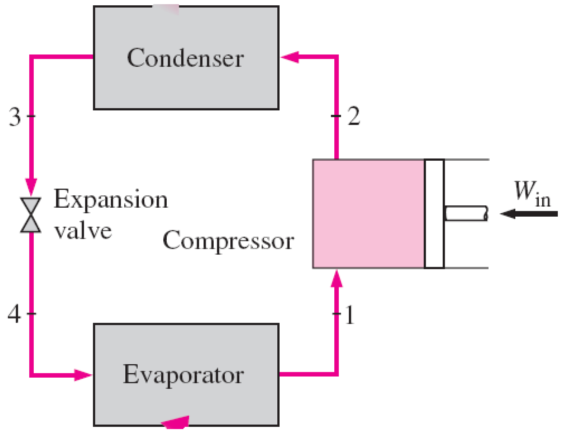
The figure bellow shows the ideal cycle schematically on a p-h diagram. We call this cycle is ideal cycle since pressure and temperature drops are ignored, superheating in the evaporator and sub-cooling in the condenser that are present in real equipment are ignored, and compressor in fluencies are omitted

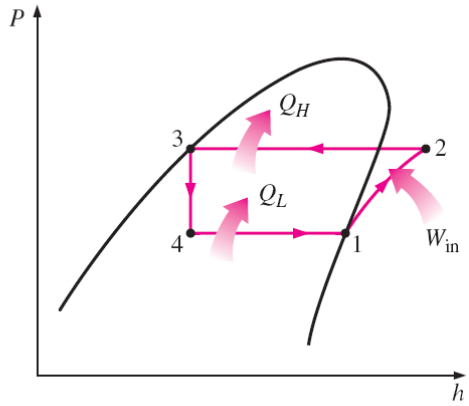
<u>Concept Development</u>

All vapour compression refrigeration system are designed and built around these basic thermodynamics principles.

1. fluid absorb heat while changing from a liquid phase to vapour phase and give up heat in changing from vapour phase to a liquid phase.
2. the temperature at which a change of phase occurs is constant during the change, but this temperature will vary with the pressure. At one fixed pressure vaporization take place only at fixed corresponding temperature. Whoever, the temperature of vaporization at a practical pressure, are different for different fluids.
3. heat will flow from body at higher temperature to a body at lower temperature.
4. in selection metallic parts of cooling and condensation units metal are selected which have a high heat conductivity.

5. heat energy and other forms of energy are mutually convertible with directional relationship imposed by the second law of thermodynamics.





The cycle consists of number of flow process and can be analyzed by the application of steady flow energy equation:

1- *Compression process (1-2):*

Process 1-2 take place in the compressor as the pressure of the vapour is increased by compression from the vaporizing pressure to the condensation pressure. For simple saturated cycle the compression is assumed isentropic. Hence the steady flow energy equation is

$$gz_1 + \frac{C_1^2}{2} + h_1 + Q = gz_2 + \frac{C_2^2}{2} + h_2 + W_{comp}$$

$$z_1 \approx z_2 \qquad Q = 0 \; isentropic \quad C_2 \approx C_1$$

$$W_{comp} = h_2 - h_1$$

2- *Condensation process (2-3)*

Usually the process $2-\bar{2}$ and $\bar{2}-3$ take place in the condenser as the hot gas discharged from the compressor is cooled to the condensation temperature. During process $2-\bar{2}-3$ the pressure of the vapour remains constant. At point $\bar{2}$ the refrigerant is saturated vapour at condensation temperature and pressure. Process $2-\bar{2}$ takes place at constant pressure, while process $\bar{2}-3$ takes place at constant pressure and temperature. The condensation is assumed to be at constant pressure

$$gz_2 + \frac{C_2^2}{2} + h_2 + Q_{cond} = gz_3 + \frac{C_3^2}{2} + h_3 + W$$

$$z_3 \approx z_2 \qquad W = 0 \; C_3 \approx C_1$$

$$Q_{cond} = h_2 - h_3$$

3- *Expansion process:*

The process described by initial and final state points 3-4 occur in the throttling valve when the pressure of the liquid is reduced from condensation pressure to evaporation pressure as the liquid passes through the throttle expansion valve. This process is throttling type adiabatic expansion in which enthalpy of working substance remains the same.

The expansion is assumed to be adiabatic process i.e $h_3=h_4$

4- Evaporation process:

The process 4-1 is the evaporation process at evaporator pressure and temperature. In this process both pressure and temperature are remain constant for simple saturated ideal cycle.

The evaporation process is assumed to be at constant pressure

$$gz_4 + \frac{C_4^2}{2} + h_4 + Q_{evap} = gz_1 + \frac{C_1^2}{2} + h_1 + W$$

$$z_1 \approx z_4 \qquad W = 0 \; C_4 \approx C_1$$

$$Q_{evap} = h_4 - h_1$$

Then the equations related the case above can be writing as follow:

Work input to compressor **$w_c = h_2 - h_1$**

Refrigeration effect **(Q_{evap})= h_1-h_4**

Heat rejection from condenser **$Q_{cond.}$= h_2-h_3**

Coefficient of Performance **(COP)=** $\frac{Q_{evap}}{W_{comp}} = \frac{h_1 - h_4}{h_2 - h_1}$

h1= hg at condenser pressure or temperature

h2=h ***superheated*** at compressor pressure and temperature

h3= h_f at condenser pressure and temperature

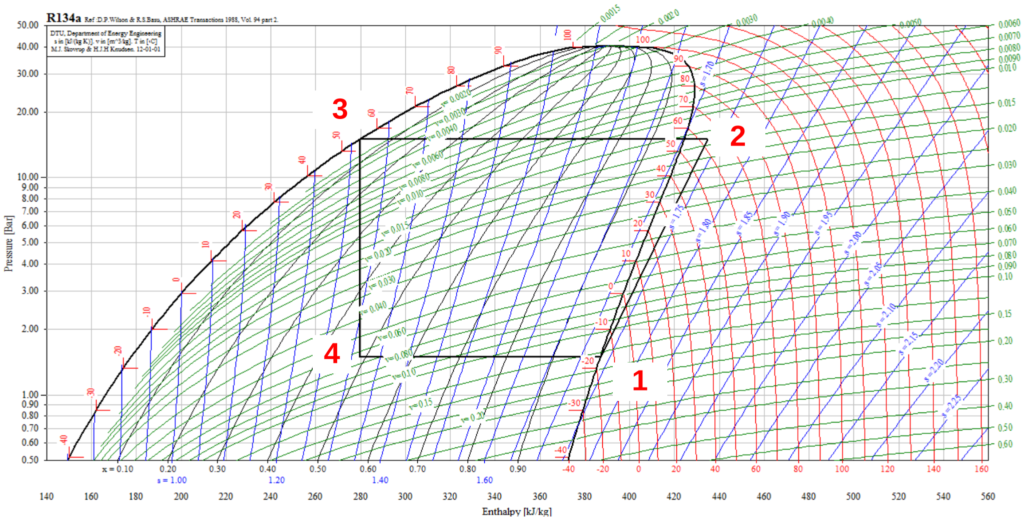
h3=h4

to find the dryness fraction at evaporator enter can be found as follow:

$h_3 = h_4 = hf + x_4.h_{fg}$ at evaporator pressure or temperature

Example 2

A refrigerator cycle uses refrigerant R-134a and operates between a low-side pressure of 0.15Mpa and high side of 1Mpa. The refrigerant mass flow rate is 0.05kg/s. find the cooling effect, work input, and COP of this machine.



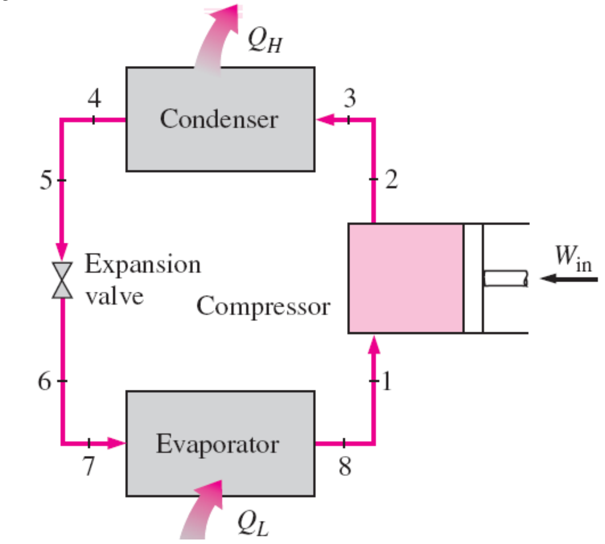
from the p-h diagram we can find the enthalpies at each point as follow:

Point	P Mpa	T ^oC	h kJ/kg	S kJ/kg K
1	0.15	-17.2	387	1.73
2	1	63	434	1.73
3	1	55	279	--
4	0.15	-17	279	--

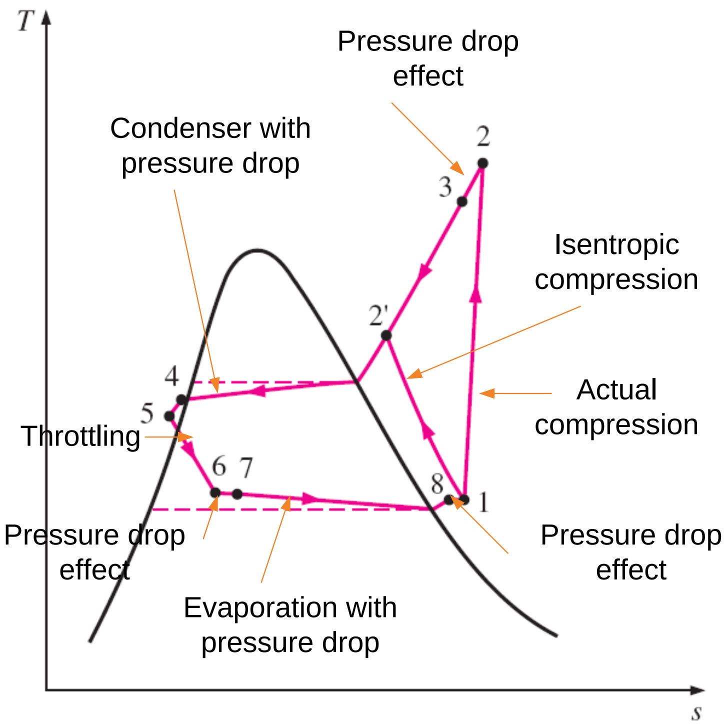
Work input to compressor $w_c = h_2 - h_1 = 434 - 387 = 47$ kJ/kg
Power input to the compressor=m(h2-h1)=0.05(434-387)=2.35kW
Refrigeration effect (Q_{evap})= $h_1 - h_4 = 387 - 279 = 108$kJ/kg
Refrigeration effect in kW=m(h1-h4)=0.05(387-279)=5.4kW

(COP)= $\dfrac{Q_{evap}}{W_{comp}} = \dfrac{108}{47} = 2.29$

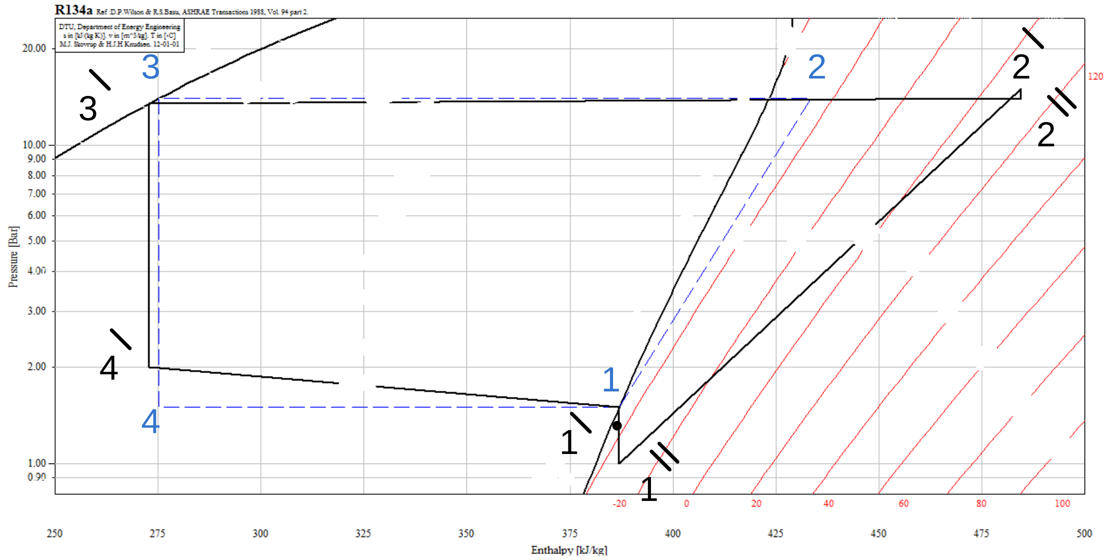
Real Cycle Analysis



Real cycle equipment with typical R22 operating temperatures and pressures



T-s diagram of actual VC cycle



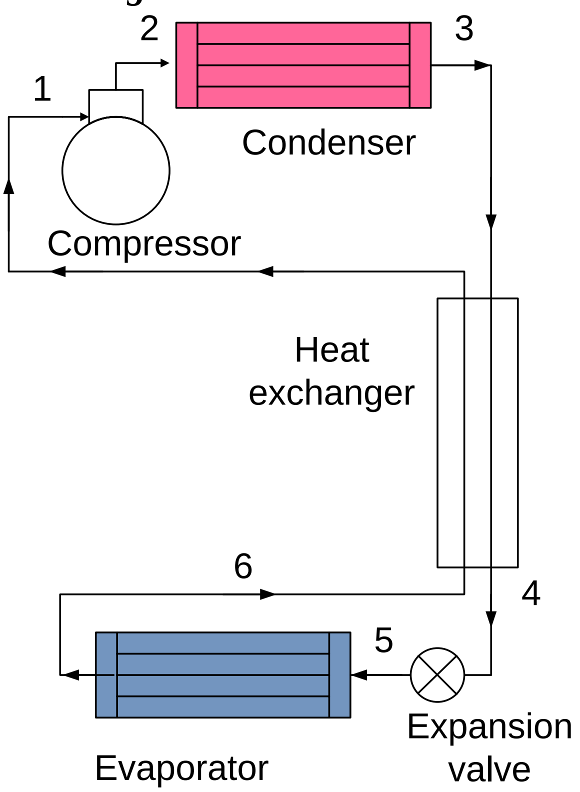
The effect of pressure loss in real cycle:

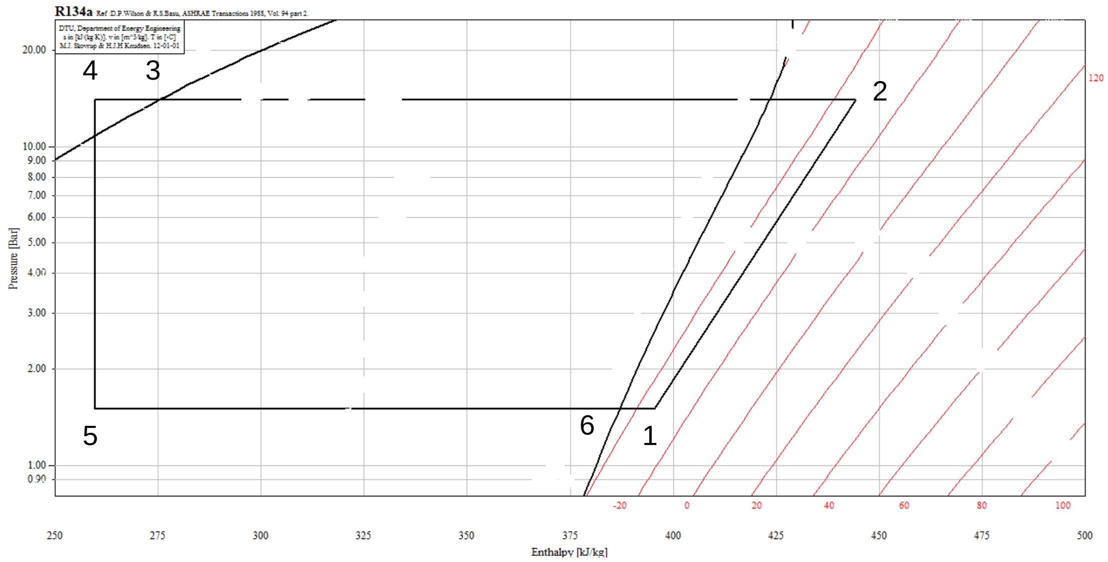
During the flow of the refrigerant through the piping, evaporator, condenser receiver, valves and pipe lines, there is pressure drop due to internal fluid friction. The figure shows a p-h representation of an actual cycle with pressure drops occurring in various components.

1. The line $4'-1$ Vaporization process in the evaporator and the refrigerant during its flow drop in pressure.
2. $1 \rightarrow \bar{1}$ Pressure drop in the suction vapour through the suction line from evaporator to compressor. This loss in pressure increases the volume of vapour compressed per ton and horse power per ton.
3. $\bar{1} \rightarrow \bar{\bar{1}}$ Pressure drop of refrigerant vapour flow through the suction valve of compressor. The effect of this pressure drop is similar to that at suction line.
4. $\bar{\bar{1}} \rightarrow 2$ Compression process for the cycle with a pressure drop. Thus the vapour has to be compressed to a pressure much above that required for simple saturated cycle.
5. $\bar{2} \rightarrow \bar{\bar{2}}$ Pressure drop in the discharge valves of the compressor.
6. $\bar{\bar{2}} \rightarrow \bar{3}$ Condensation process and the pressure drop in the condenser piping.
7. $\bar{3} \rightarrow \bar{4}$ Expansion process by expansion device.

the dotted lines represent the ideal vapour compression cycle

The effect of heat exchangers:



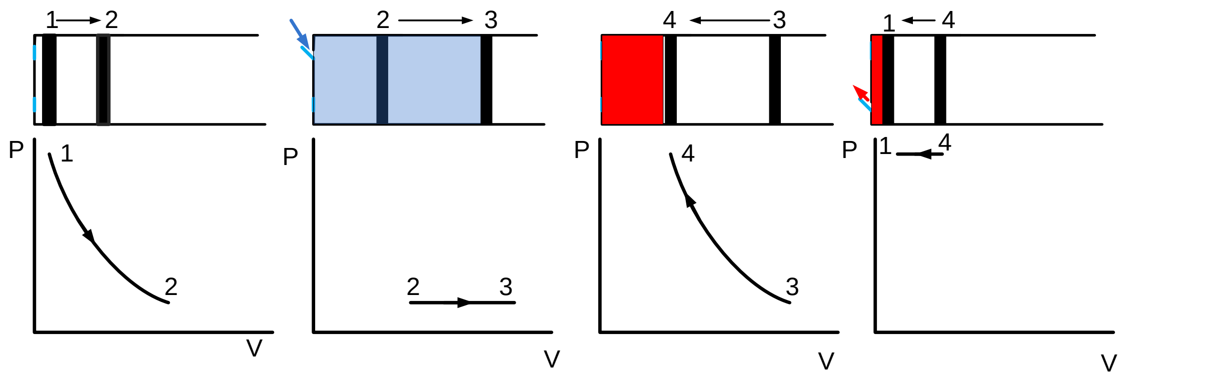


Some refrigeration cycles use a liquid to suction heat exchanger , which sub-cool the liquid from condenser with suction vapour coming from the evaporator. Saturated liquid at point 3 coming from condenser is cooled to point 4 by means of vapour at point 6 being heated to point 1. from the heat balance $: h_3 - h_4 = h_1 - h_6$ the refrigeration effect is either $h_6 - h_5$ *or* $h_1 - h_5$. The system use heat exchanger may seem to have obvious advantages because the increased refrigeration effect. Both capacity and COP may seemed to improved. This is not necessarily true, however. Even thought the refrigeration effect is increased, the compression is pushed farther out into the superheat region, where the work of compression in kJ/kg is greater than it is close the saturated vapour. The heat exchanger is important because of two reasons:

1. The vapour entering the compressor must be superheated to ensure that no liquid enters the compressor.
2. to sub cool the liquid from condenser to prevent bubbles of vapour from impeding the flow of refrigerant though the expansion valve.

Compressor capacity

The capacity of the compressor is ***the compressor swept volume per unit time***, while, the volumetric efficiency (η_{vol}) of the compressor is ***the actual of the sucked gas to the*** theoretical piston displacement. And clearance volumetric efficiency is the ***volumetric efficiency of the compressor when the clearance volume of the compressor is included.***



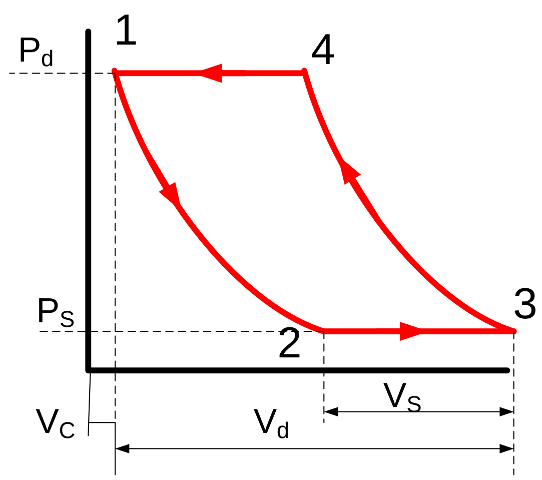
Suction of gas	***Counties suction of gas***	***Compression of gas***	***Compressed of gas***
$PV^{\gamma}=C$	P=C	$PV^{\gamma}=C$	P=C

Vc: Clearance volume=V_1
Vd: Discharge volume (V_3-V_1)
Vs: Sucked volume theoretical (V_3-V_2)

C: Clearance index $=\dfrac{Vc}{Vd}=\dfrac{V_1}{V_3-V_1}$

$$\eta_{cv}=\frac{Vs}{Vd}=\frac{V_3-V_2}{V_3-V_1}$$

V_2 Cannot be measured



$$P_1V_1^{\gamma} = P_2V_2^{\gamma}$$

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} = V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

$$\eta_{cv} = \frac{V_3 - V_2}{V_3 - V_1} = \frac{V_3 - V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}}{V_3 - V_1}$$

$$\eta_{cv} = \frac{V_3 - V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}} - V_1 + V_1}{V_3 - V_1}$$

$$\eta_{cv} = \frac{V_3 - V_1 - V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}} + V_1}{V_3 - V_1}$$

$$\eta_{cv} = \frac{V_3 - V_1}{V_3 - V_1} + \frac{V_1}{V_3 - V_1} - \frac{V_1}{V_3 - V_1}\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

$$\eta_{cv} = 1 + C - C\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

$$where \quad C = \frac{V_1}{V_3 - V_1}$$

$$\sin ce \quad \frac{Vs}{Vd} = \left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

then

$$\eta_{cv} = 1 + C - C\frac{Vs}{Vd}$$

$$V_2 = V_1\frac{P_d}{P_s}$$

$$\eta_{cv} = 1 + C - C\frac{P_d}{P_s}$$

If the compression process is done at constant temperature then the clearance volume can be written as:

$$P_1V_1^{\gamma} = P_2V_2^{\gamma}$$

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} = V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

$$\eta_{cv} = \frac{V_3 - V_2}{V_3 - V_1} = \frac{V_3 - V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}}{V_3 - V_1}$$

$$\eta_{cv} = \frac{V_3 - V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}} - V_1 + V_1}{V_3 - V_1}$$

$$\eta_{cv} = \frac{V_3 - V_1 - V_1\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}} + V_1}{V_3 - V_1}$$

$$\eta_{cv} = \frac{V_3 - V_1}{V_3 - V_1} + \frac{V_1}{V_3 - V_1} - \frac{V_1}{V_3 - V_1}\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

$$\eta_{cv} = 1 + C - C\left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

$$where \quad C = \frac{V_1}{V_3 - V_1}$$

$$\sin ce \quad \frac{Vs}{Vd} = \left(\frac{P_d}{P_s}\right)^{\frac{1}{\gamma}}$$

then

$$\eta_{cv} = 1 + C - C\frac{Vs}{Vd}$$

$$V_2 = V_1\frac{P_d}{P_s}$$

$$\eta_{cv} = 1 + C - C\frac{P_d}{P_s}$$

Piston displacement (V_D)of compressor can be calculated as :

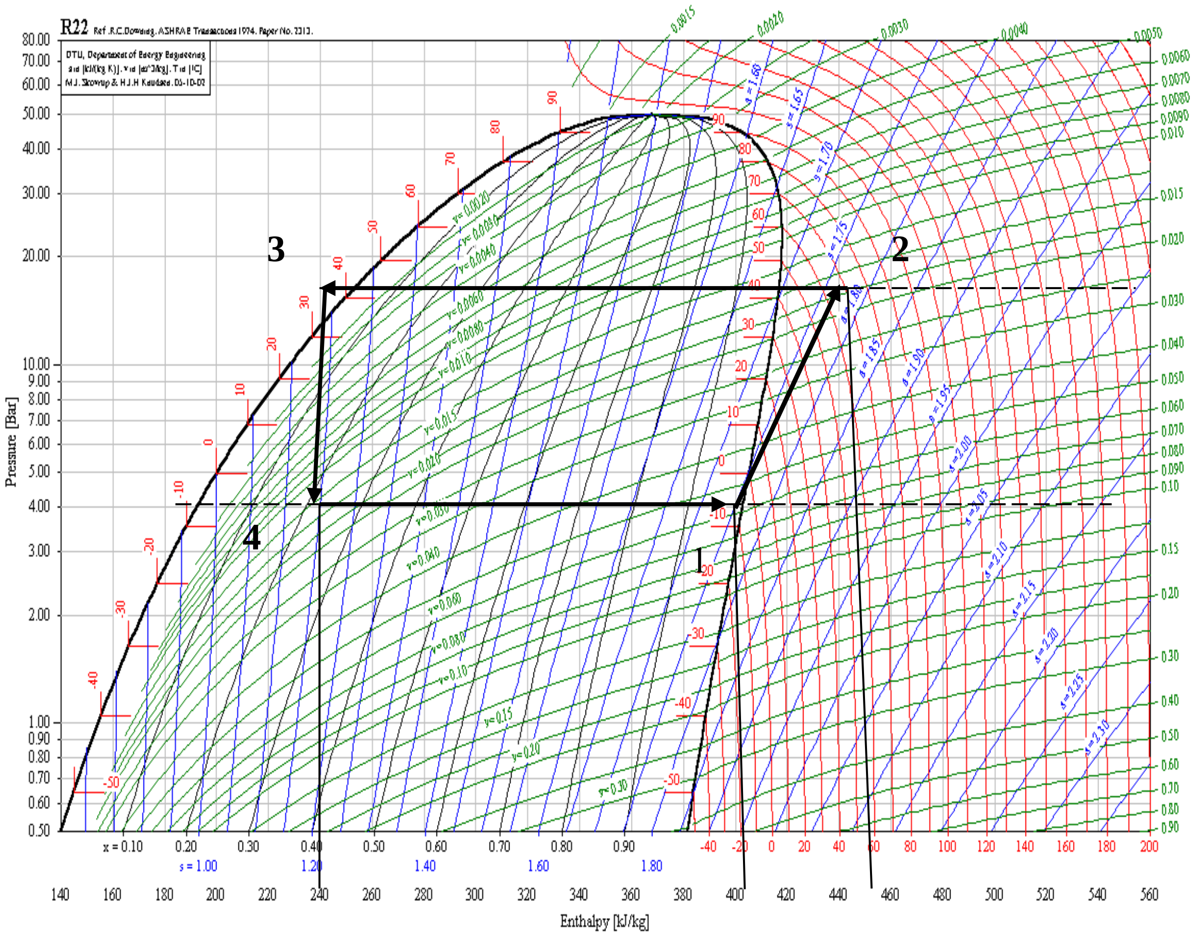
$$V_D = No.\frac{\pi.d^2}{4}.l.\frac{rpm}{60}$$

V_D	Piston displacement	m^3/s
No.	Number of cylinder in the compressor	-----
d	Piston diameter	m
l	Piston stroke	m
rpm	Revelation per min.	1/min.

Examples

Example 2:

A standard vapour compression cycle developed 50 kW of refrigeration using R-22 operates with condensing temperature of 35°C and evaporation temperature of (-10 °C). Calculate a-the refrigeration effect in kJ/kg b- the mass flow rate of refrigerant c-the power required for compression d-COP e- power per kW of refrigeration f-discharge temperature.



h1=hg=401 kJ/kg $h2=h_{super\ heated}$=435kJ/kg   h3=h4=hf=243kJ/kg

a- Refrigeration effect= Qevap=(h1-h4)=401-243=158kJ/kg

b- mass flow rate of refrigerant:

Cycle Capacity $= \dot{m}(h1 - h4)$

$$\dot{m} = \frac{50}{158} = 0.316 kg/s$$

c- power required for compression P= $\dot{m}(h2-h1)=0.316(435-401)=10.744kW$

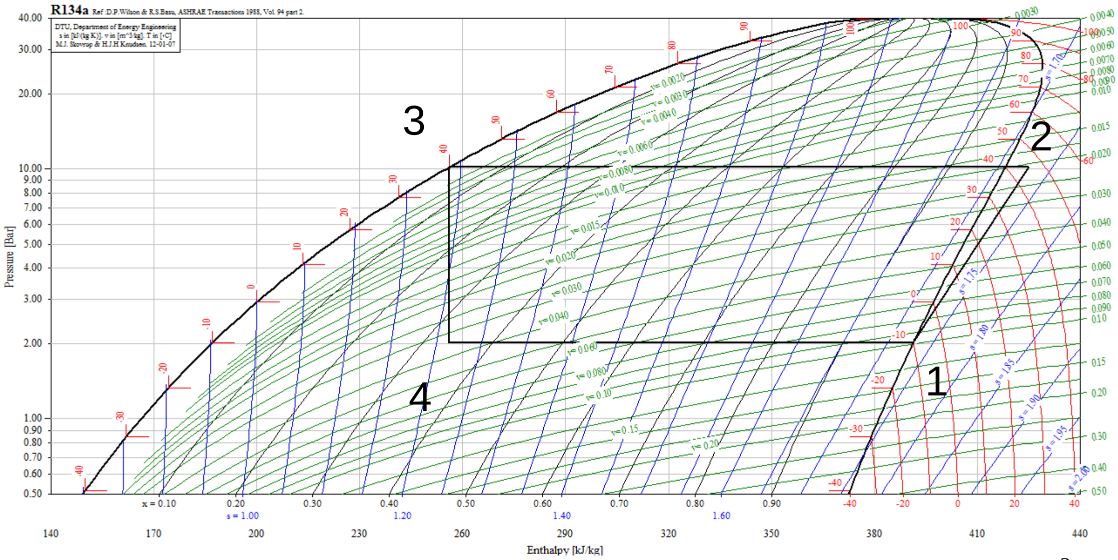
d- $COP=\dfrac{Qevap}{Wcomp}=\dfrac{h1-h4}{h2-h1}=\dfrac{401-234}{435-401}=4.911$

e- power per kW of refrigeration= $\dfrac{1}{COP}=\dfrac{1}{4.911}0.203kW/kWrefrig.$

f- discharge temperature. $=T_2=60^oC$

Example3

A two cylinder Freon 134a compressor has a bore and stroke of 5.65 cm and 5 cm respectively. It is speed is 1450 rpm and 100% volumetric efficiency. If the liquid reached the expansion valve at 40 ^oC saturated. Find the mass flow rate of the refrigerant and refrigeration capacity when the suction temperature is (-10 ^oC). How will be the result if the clearance index is 4%.



h1=hg=391 kJ/kg   h2=424 kJ/kg   h3=hf=256 kJ/kg v_1=0.098 m^3/kg,
v_2=0.02 m^3/kg

$$Displacement\ volume=\eta_{cv}=1+C-C\frac{Vs}{Vd}=2\times\frac{\pi(5.65\times10^{-2})}{4}\times5\times10^{-2}\times\frac{1450}{60}\times1=5.95\times10^{-3}m^3/s$$

$$\dot{m}=\frac{V_D}{v_1}=\frac{5.95\times10^{-3}}{0.098}=0.06\quad kg/s$$

$$capacity=\dot{m}(h1-h4)=0.06\times(391-256)=8.1\ kW$$

b

$$\eta_{cv}=1+C-C\frac{vs}{vd}=1+0.04-0.04\frac{0.098}{0.02}=0.844$$

$$VD_2=VD_1=5.95\times10^{-3}\times0.844=5.338\times10^{-3}=5.02x10^{-3}m^3/s$$

$$mf=\dot{m}=\frac{V_D}{v_1}=\frac{5.02\times10^{-3}}{0.098}=0.0512\ kg/s$$

$$capacity=\dot{m}(h1-h4)=0.0512\times(391-256)=6.91\ kW$$

Sheet No. Three

1. the temperature in evaporator coil is -6 ^oC and that in the condenser coil is 22 ^oC. assuming that the machine operates on the reversed Carnot cycle. Calculate the COP the refrigeration effect per kW of input work, and the heat rejected to the condenser. (9.54;9.54kW;10.4kW)
2. a Carnot refrigeration cycle absorbs heat at (-12 ^oC) and rejects it at 40 ^oC a- calculate the COP of this cycle d- If the cycle is absorbing 15 kW at (-12 ^oC) temperature, how much power is required. C- if the Carnot heat pump operates between the same temperature, what is the performance factor d- what is the rate of heat rejection at 40 ^oC if the heat pump absorbs 15 kW at the (-12 ^oC) temperature. (18kW)
3. in a standard vapour compression cycle using R-22 the evaporation temperature is (-5 ^oC) and the condensing temperature is 30 ^oC, calculate a-the work of compression b- the refrigeration effect c-the heat rejected in the condenser d- COP (6.47)
4. a refrigeration system using R-22 is to have a refrigerating capacity of 80 kW. The cycle is standard vapour compression cycle in which the evaporation temperature is (-8 ^oC) and the condensing temperature 42 ^oC a- determine the volume flow of refrigerant measured in cubic meter per second at the inlet to the compressor. B- calculate the power required by the compressor. C-at the entrance to the evaporator what is the fraction of vapour in the mixture expressed both on mass basis and volume basis(0.292,0.971)
5. a refrigerant R-22 vapour compression system includes a liquid to suction heat exchanger in the system. The heat exchanger warms saturated vapour coming from evaporator from (-10 ^oC) to 5 ^oC with liquid which comes from the condenser at 30 ^oC. the compression are isentropic in both cases below. A – calculate the COP of the system without the heat exchanger but with condensing temperature at 30 ^oC and evaporator temperature of (-10 ^oC)(5.46) b- calculate the COP with the heat exchanger (5.37) c- if the compressor is capable of pumping 12 lit/s measured at the compressor suction, what is the refrigeration capacity of the system without heat exchanger and with heat exchanger(30.3kW, 29.9kW)
6. calculate the displacement of a compressor having 176kW capacity if the refrigeration effect is 1097kJ/kg and the volume of the suction gas is 0.2675 m^3/kg. Assuming a volumetric efficiency of 75%, what cylinder size will be need if the speed is to be 25rps and there are 6 cylinders with equal bore and stroke (0.0429m^3/s and 78.6 mm)
7. a four cylinder 75 mm bore and 75 mm stroke compressor run at 25 rps and has a volumetric efficiency of 75%. If the volume of the suction gas is 0.248 m3/s and the machine has operating efficiency of 75%, what power will required on simple saturation cycle when difference between enthalpies of the suction and discharge gases is 150kJ/kg. If the refrigeration effect is 1087 kJ/kg what is the output in kW of refrigeration. State the COP .(20Kw, 108.7kW,COP=7.25)

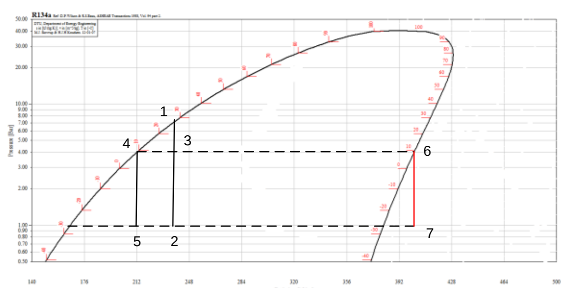
8. *an ammonia vapour compressor refrigerator has a single acting stage, single acting reciprocating compressor which has a bore of 127 mm, a stroke of 152 mm and speed 240 rpm. The pressure in the evaporator is 1.6 bar and that in condenser is 13.9 bar. The volumetric efficiency of the compressor is 80% and mechanical efficiency is 90%. The vapour is dry saturated on leaving evaporator and liquid leaves the condenser at 32 ^oC. calculate the mass flow rate of the refrigerant and the power ideally required to drive the compressor(0.502 kg/min, 9.04 kW, 2.73 kw)*
9. *the air conditioning in a car uses R-134 a when the power input to the compressor is 1.5kW, bringing the R-134a from 202 kPa to 1200 kPa by compressor. The cold space is heat exchanger that cool atmospheric air from outside down ti 10 ^oC and blow it into the car. What is the mass flow rate of the refrigerant and what is the low temperature heat transfer. How much is the mass flow rate of air at 10 ^oC.*
10. *a small heat pump unit is used to heat water. Assume that the unit uses R-22. the evaporator temperature is 15 ^oC and the condenser temperature is 40 ^oC. if the amount of hot water need is 0.1 kg/s. determine the amount of energy saved by using the heat pump instead of direct heating the water from 15 to 40^oC.*
11. *a heat pump using R-717 as a refrigerant operates between saturated temperature of 6 ^oC and 38 ^oC. the refrigerant is compressed isentropically from dry saturated and there is 6 K of under cooling in the condenser. Calculate P.F and the mass flow rate of refrigerant and the heat available per kW input. (8.8, 25.065 kg/hr, 8.8kW)*

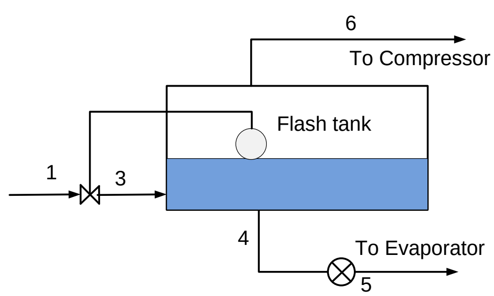
Compound Vapour Compression System

In vapour compression cycle the major operating cost is the energy input to the system in the form of mechanical work thus with the same refrigeration effect produce per kg of refrigeration circulated, the attempts can be made to save the mechanical work input. But this should not involve too heavy an increase in operating expenses as well as he initial plant cost. The staging of refrigerant compressors with inter-cooling reducing in compression work done per kg of refrigeration circulated. But it has been found that the staging with inter cooling may be economical only when very high pressure ratio is involved and when very low evaporator temperature are desired or when high condenser temperature. Compound compression system becomes economical in large plants.

1-1 Removing Of Flash Gas

When saturated liquid expands through an expansion valve the percentage of vapour present increases. The expansion process shown in p-h diagram takes place from 1-2. The end point of expansion 2, could be achieved by interrupting the expansion at 3 and separating the liquid and vapour phases which are 4 and 6 respectively. The expansion could then continue by expanding the liquid a 4 and vapour at 6 o final pressure, giving 5 and 7 respectively. The combination of refrigerant a states 5 and 7 gives point 2. The expansion from 6 to 7 is wasteful. The refrigerant at 7 can produce no effect and also will be required to compress the vapour back to pressure it had at 6.

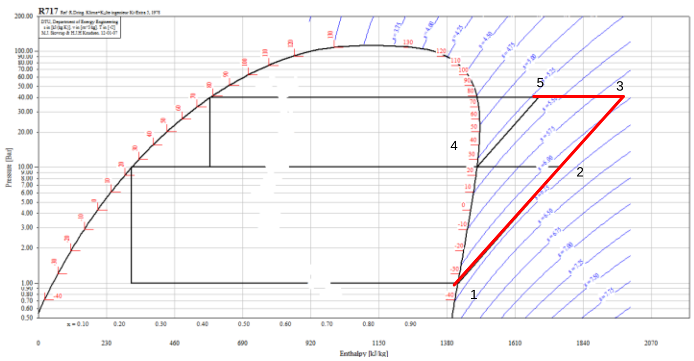


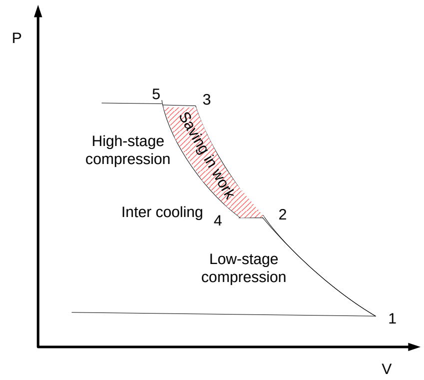


The equipment to achieve the separation is called ***Flash Tank*** as shown in the figure.

1-2-Inter cooling

Inter cooling between two stage compression reduces the work of compression per kg of vapour. The saving work from 2 to on the p-v diagram is shown in the figure below.





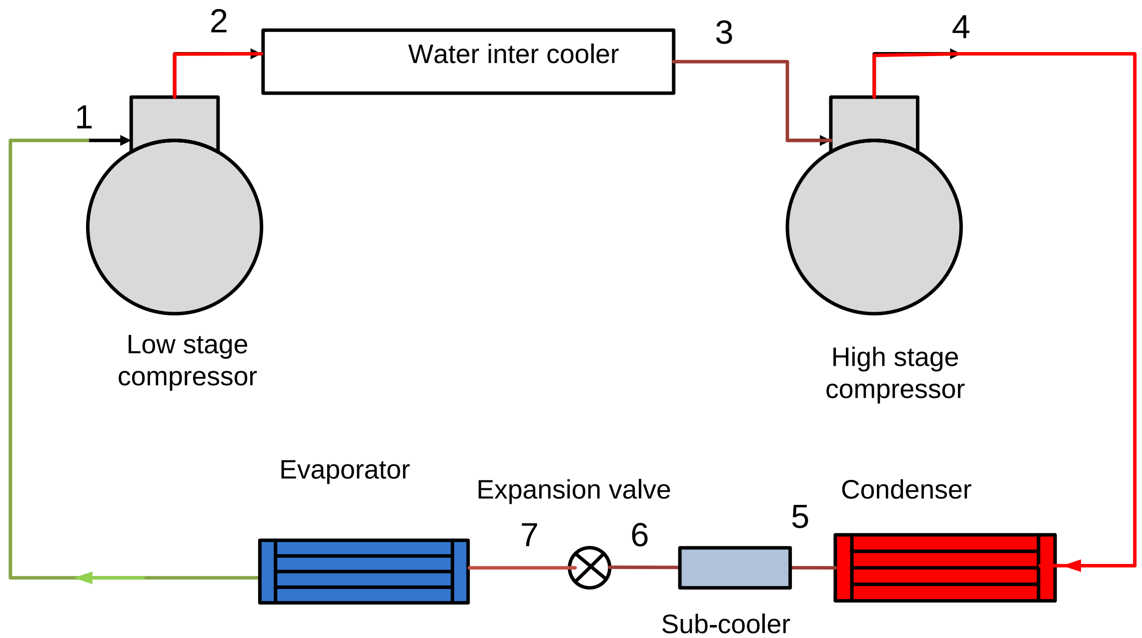
1-2 Low stage compression
2-4 Inter cooler
4-5 High stage compression

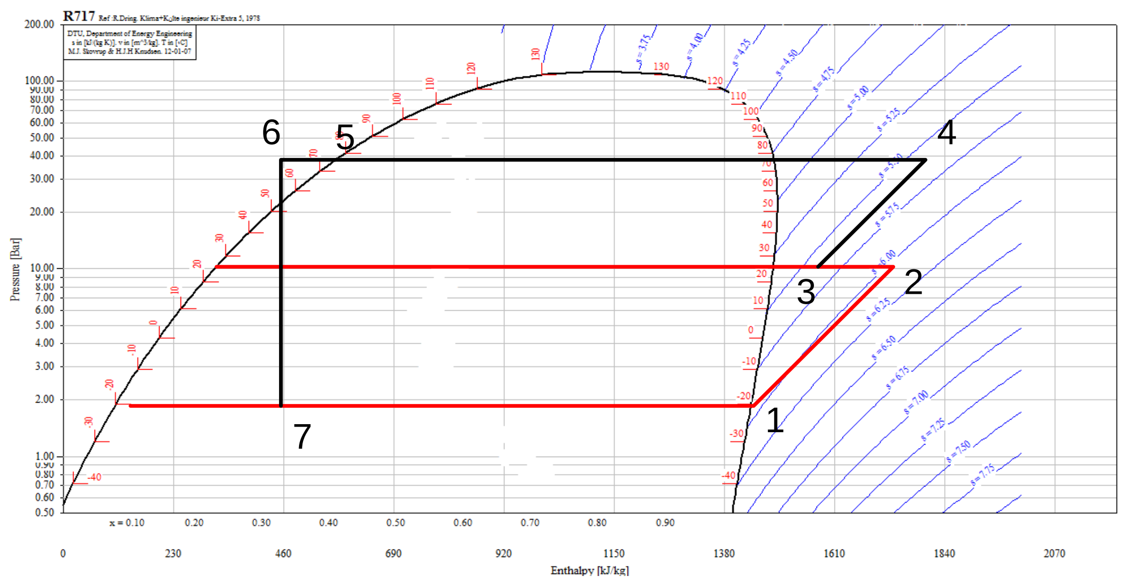
1-3 Single stage compression
23542 Saving in work done

Types of intercoolers
There are three types of intercoolers as mentioned below.
1-water Inter-cooler
2-Liquid Refrigerant Inter-cooler
3-Flash Gas Inter-cooler

1-3 Compound compression with water inter-cooler

the low pressure compressor draws the refrigerant vapour from the evaporator and compresses it to some intermediate pressure and discharge it to intercooler. The gas in the intercooler is cooled by water. The gas is then led to high pressure compressor and compressed to condenser pressure. Heat is rejected in the inter cooler at constant pressure. The vapour compression system with water intercooler is shown in the figure.

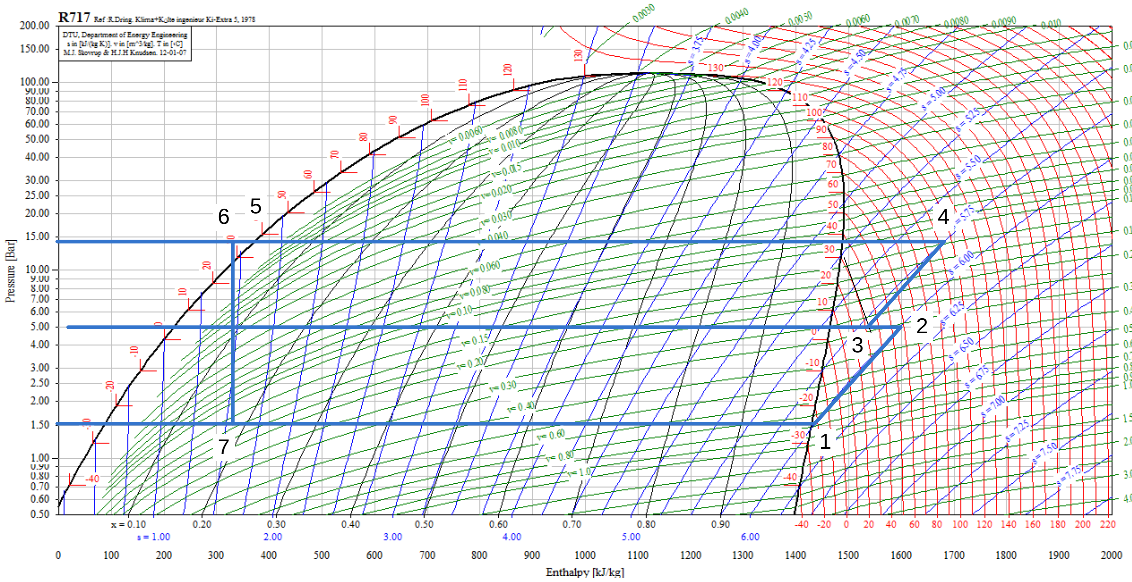




1-2 Compression in low stage compressor	5-6 Sub-cooling in sub-cooler
2-3 Inter-cooling in water inter cooler	6-7 Expansion in the expansion valve
3-4 Compression in High stage compressor	7-1 Evaporation in the evaporator
4-5 Condensing in the condenser	

Example 1: The following data apply to 100 ton compound ammonia compression system with water inter-cooler, condenser pressure 14 bar, evaporator pressure 1.5 bar and inter-cooler pressure 5 bar, the ammonia is cooled to 32^{o}C in the water inter-cooler and sub-cooled to 30 ^{o}C in the sub-cooler,. The temperature of suction gas is -18 ^{o}C; find

a- mass flow rate of refrigerant in the cycle b- power consumption in each compressor c- heat rejection in water inter-cooler d- displacement volume of in both low and high stage and low stage compressor



h_1	h_2	h_3	h_4	h_6
1450	1600	1540	1710	300

capacity=mr.(h_1-h_7)

$$mr = \frac{100 \times 3.516}{(1450 - 300)} = 0.3057 kg/s$$

Comp. power L.P.= mr.(h_2-h_1)= 0.307x(1600-1450)=45.86 kW

Comp. power H.P.= mr.(h_4-h_3)= 0.307x(1710-1540)=51.97 kW

$Q_{inter\text{-}cooler}$= mr.(h_2-h_3)=0.307x(1600-1540)=18.34 kW

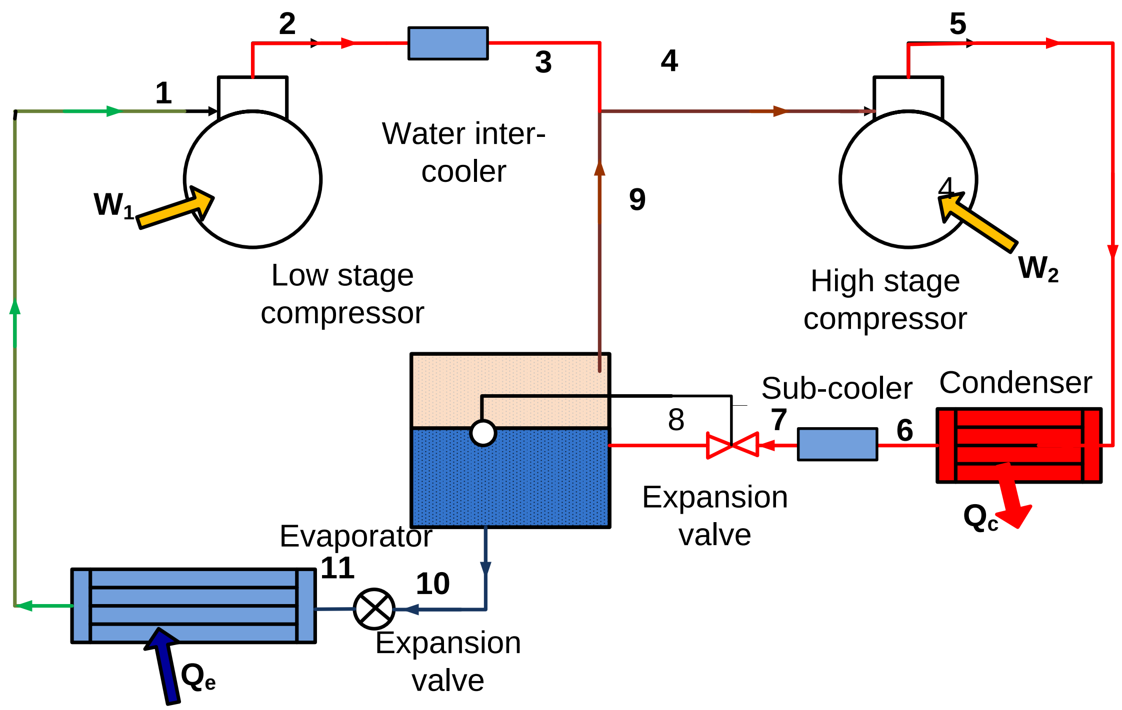
$$vd.L.P = 0.9 \times 0.307 = 0.275 m^3/s$$

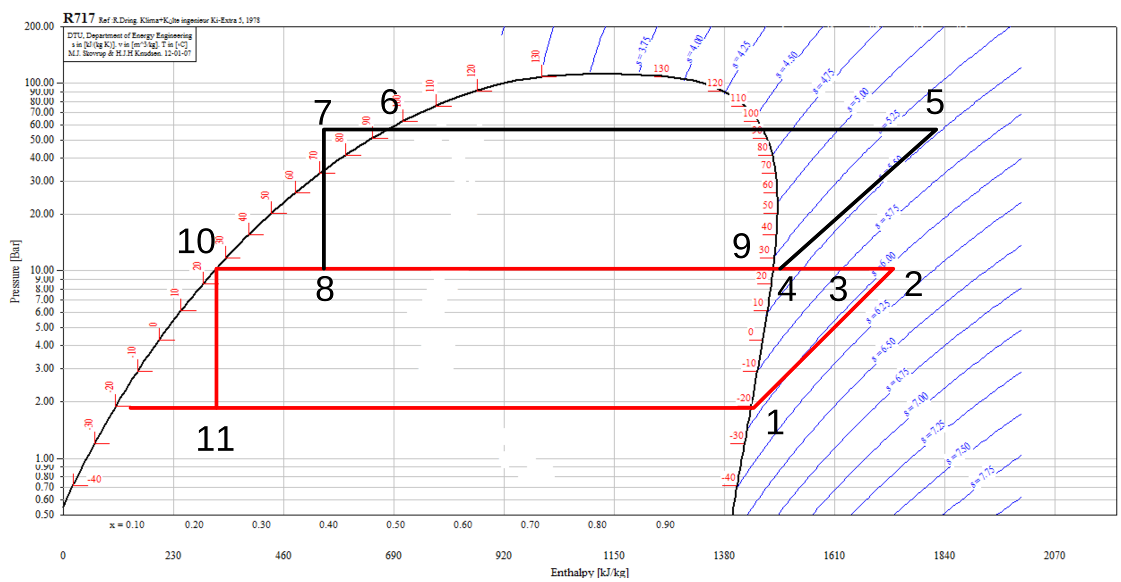
$$vd.H.P = 0.28 \times 0.307 = 0.0856 m^3/s$$

$$COP = \frac{Capacity}{Wcomp1 + Wcomp2} = \frac{100 \times 3.516}{45.86 + 51.97} = 3.593$$

1-4Compound compression with liquid Flash cooler

In this type of inter-cooling, the gas leaving the low stage compression is mixed with the gas formed due to expansion process, and lead them to a second stage to recompression.





Heat balance of flash inter-cooler

$m_8h_8 = m_{10}h_{10} + m_9h_9$

$m_8 = m_{10} + m_9$

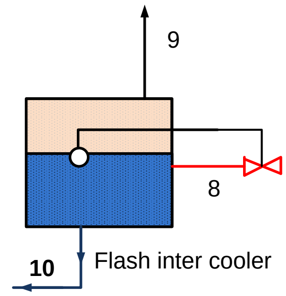
$m_9 = m_8 - m_{10}$

$m_8\,(h_8-h_9) = m_{10}(h_{10} - h_9)$

$m_8h_8 = m_{10}h_{10} + (m_8-m_{10})h_9$

$m_8\,(h_8-h_9) = m_{10}(h_{10} - h_9)$

$$m8 = \frac{m10(h10-h9)}{h8-h9}$$



Heat balance of mixing process

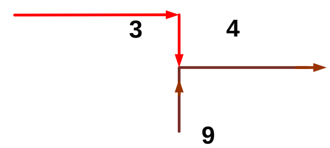
Heat balance

$m_4.h_4 = m_3.h_3 + m_9h_9$

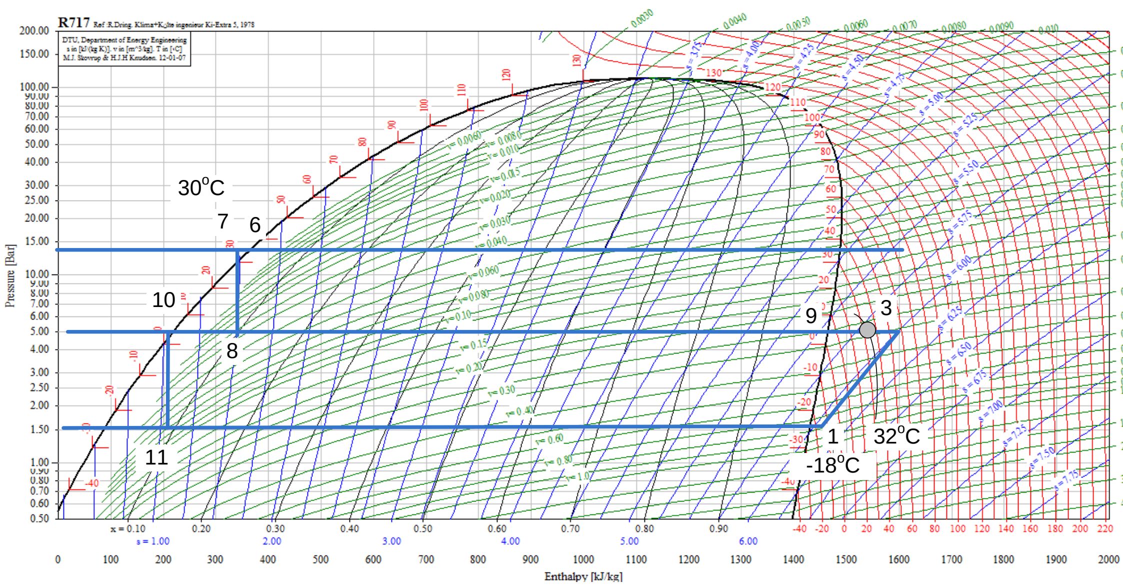
mass balance

$m_4 = m_3 + m_9$

$$h4 = \frac{m3h3 + m9h9}{m3 + m9}$$



Example 2: The following data apply to 100 ton compound ammonia compression system with water inter-cooler and liquid refrigerant inter-cooler, condenser pressure 14 bar, evaporator pressure 1.5 bar and inter-cooler pressure 5 bar, the ammonia is cooled to 32^{o}C in the water inter-cooler and sub-cooled to 30 ^{o}C in the sub-cooler,. The temperature of suction gas is -18 ^{o}C; find a- mass flow rate of refrigerant in the cycle b- power consumption in each compressor c- heat rejection in water inter-cooler d-COP



h_1	h_2	h_3	h_4	h_5	h_6	h_9	h_{10}
1450	1600	1540	?	?	360	1490	200

Capacity $= m.(h_1 - h_{11})$

$$mf = \frac{100 \times 3.516}{(1450 - 200)} = 0.28128$$

$m_8h_8 = m_{10}h_{10} + m_9h_9$

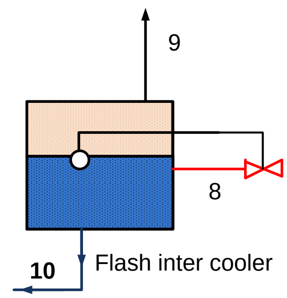
$m_8 = m_{10} + m_9$

$m_9 = m_8 - m_{10}$

$m_8 (h_8\text{-}h_9) = m_{10}(h_{10} - h_9)$

$m_8h_8 = m_{10}h_{10} + (m_8\text{-}m_{10})h_9$

$m_8 (h_8\text{-}h_9) = m_{10}(h_{10} - h_9)$



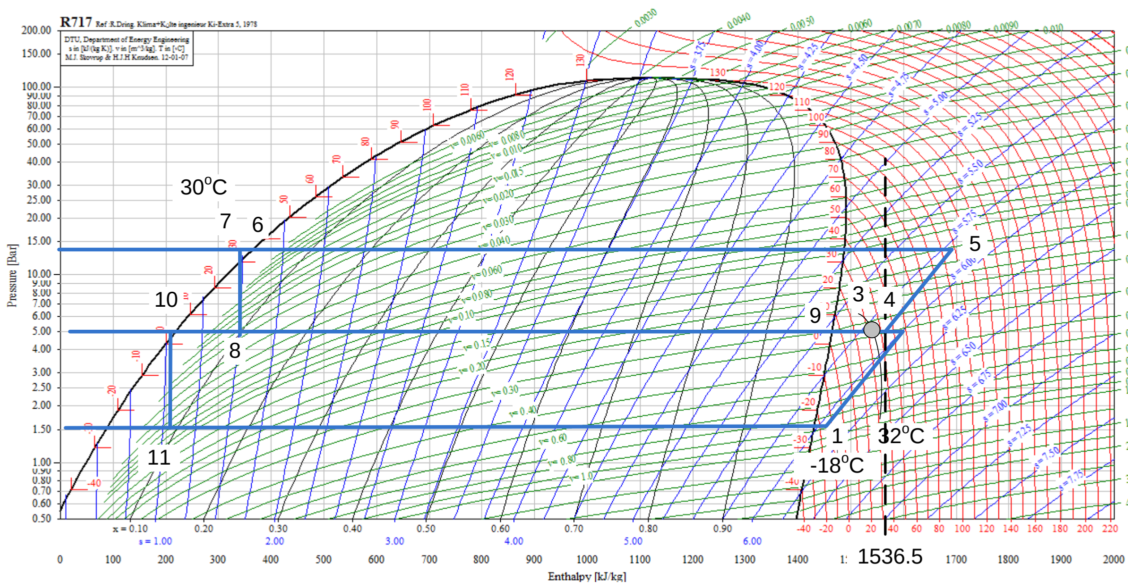
$$m8 = \frac{m10(h10 - h9)}{h8 - h9}$$

$$m8 = \frac{0.281X(200 - 1490)}{360 - 1490} = 0.32kg / s$$

$m_9 = m_8 - m_{10} = 0.32 - 0.281 = 0.039$ kg/s

$$h4 = \frac{m3h3 + m9h9}{m4} = \frac{0.28128 \times 1540 + 0.0236 \times 1490}{0.3049} = 1536.5kJ / kg$$

After finding the value of h4, it can be located points 4 and 5 on the P-h diagram and it can be h5= 1680 kJ/kg



Comp. power L.P.= $m_{f1}.(h_2\text{-}h_1)$= 0.28128(1600-1450)=42.192 k w

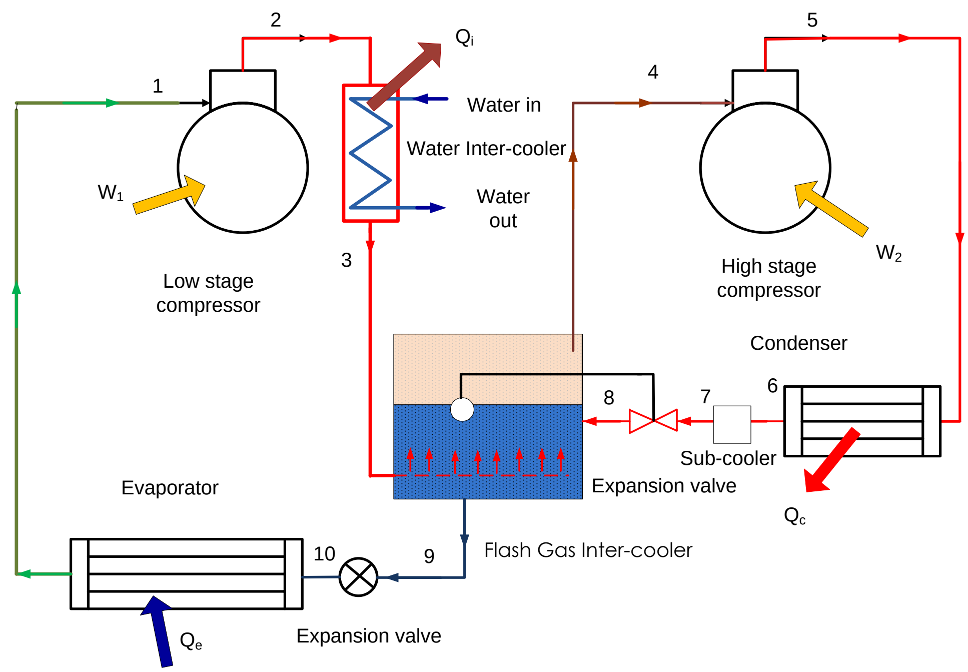
Comp. power H.P.= $m_{f4}.(h_5\text{-}h_4)$= 0.3049x(1680-1536.5)=43.78 kW

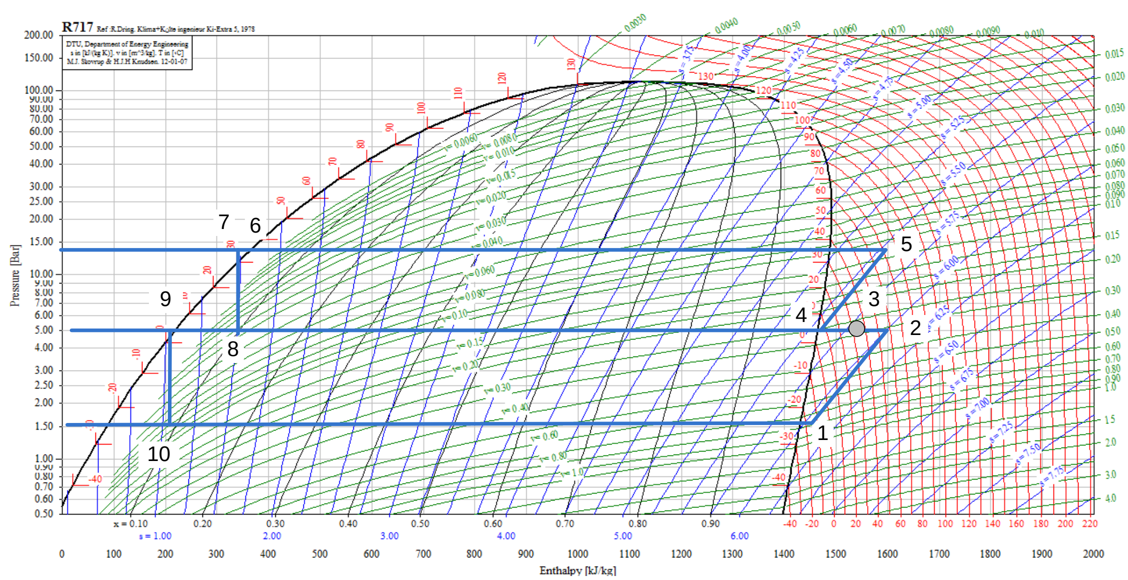
$Q_{inter\text{-}cooler}$= $m_{f1}.(h_2\text{-}h_3)$=0.28128x(1600-1540)=16.87 kW

$$COP = \frac{Capacity}{Wcomp1 + Wcomp2} = \frac{100 \times 3.516}{42.192 + 43.78} = 4.089$$

1-5Compound compression with Flash gads inter cooler

the gas leaving the first stage compression passes through a flash intercooler. The flash intercooler is a tank filled with liquid refrigerant at intermediate pressure, the liquid from this tank can be removed from the bottom and the gas can be removed from the top. The vapour passes to high pressure cylinder and is compressed to condenser pressure, and liquid passes through the second expansion valve to the evaporator. From the figure it can be seen that there are two cycles. Low pressure cycle namely 1-2-3-4-9-10-11-1 produces refrigeration effect in the evaporator, and the high pressure cycle namely 4-5-6-7-8-4 produce heat rejection from the system at condenser.



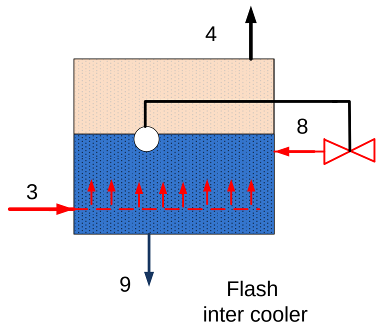


Flash inter cooler heat balance

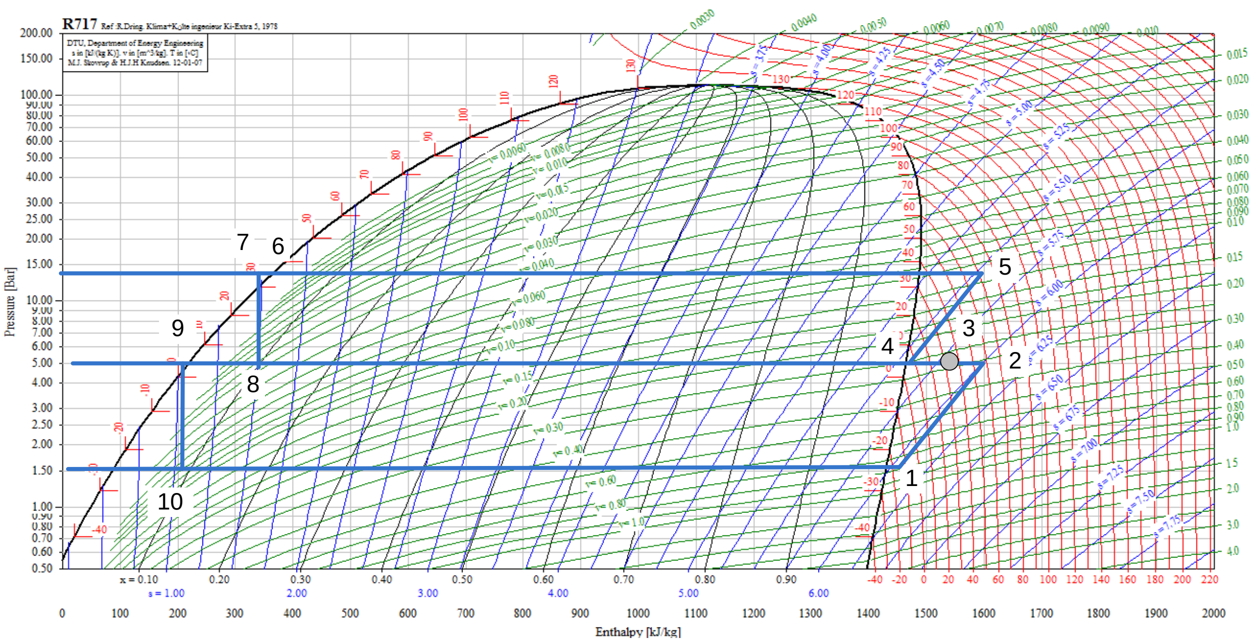
$m_8 h_8 + m_3 h_3 = m_4 h_4 + m_9 h_9$

$m_3 = m_9$ and $m_8 = m_4$

$$m4 = \frac{m3(h9 - h3)}{h8 - h4}$$



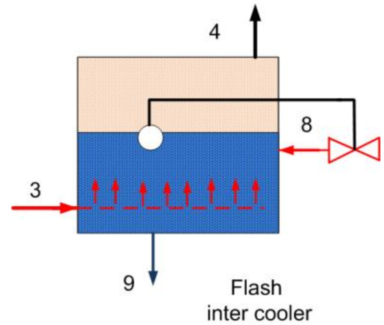
Example 3: The following data apply to 100 ton compound ammonia compression system with water inter-cooler and flash gas inter-cooler, condenser pressure 14 bar, evaporator pressure 1.5 bar and inter-cooler pressure 5 bar, the ammonia is cooled to 32°C in the water inter-cooler and sub-cooled to 30 °C in the sub-cooler,. The temperature of suction gas is -18 °C; find a- mass flow rate of refrigerant in the cycle b- power consumption in each compressor c- heat rejection in water inter-cooler d-COP



h_1	h_2	h_3	h_4	h_5	h_7	h_9	h_{10}
1450	1600	1540	1490	1610	360	200	200

Capacity =m.(h_1-h_{11})

Capacity =100X 3.516 kW



$$m4 = \frac{0.2818(200-1540)}{360-1490} = 0.33 kg/s$$

$$mf = \frac{100\times 3.516}{(1450-200)} = 0.28128 = m1 = m10$$

$m_8.h_8 + m_3.h_3 = m_4.h_4 + m_9.h_9$

$m_9 = m_{10} = m_1 = m_2 = m_3$

$m_4 = m_5 = m_6 = m_7 = m_8$

$m_4.h_8 + m_3.h_3 = m_4.h_4 + m_3.h_9$

$m_4.(h_8 - h_4) = m_3.(h_9 - h_3)$

$$m4 = \frac{m3(h9-h3)}{h8-h4}$$

$$m4 = \frac{0.2818(200-1540)}{360-1490} = 0.33 kg/s$$

Comp. power L.P.= m_{f1}.(h_2-h_1)= 0.28128(1600-1450)=42.192 k w

Comp. power H.P.= m_{f4}.(h_5-h_4)= 0.33(1610-1490)=39.6 kW

$Q_{inter-cooler}$= m_{f1}.(h_2-h_3)=0.28128x(1600-1540)=16.87 kW

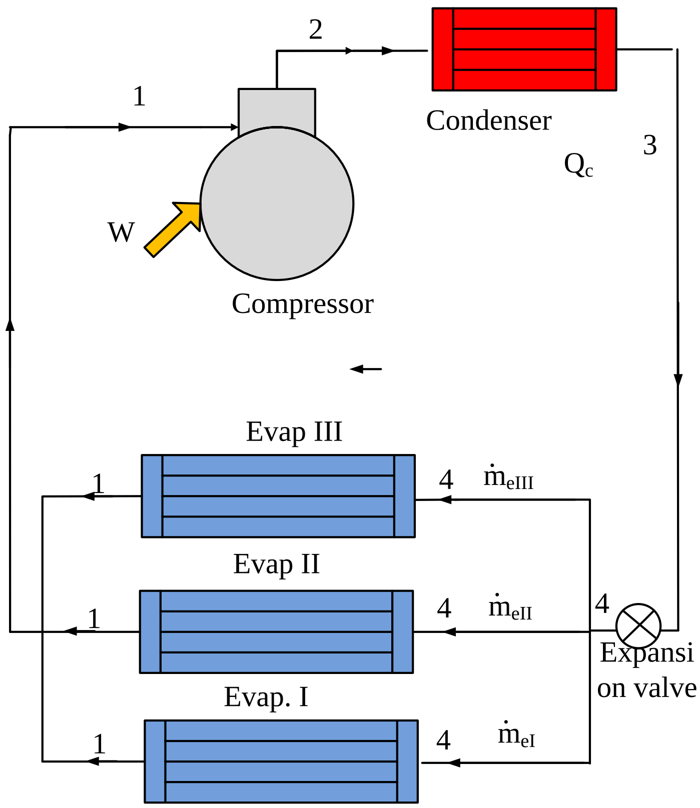
$$COP = \frac{Capacity}{Wcomp1 + Wcomp2} = \frac{100\times 3.516}{42.192+39.6} = 4.298$$

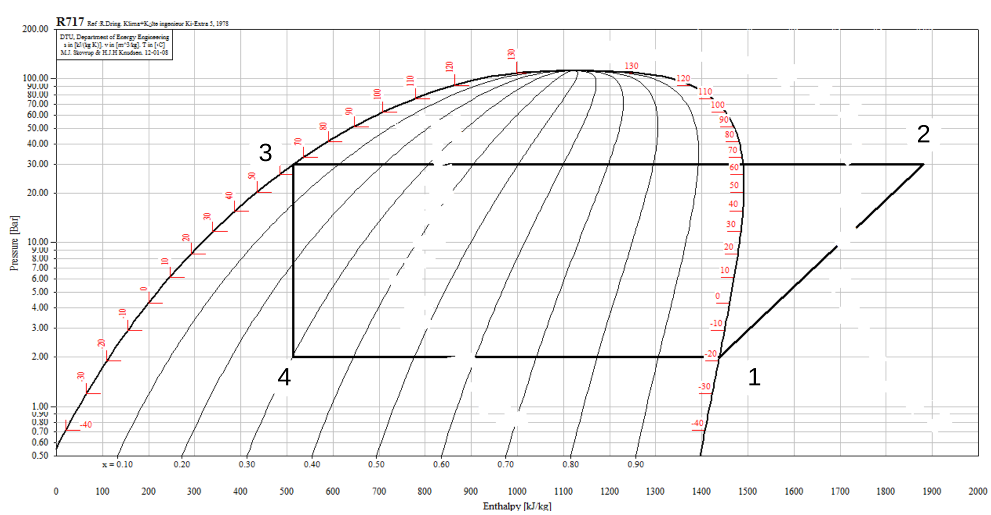
Multiple Evaporator Systems

In many refrigeration plants, it may be required to have different temperature maintained at various zones or compartments. Foe example in a food market where different food product stored in large quantities, it is imperative that each food product should be stored at different conditions of temperature and humidity.

1-All Evaporators Operating at The Same Temperature

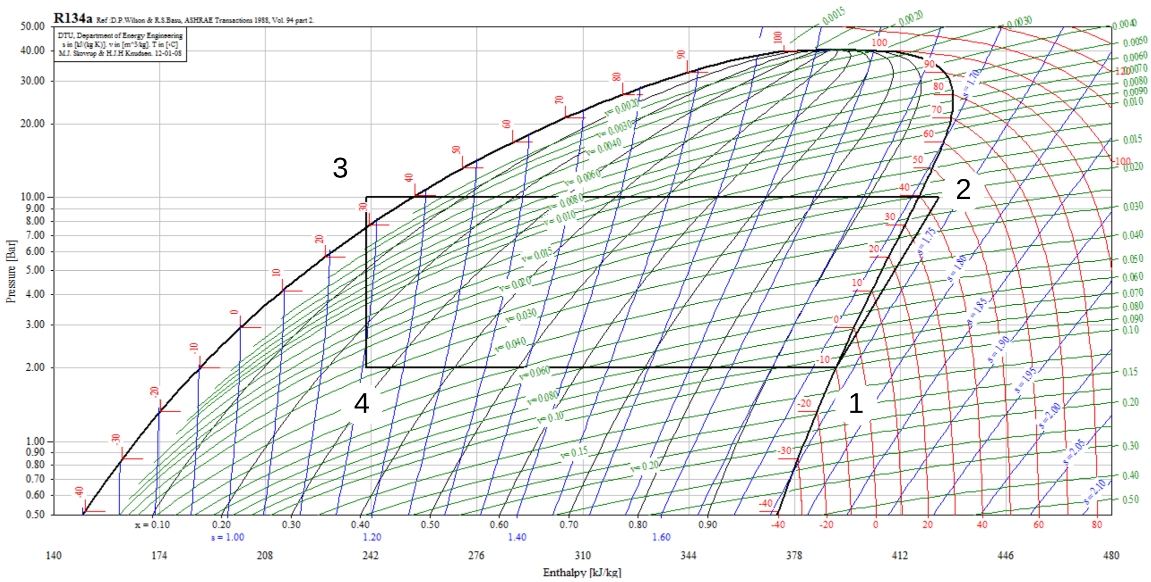
This method is not desirable particularly when food stuffs or other materials are kept in room maintained at high temperature. Due to very large different in room air and the evaporator coil temperature and when the coil temperature is much below the dew point temperature of the room air, there is much dehumidification, resulting in dehydrating the food. Thus from above it could be seen that a huge evaporator should be used.





Example 4:

A single compressor system using 134a has three evaporators of 10, 20 and 30 tons capacities. All evaporators operating at the same temperature of -10^{o}C. The condenser pressure is 10 bar and the liquid is sub cooled in the condenser by 10^{o}C. The discharge from the evaporator is dry saturated vapour and the compression is assumed to be isentropic. Determine 1-the refrigeration effect 2- rate of flow of the refrigerant in kg/s and 3-the theoretical horse power required.



h_1	h_2	h_3
391	425	240

Refrigeration effect $=h_1-h_4=391 - 240= 151$ kJ/kg
Total capacity of the cycle=10+20+30=60ton=60x3.516=210.84 kW
Capacity =m.x reff. Effect
210.8= mx 151
m=1.39 kg/s
Power=m.(h_2-h_1)=1.39 x (425-391) = 47. 26 =1.34x47.26 = 63.3 HP
Mass flow rate at each evaporator
Evap. III

$$m_{III} = \frac{10 \times 3.516}{151} = 0.23\ kg/s$$

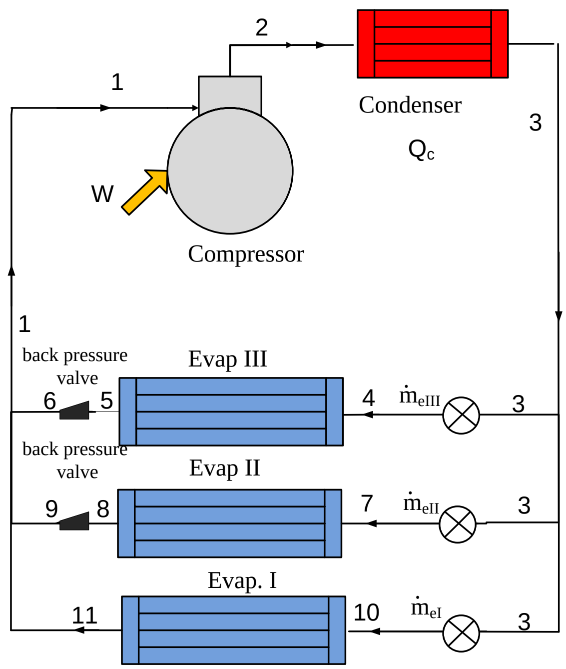
$$m_{II} = \frac{20 \times 3.516}{151} = 0.46\ kg/s$$

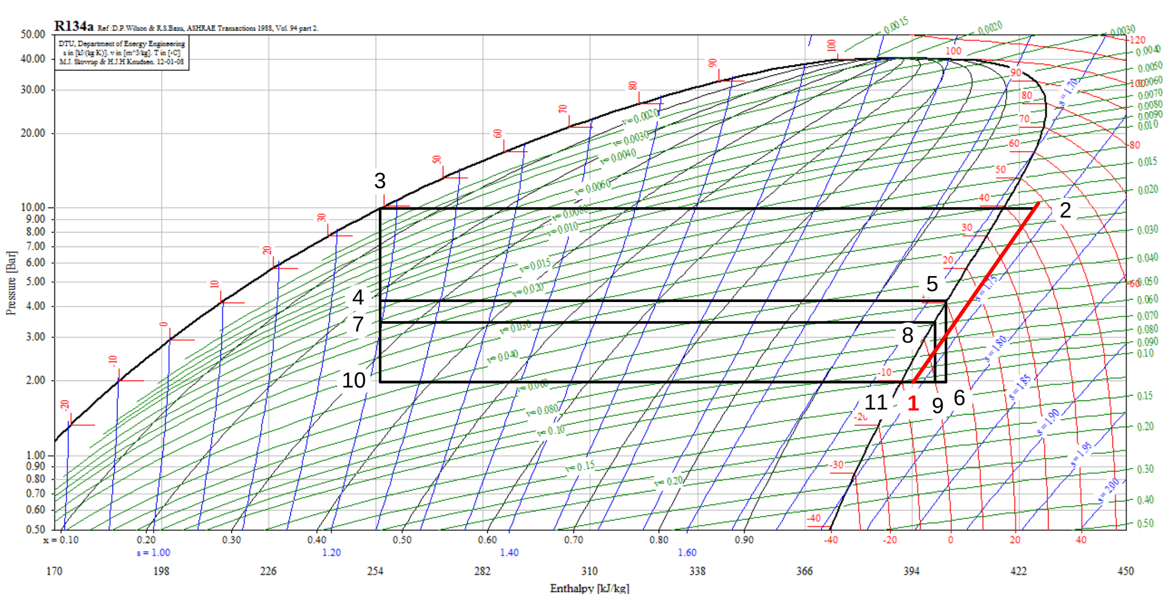
$$m_{II} = \frac{30 \times 3.516}{151} = 0.69\ kg/s$$

Individual expansion valve and back pressure valves system

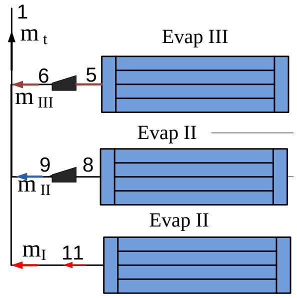
This system of one compressor and three evaporators is shown in the figure.

In this system, the range of pressure ratio for the compressor has to be maintained such that it compresses that gas from the lowest pressure corresponding to the lowest temperature evaporator, to the condenser pressure. The mass flow rate of refrigerant to various evaporators depends upon the loading of the evaporators. Suppose the evaporator I operate at higher temperature than evaporator II and evaporator II is at higher temperature than III, therefore a suitable quantity of refrigerant liquid say (m_1) out of m kg passes to expansion valve I and throttles to the pressure corresponding to the temperature of evaporator I. this pressure is higher than the pressure corresponding to the temperature corresponding to the evaporator III. And the suction of the compressor is maintained at the pressure required for evaporator III. Therefore gas leaving evaporator I is throttled in the back pressure valve to the pressure of evaporator III. Same process is applicable to the refrigerant of mass flow rate m2 corresponding to the load of evaporator II. The gas leaving evaporator I is mixed with gas leaving evaporator II and III after back pressure valves and sent to compressor suction.





Heat balance of suction line



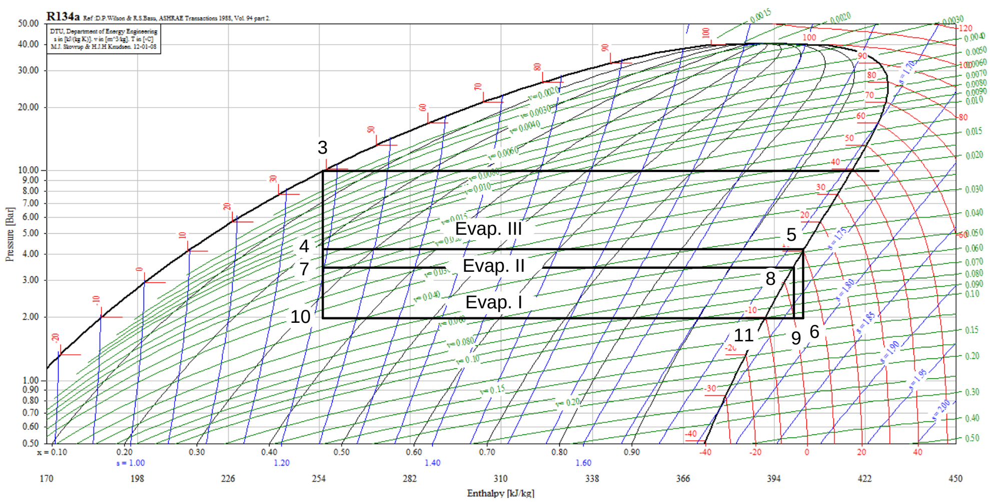
$m_t . h_1 = m_{III} . h_6 + m_{II} . h_9 + m_I . h_{11}$

$mt = m_{III} + m_{II} + m_I$

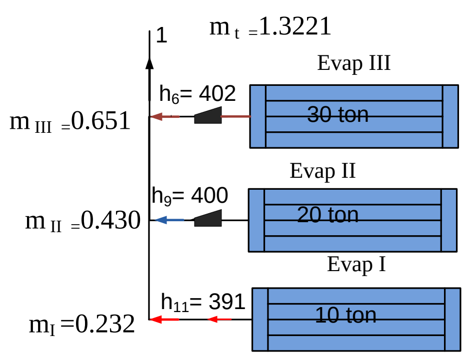
$$h_1 = \frac{m_{III}h_6 + m_{II}h_9 + m_I h_{11}}{m_{III} + m_{II} + m_I}$$

Example 5

A refrigeration installation using R-134a comprises one compressor, one condenser and three evaporators of capacities of 10, 20 and 30tons respectively. The temperature to be maintained in the evaporators is 10, 5 and -10^{o}C respectively. Each evaporator is fitted with an expansion valve. The condenser pressure is to be 10 bar. The exit condition from evaporator is dry saturated vapour and liquid is sub cooled by 10^{o}C in the condenser. determine 1-the refrigeration effect 2- rate of flow of the refrigerant in kg/s and 3-the theoretical horse power required.



h_1	h_2	h_3	h_4	h_5	h_6	h_7	h_8	h_9	h_{10}	h_{11}
?	?	240	240	402	402	240	400	400	240	391



Evaporator I

Ref. effect= h_{11}-h_{10}=391-240= 151 kJ/kg

$$m_I = \frac{10 \times 3.516}{151} = 0.232 \quad kg/s$$

Evaporator II

$$m_{II} = \frac{20 \times 3.516}{(h8 - h7)} = \frac{20 \times 3.516}{400 - 240} = 0.439 \quad kg/s$$

Evaporator III

$$m_{III} = \frac{30\times3.516}{(h5-h4)} = \frac{30\times3.516}{402-240} = 0.651\,kg/s$$

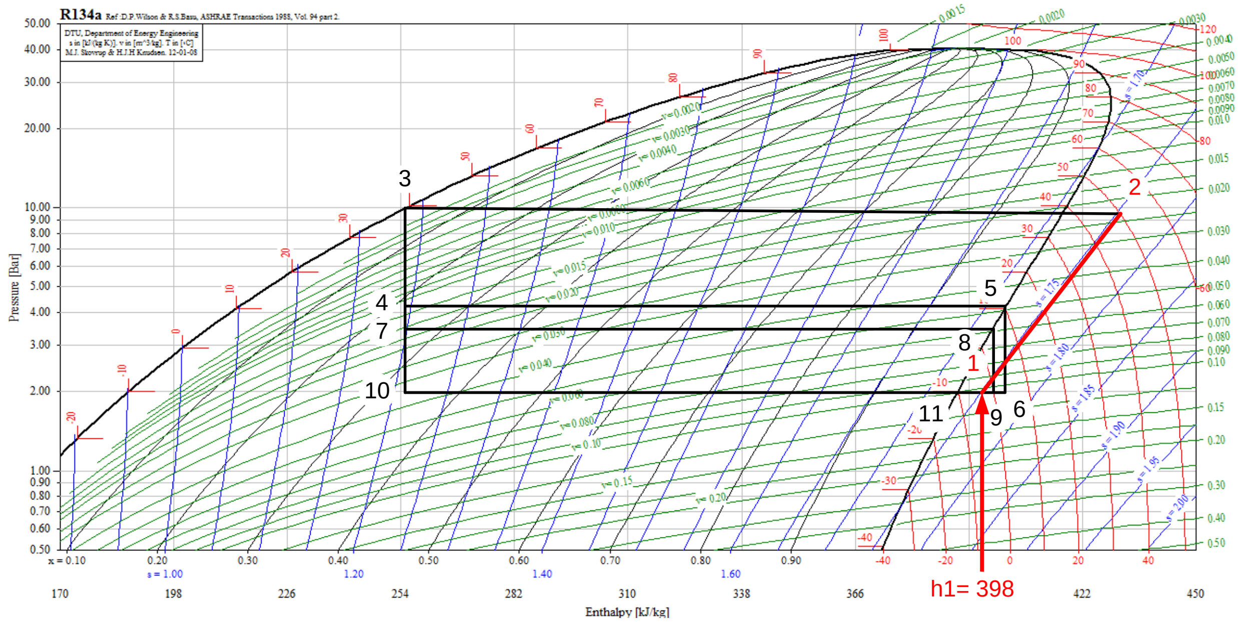
mt=0.232+0.439+0.651=1.322 kg/s

$m_t.\ h_1 = m_{III}.\ h_6 + m_{II}\ .\ h_9 + m_I.\ h_{11}$

$mt = m_{III} + m_{II} + m_I$

$$h_1 = \frac{m_{III}h_6 + m_{II}h_9 + m_I h_{11}}{m_{III} + m_{II} + m_I}$$

$$h_1 = \frac{0.651\times402 + 0.439\times400 + 0.323\times391}{1.413} = 398\ kJ/kg$$



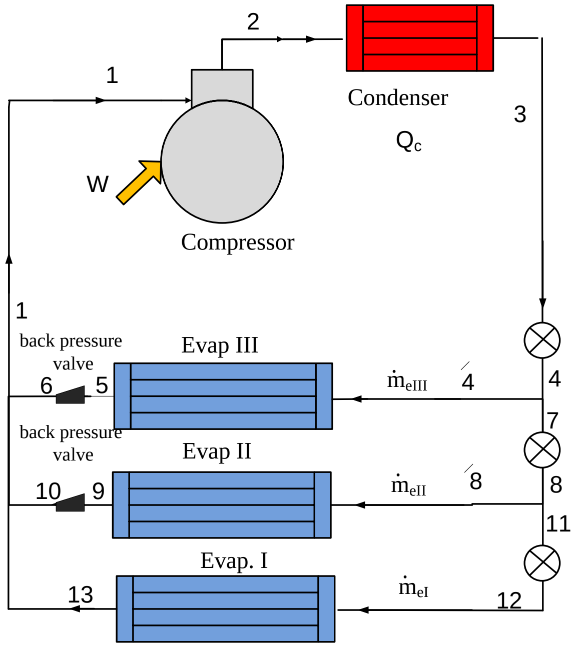
h_2= 430 kJ/kg

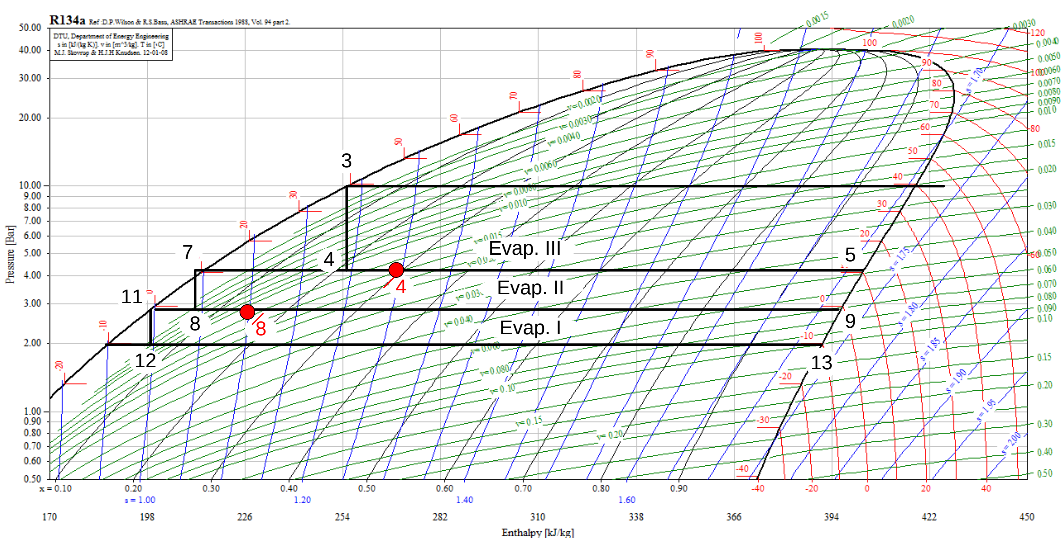
comp. Power=m(h_2-h_1)= 1.322x(430 - 398)= 42.3 4kW=42.3 x1.34= 56.6 HP

Multiple expansion valves and back pressure valves system

In this system the pressure ratio of the compressor is based on the condenser pressure and the lowest temperature evaporator pressure. The first expansion valve handle total mass flow rate in that the liquid refrigerant throttles from condenser pressure to the pressure of evaporator I. all the vapour formed plus enough liquid passes through evaporator I. the remaining liquid flow through the second expansion valve and expands to the pressure of evaporator II. The formed vapour and enough liquid passes through evaporator II. Remaining liquid expands in third expansion valve and passes through evaporator III.

The leaving gas from evaporators I & II is throttled in the back pressure valves to the pressure of the evaporator III. The three streams mix and form the suction to the compressor.

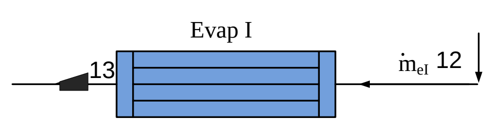




Cycle analysis

$Q_{evap\ I} = m_{12}\ (h_{13}-h_{12})$

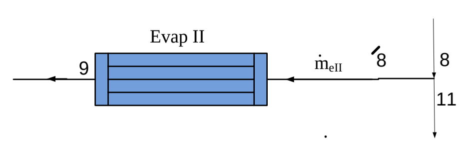
Heat balance



$$m_8 . h_8 = m_{\acute{8}} . h_{\acute{8}} + m_{11} . h_{11}$$

$$m_8 = m_{\acute{8}} + m_{11}$$

$$(m_{\acute{8}} + m_{11}) . h_8 = m_{\acute{8}} . h_{\acute{8}} + m_{11} . h_{11}$$



$$h_{\acute{8}} = \frac{(m_{\acute{8}} + m_{11}) . h_8 - m_{11} . h_{11}}{m_{\acute{8}}}$$

$$Q_{\text{evapII}} = m_{\acute{8}}.(h_9 - h_{\acute{8}}) = m_{\acute{8}}.\left(h_9 - \frac{(m_{\acute{8}} + m_{11}).h_8 - m_{11}.h_{11}}{m_{\acute{8}}}\right)$$

$$Q_{\text{evapII}} = m_{\acute{8}}.h_9 - (m_{\acute{8}} + m_{11}).h_8 + m_{11}.h_{11}$$

$$Q_{\text{evapII}} = m_{\acute{8}}.(h_9 - h_8) - m_{11}.(h_8 - h_{11})$$

$$Q_{\text{evapII}} + m_{11}.(h_8 - h_{11}) = m_{\acute{8}}.(h_9 - h_8)$$

$$m_{\acute{8}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + \frac{m_{11}.(h_8 - h_{11})}{(h_9 - h_8)}$$

$$m_{\acute{8}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + \frac{\frac{m_{11}.(h_8 - h_{11})}{(h_9 - h_{11})}}{\frac{(h_9 - h_8)}{(h_9 - h_{11})}}$$

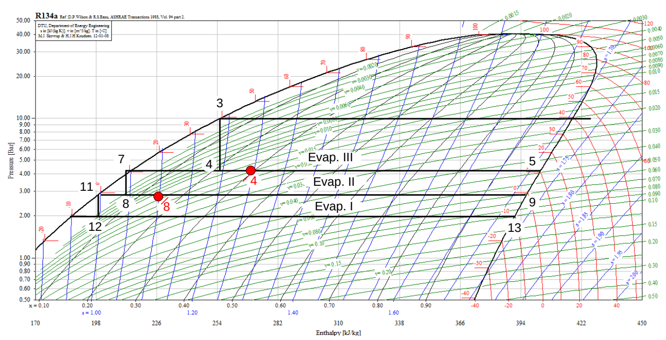
$$m_{\acute{8}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + m_{11}\frac{x_8}{(1 - x_8)}$$

$$m_8 = m_{\acute{8}} + m_{11}$$

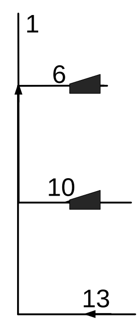
Similarly

$$m_{\acute{4}} = \frac{Q_{\text{evapIII}}}{(h_5 - h_4)} + m_7 \frac{x_4}{(1 - x_4)}$$

$$m_4 = m_{\acute{4}} + m_7$$



Heat balance of point 1



$$m_1.h_1 = m_6.h_6 + m_{10}.h_{10} + m_{13}.h_{13}$$

$$h_1 = \frac{m_6.h_6 + m_{10}.h_{10} + m_{13}.h_{13}}{m_1}$$

$$m_1 = m_4$$

$$m_{\acute{4}} = m_5 = m_6$$

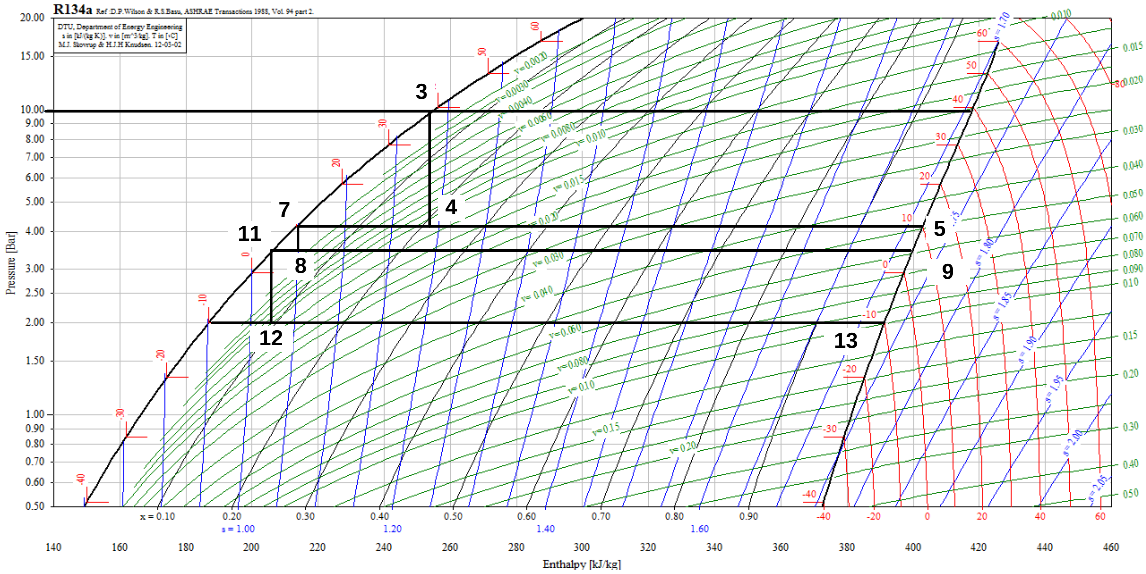
$$m_{\acute{8}} = m_9 = m_{10}$$

$$m_{12} = m_{13}$$

$$h_1 = \frac{m_{\acute{4}}.h_6 + m_{\acute{8}}.h_{10} + m_{12}.h_{13}}{m_4}$$

Example 6

A refrigeration plant using R-134a, the plant have one compressor, one condenser and three evaporators of 30, 20 and 10 ton papacies, maintained at 10, 5 and -10^{o}C respectively. The system comprises multiple- expansion valves and back pressure valves. The condenser pressure is 10 bar. The vapour leaving the evaporators are dry saturated. Assume isentropic compression. Determine; 1-the refrigeration effect 2- rate of flow of the refrigerant in kg/s and 3-the theoretical horse power required.



h	3	5	7	9	11	13	1	2	
kJ/kg	255	403	213	400	204	391	?	?	

Evaporator I

$Q_{evap\ I} = m_{12}\,(h_{13}-h_{12})$

$10 \times 3.516 = m_{12}\,(391-204)$

$$m_{12} = \frac{10 \times 3.516}{391 - 204} = 0.188\ kg/s$$

$x_8 = 0.04$

$$m_{\acute{8}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + m_{11}\frac{x_8}{(1 - x_8)} = \frac{20 \times 3.516}{(400 - 213)} + 0.188\frac{0.04}{(1 - 0.04)} = 0.376 + 0.0078$$

$$m_{\acute{8}} = 0.383 kg/s$$

$$m_8 = m_{\acute{8}} + m_{11} = 0.383 + 0.188 = 0.571\frac{kg}{s}$$

$$x_4 = 0.22$$

$$m_{\acute{4}} = \frac{Q_{\text{evapIII}}}{(h_5 - h_4)} + m_8\frac{x_4}{(1 - x_4)} = \frac{30 \times 3.516}{403 - 255} + 0.571 \times \frac{0.22}{1 - 0.22} = 0.712 + 0.161$$

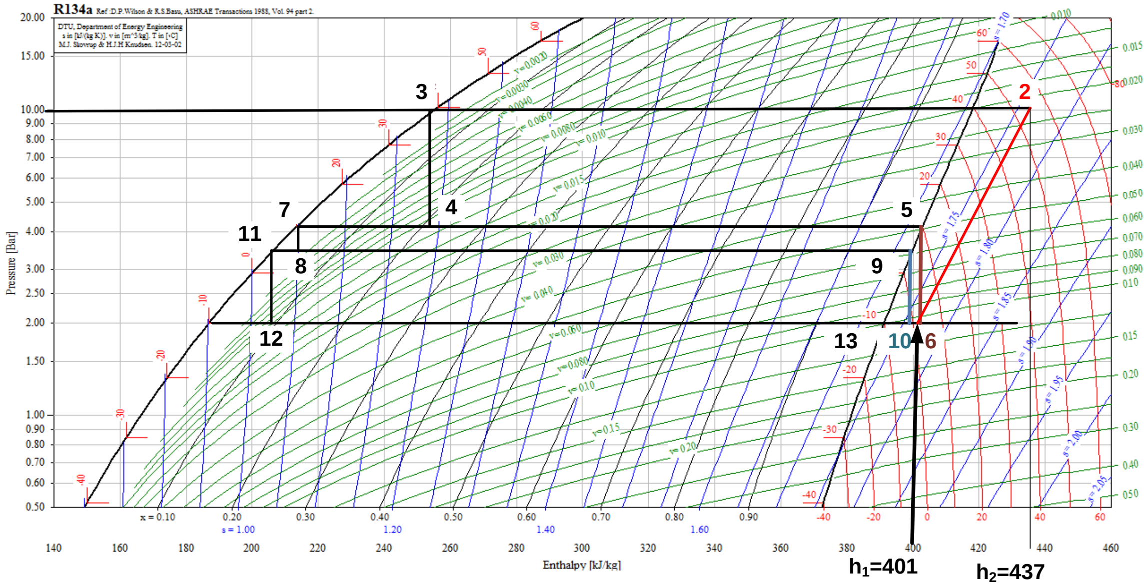
$$m_{\acute{4}} = 0.873\ kg/s$$

$$m_4 = m_{\acute{4}} + m_7 = 0.873 + 0.571 = 1.44\ kg/s$$

Heat balance of point 1

$$h_1 = \frac{m_{\acute{4}}.h_6 + m_{\acute{8}}.h_{10} + m_{12}.h_{13}}{m_4} = \frac{0.873 \times 403 + 0.383 \times 400 + 0.188 + 391}{1.44}$$

$$h_1 = 401\ kJ/kg$$



$h_2 = 437\ kJ/kg$

$Comp.Power = m_4(h_2 - h_1) = 1.44 \times (437 - 401) = 51.84 \quad kW$

$Comp.HP = 51.84 \times 1.34 = 69.5\ HP$

Components of vapour compression Refrigeration cycle

The objectives of this lesson are to:

1. Discuss basic components of a vapour compression refrigeration system
2. Present classification of refrigerant compressors based on working principle and based on the arrangement of compressor motor or external drive
3. Describe the working principle of reciprocating compressors
4. Discuss the performance aspects of ideal reciprocating compressors with and without clearance

At the end of the lesson, the student should be able to:

1. List important components of a vapour compression refrigeration system
2. Classify refrigerant compressors based on their working principle and based on the arrangement of compressor motor/external drive
3. Enumerate salient features of positive displacement type compressors, dynamic compressors, open and hermetic compressors
4. Draw the schematic of a reciprocating compressor and explain its working principle
5. Define an ideal reciprocating compressor without clearance using pressure-volume and pressure-crank angle diagrams
6. Calculate the required displacement rate and power input of an ideal compressor without clearance
7. Define an ideal reciprocating compressor with clearance using pressure volume and pressure-crank angle diagrams
8. Calculate the volumetric efficiency and power input of an ideal compressor with clearance, and
9. Discuss the effects of compression ratio and index of compression on the volumetric efficiency of a reciprocating compressor with clearance

Introduction

As we seen before the typical vapour compression refrigeration cycle consist of four major units, and they are compressor, condense, evaporator and expansion valve.

1- Compressors;

A compressor is the most important and often the costliest component (typically 30 to 40 percent of total cost) of any vapour compression refrigeration system. The function of a compressor is to raise the temperature of refrigerant vapour by rising pressure of the refrigerant to a level at which it can exchange heat with the ambient temperature and condenses by rejecting heat to the cooling medium in the condenser.

1-1 compressor classifications:

The compressors can be classified by different principles as follows:

1-1-1 Based on the working principle: two type of compressors are use in the vapour compression cycle namely:

a- Positive displacement compressors:

In positive displacement type compressors, compression is achieved by:

- trapping a refrigerant vapour into an enclosed space
- reducing its volume.

Since a fixed amount of refrigerant is trapped each time, its pressure rises as its volume is reduced. When the pressure rises to a level that is slightly higher than the condensing pressure, then it is expelled from the enclosed space and a fresh charge of low-pressure refrigerant is drawn in and the cycle continues.

Since the flow of refrigerant to the compressor is not steady, the positive displacement type compressor is a ***pulsating flow device***. However, since the operating speeds are normally very high the flow appears to be almost steady on macroscopic time scale. Since the flow is pulsating on a microscopic time scale, **positive displacement type compressors are prone to high wear, vibration and noise level**. Depending upon the construction, positive displacement type compressors used in refrigeration and air conditioning can be classified into:

i. Reciprocating type
ii. Screw type
iii. Scroll type
iv. Rotary compressor

b- Dynamic compressors

In dynamic compressors,

- the pressure rise of refrigerant is achieved by imparting kinetic energy to a steadily flowing stream of refrigerant by a rotating mechanical element
- then converting into pressure as the refrigerant flows through a diverging passage.

Unlike positive displacement type, the dynamic type compressors are steady flow devices, hence are subjected to less wear and vibration.

1-1-2 Based on arrangement of compressor motor or external drive:

a- Open type

the rotating shaft of the compressor extends through a seal in the crankcase for an external drive. The external drive may be an electrical motor or an engine (e.g. diesel engine). The compressor may be belt driven or gear driven.

Open type compressors are normally used in medium to large capacity refrigeration system for all refrigerants and for ammonia.

Open type compressors are characterized by:

- high efficiency,
- flexibility,
- better compressor cooling and serviceability.

However, since the shaft has to extend through the seal, refrigerant leakage from the system cannot be eliminated completely.

Hence refrigeration systems using open type compressors require a refrigerant reservoir to take care of the refrigerant leakage for some time, and then regular maintenance for charging the system with refrigerant, changing of seals, gaskets etc. Figure 1 shows the open type compressors.

b- Hermetic type

In hermetic compressors, the motor and the compressor are enclosed in the same housing to prevent refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and for power input socket. The compressor gets heated-up due to friction and also due to temperature rise of the vapor during compression. In Open type, both the compressor and the motor normally reject heat to the surrounding air for efficient operation.

In hermetic compressors heat cannot be rejected to the surrounding air since both are enclosed in a shell. Hence, **the cold suction gas is made to flow over the motor and the compressor before entering the compressor. This keeps the motor cool.** The cooling rate depends upon the flow rate of the refrigerant, its temperature and the thermal properties of the refrigerant. If flow rate is not sufficient and/or if the temperature is not low enough the insulation on the winding of the motor can burn out and short-circuiting may occur. Hence, hermetically sealed compressors give satisfactory and safe performance over a very narrow range of design temperature and should not be used for off-design conditions. The COP of the hermetic compressor based systems is lower than that of the open compressor based systems **since a part of the refrigeration effect is lost in cooling the motor and the compressor.** However, hermetic compressors are almost universally **used in small systems such as domestic refrigerators, water coolers, air conditioners** etc, where efficiency is not as important as customer convenience (due to absence of continuous maintenance). In addition to this, the use of hermetic compressors is ideal in systems, **which use capillary tubes as expansion devices and are critically charged systems**. Hermetic compressors are normally not serviceable. **They are not very flexible as it is difficult to vary their speed to control the cooling capacity.** , figure 2 shows the hermetic compressor.

c- Semi-hermetic type

In some (usually larger) hermetic units, the cylinder head is usually removable so that the valves and the piston can be serviced. This type of unit is called a semi-hermetic (or semi-sealed) compressor, figure 3 shows the semi-hermetic compressor.

Figure 4 shows the classifications of Refrigeration compressors

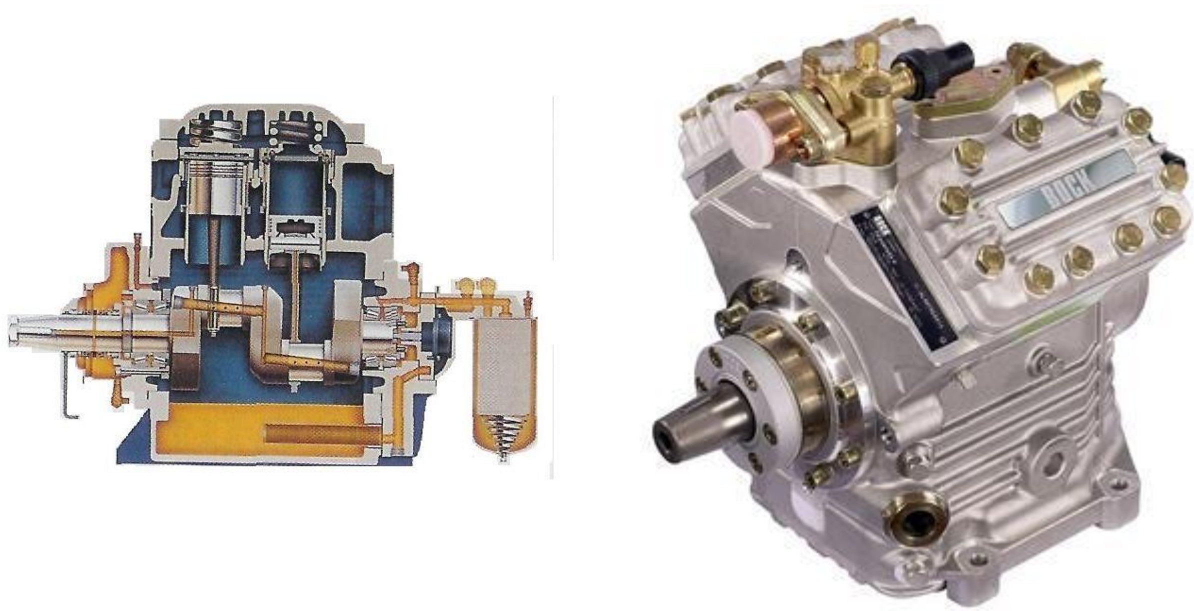


Figure 1 open type compressor

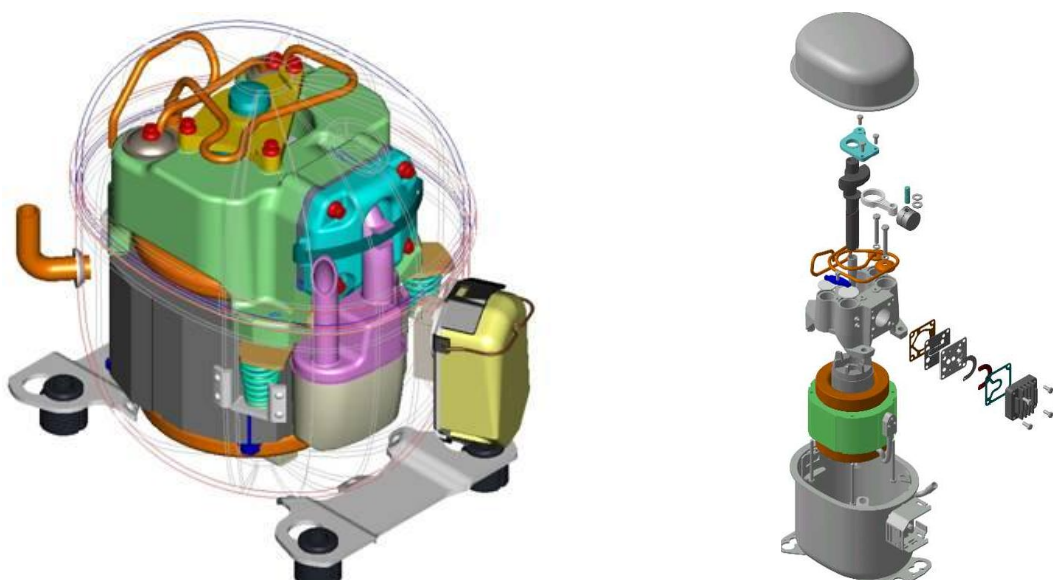


Figure 2 hermetic compressor

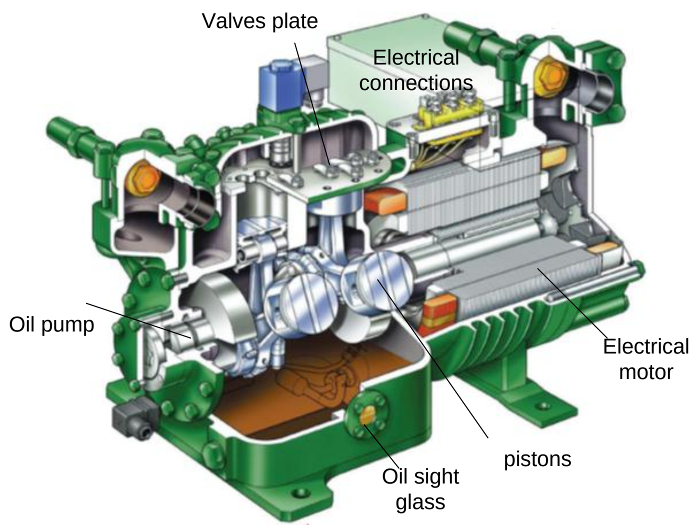


Figure 3 semi-hermetic compressor

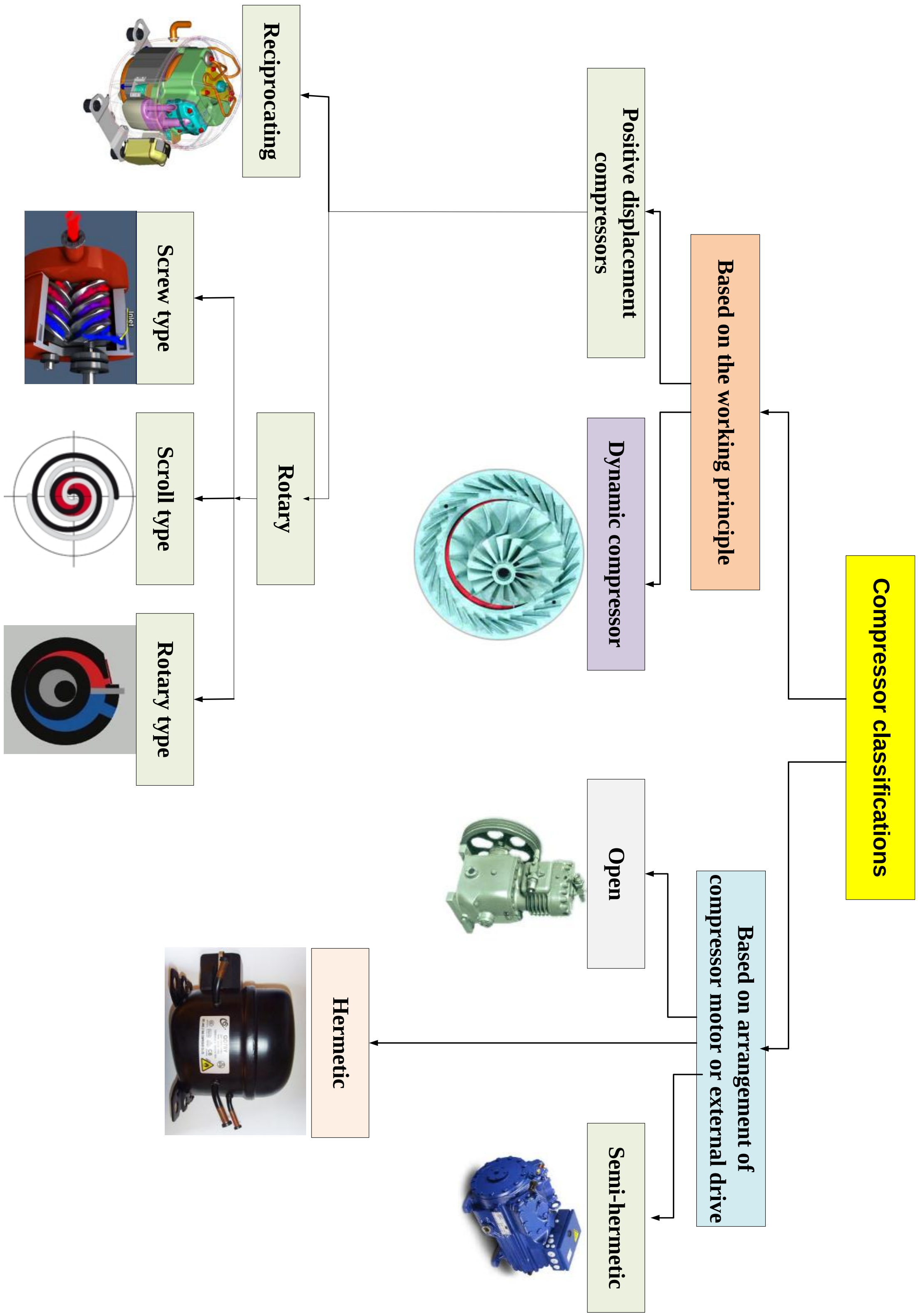
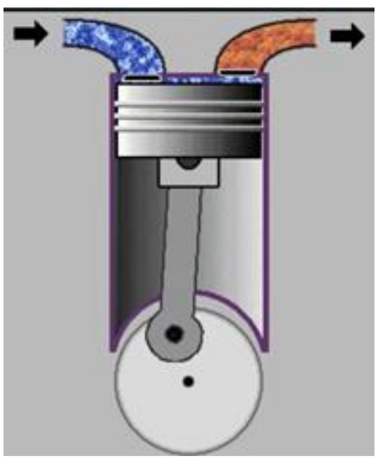


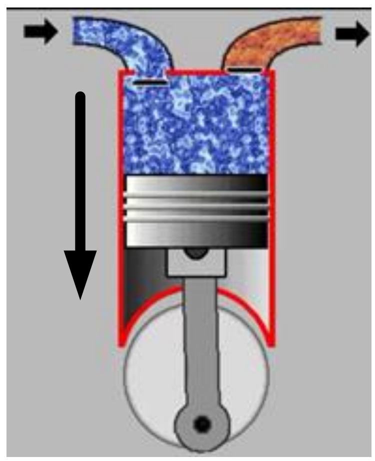
Figure 4 compressor classifications

i- Reciprocating compressor

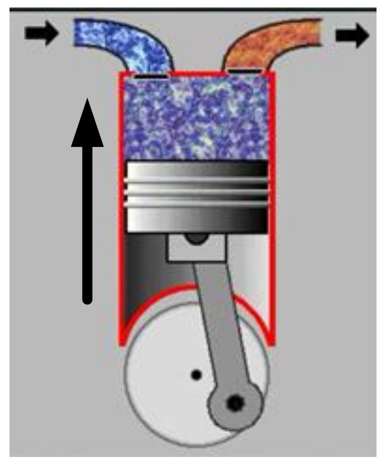
The reciprocating compressor (piston type) is very widely used, being adaptable in size, number of cylinders, speed and method of drive. It works on the two-stroke cycle. Automatic pressure-actuated suction and discharge valves are used as shown in Figure 5 . As the piston descends on the suction stroke, the suction valve opens to admit gas from the evaporator. At the bottom of the stroke, this valve will close again as the compression stroke begins. When the cylinder pressure becomes higher than that in the discharge pipe, the discharge valve opens and the compressed gas passes to the condenser. Gas left in the clearance at the top of the stroke must re-expand before a fresh charge can enter the cylinder, see Figure 5e. The suction valve will not open until the cylinder pressure is lower than the suction pressure. A larger re-expansion or *clearance volume* means the piston must travel further down the cylinder before the pressure falls below suction pressure. The further the piston travel, without the valve opening, the higher the losses.



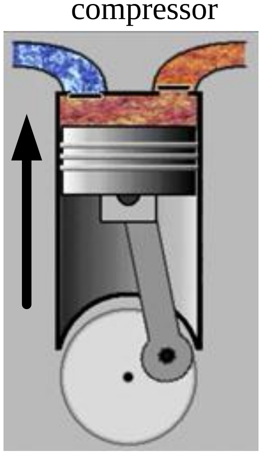
a- Reciprocating compressor



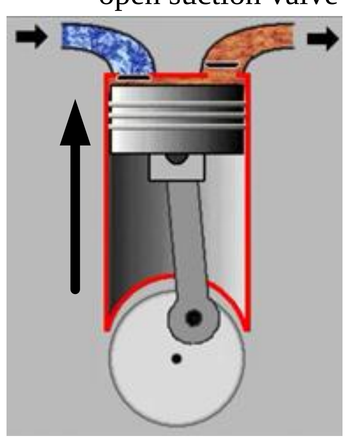
b- Piston moves down open suction valve



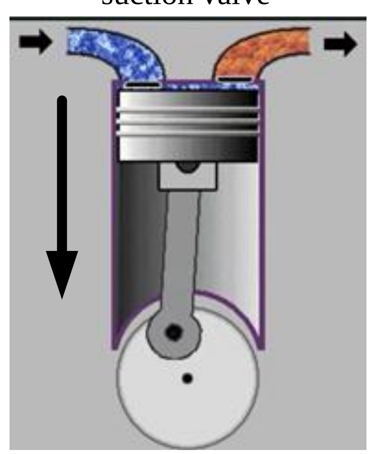
c- Piston moves up closed suction valve



d- Piston moves up compressed vapour



e- Open discharge valve



f- Piston moves down

Figure 5 how reciprocating compressor works

Multi cylinder compressor

To attain a higher capacity, compressors were made larger during the first century of development, having cylinder bores up to 375 mm and running at speeds up to 400 rev/min. The resulting component parts were heavy and cumbersome. To take advantage of larger-scale production methods and provide interchangeability of parts, modern compressors are multi-cylinder, with bores not larger than 175 mm and running at higher shaft speeds. Machines of four, six and eight cylinders are common. Figure 6 shows a large reciprocating machine built with a welded steel crankcase, the largest of this type being suitable for up to 1000 kW refrigeration. Cylinders are commonly arranged in banks with two, three or four connecting rods on the same throw of the crankshaft to give a short, rigid machine.

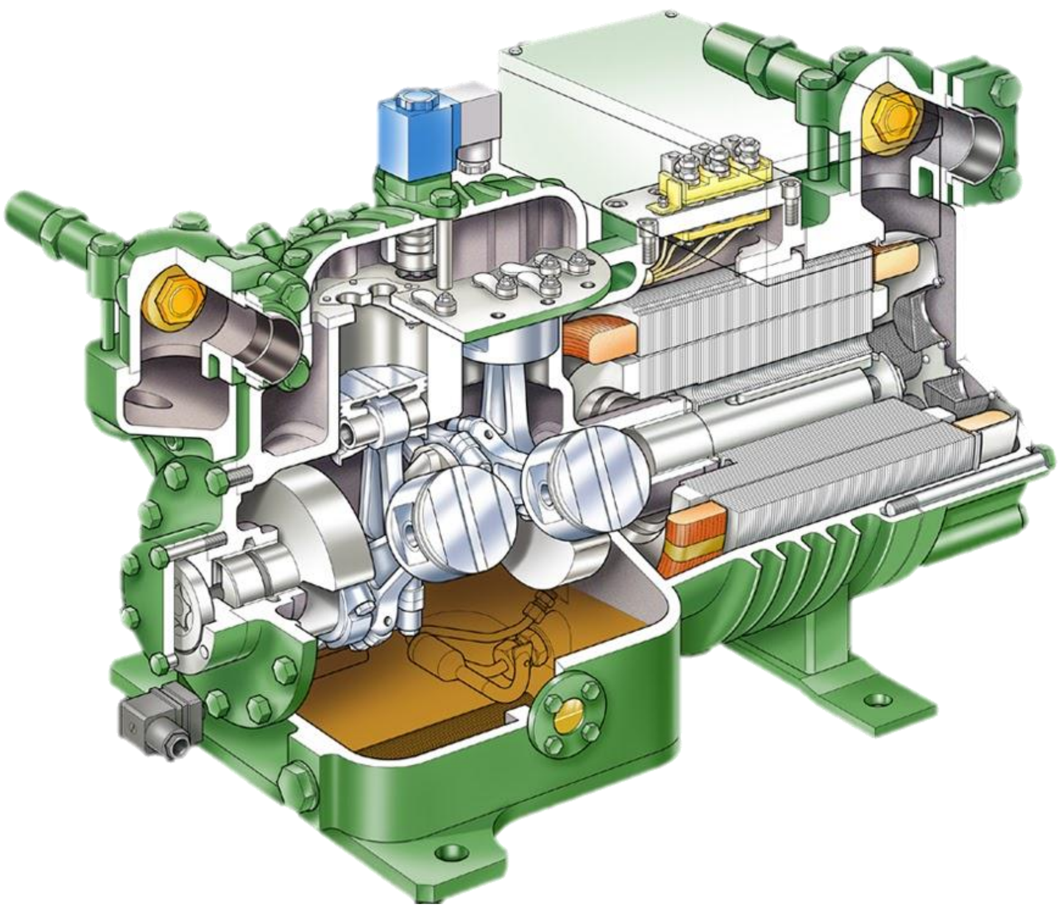


Figure 6 four cylinder reciprocating compressor

Valves:

Piston compressors may be generally classified by the type of valve, and this depends on size, since a small swept volume requires a proportionally small inlet and outlet gas port.

- Small compressors have spring steel reed valves for both inlet and outlet arranged on a valve plate and the differing pressures kept separated by the cylinder head (7).
- Above a bore of about 80 mm, a ring plate valves are used, these valves are made of thin spring steel or titanium, limited in lift and damped by light springs to assist even closure and lessen bouncing, see figure 8.

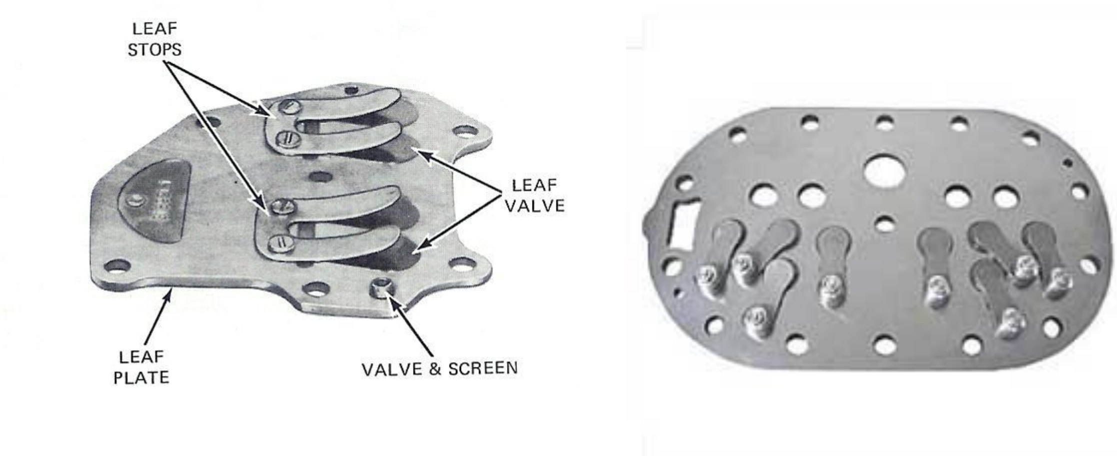


Figure 7 spring steel reed valves

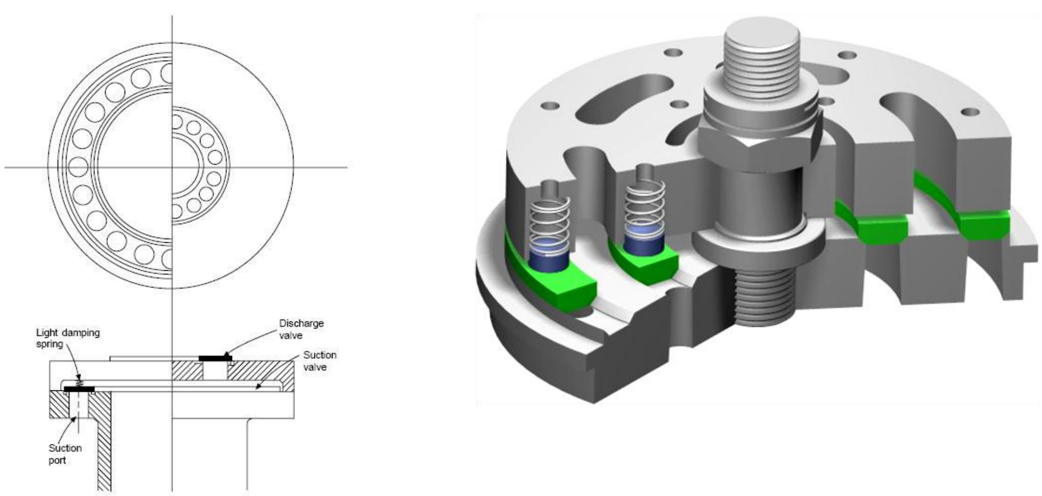


Figure 8 ring plate valves

New technology in compressors:

The new technology provides real-time intelligence monitoring, which lets the service technician know the status of what is happening inside the compressor before any major problems develop. Today, compressor monitoring operations can be centralized. This technology allows service technicians to systematically troubleshoot compressor problems before arriving at the site. It also improves troubleshooting accuracy and speed of service. Centralized monitoring technology can gather data, transmit operating information, and visually display compressor status and alarm codes on the front control box. It also can record and retain a history of the compressor's operating information and past alarms. This technology allows service technicians to be dispatched automatically if an alarm problem exists. **Figure 9** shows a modern discus compressor that incorporates onboard diagnostics. This new technology has the following advantages:

1. Monitors the compressor's discharge temperature
2. Provides contactor protection
3. Enables remote diagnostics
4. Integrates the compressor's system electronics including:

- the high- and low-pressure controls,
- the cooling and temperature control,
- oil pressure monitoring,
- motor protection devices,
- and input/ output (I/O) boards

5. Reduces the number of brazed joints on the compressor that can develop leaks
6. Guarantees consistent field installation

Figure 9 modern compressor that incorporates onboard diagnostics.

ii- Screw Compressors:

The *screw compressor* can be visualized as a development of the gear pump. For gas pumping the rotor profiles are designed to give **maximum swept volume and no clearance volume where the rotors mesh together**.

The more usual form has **twin meshing rotors on parallel shafts** (see Figures 10). As these turn inside the closely fitting casing, the space between two grooves comes opposite the inlet port, and gas enters. On further rotation, this pocket of gas becomes sealed from the inlet port and moved down the barrels. A meshing lobe of the male rotor then reduces the pocket volume compressing the gas, which is finally released at the opposite end, where the exhaust port is uncovered by the movement of the rotors as shown in figure 11. In most designs the female rotor is driven by the male rotor, and a study on optimizing rotor design for refrigeration applications is reported by Stosic et al. (2003). Maintenance of adequate lubrication is essential. Lubrication, cooling and sealing between the working parts is usually assisted by the injection of oil along the length of the barrels. This oil must be separated from the discharge gas and is then cooled and filtered before returning to the lubrication circuit.

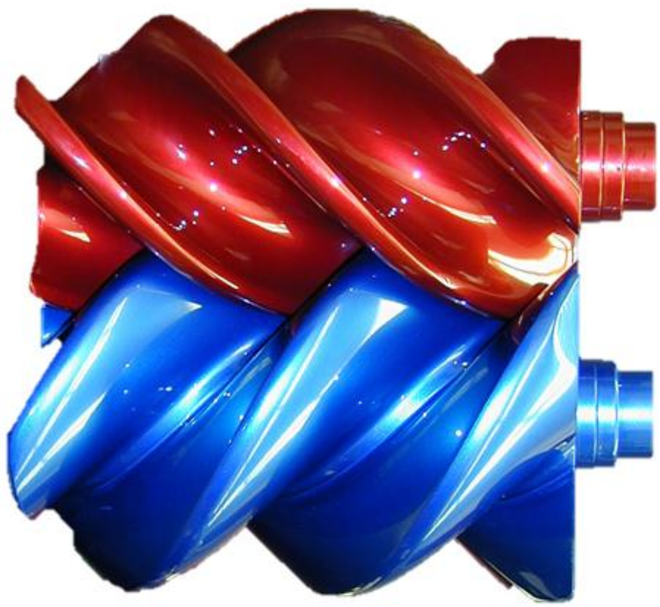
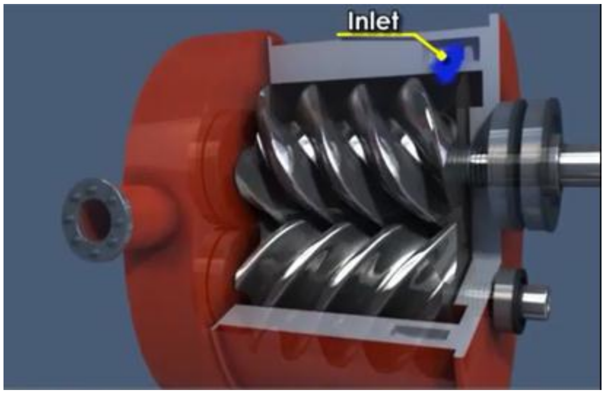
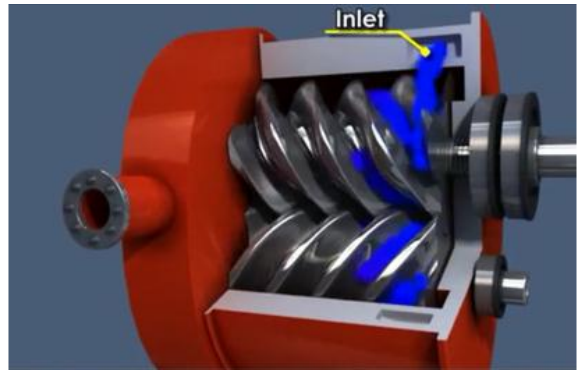


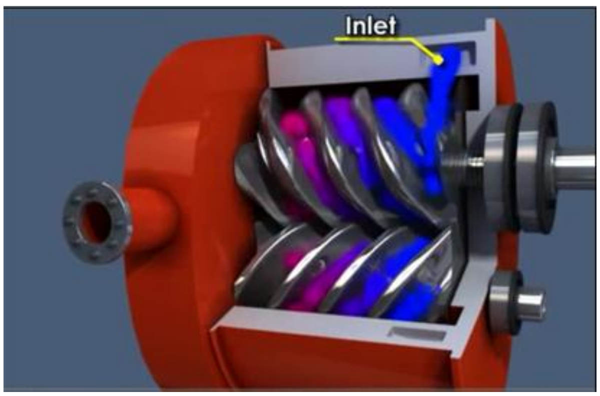
Figure 10 twin meshing rotors on parallel shafts



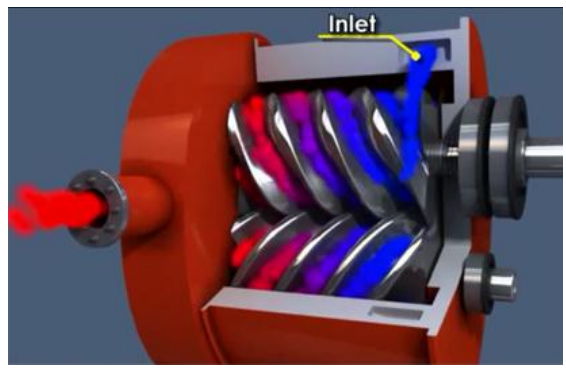
a- Inlet of refrigerant vapour



b- Suction stroke



c- Compression stroke



d- Discharge of high pressure vapour

Figure 11 how twin screw compressor works

The single screw compressor has a single grooved rotor, with rotating star tooth seal vanes to confine the pockets of gas as they move along the rotor flutes (see Figure 12). Once again, various geometries are possible, but compressors currently being manufactured have a rotor with six flutes and stars with eleven teeth. The normal arrangement is two stars, one on either side of the rotor. Each rotor flute is thus used twice in each revolution of the main rotor, and the gas pressure loading on the rotor is balanced out, resulting in much lighter bearing loads than for the corresponding twin screw design. The stars are driven by the rotor, and because no torque is transmitted the lubrication requirements in the mesh are lighter also.

a- Screw with two stars

b- Suction stroke of compressor

c- Compression stroke

d- Discharge of compressed vapour

Figure 12 normal arrangement of ingle screw compressor with two stars

iii- Scroll compressors

Scroll compressors are positive displacement machines that compress refrigerants with two inter-fitting spiral-shaped scroll members as shown in Figure 13. One scroll remains fixed whilst the second scroll moves in orbit inside it, as shown in figure 14. Note that the moving scroll does not rotate but orbits with a circular motion. Typically two to three orbits, or crankshaft revolutions, are required to complete the compression cycle. The scroll has certain common features with the screw. There is no clearance volume and hence no re-expansion loss. However, there is a very important difference in the sealing of the compression pockets. The screw relies on clearance between rotors and casing whereas the scroll can be built with contacting seals, i.e. the scrolls touch each other at the pocket boundaries. This is possible because the orbiting motion gives rise to much lower velocities than rotating motion, and also the load on the flanks and tips of the scrolls can be controlled. Figure 15 shows how scroll compressor works.

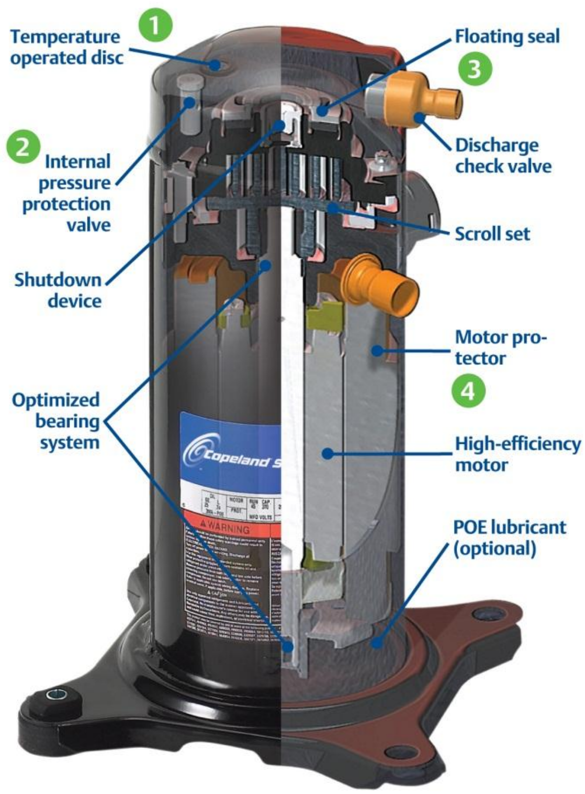


Figure 13 scroll compressor

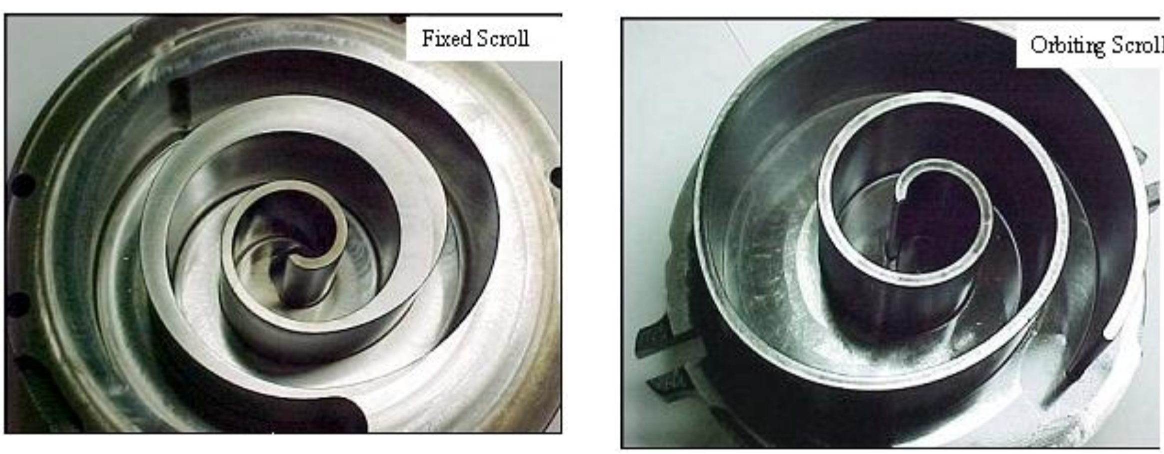
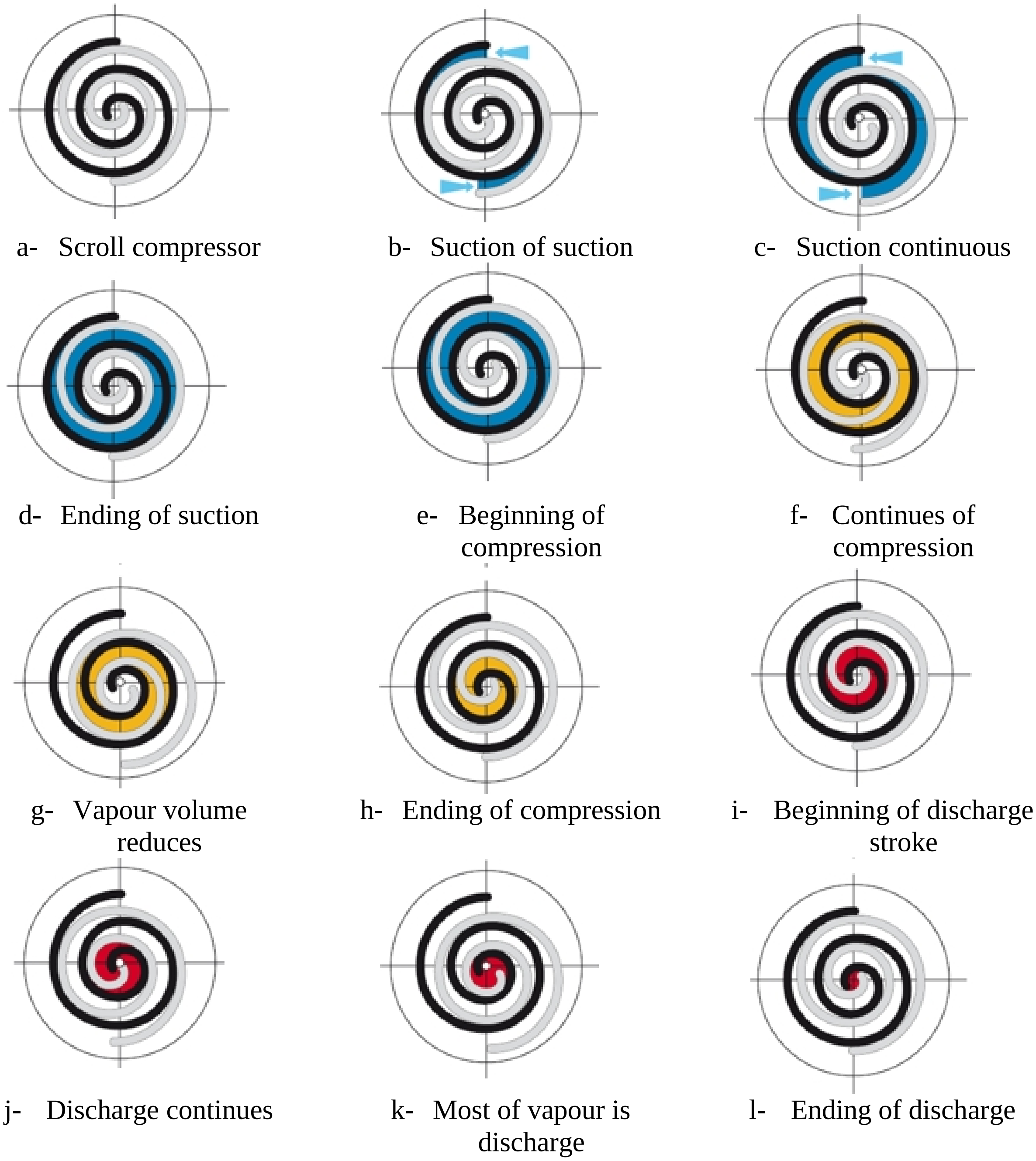


Figure 14 fixed and orbiting scroll



a- Scroll compressor
b- Suction of suction
c- Suction continuous
d- Ending of suction
e- Beginning of compression
f- Continues of compression
g- Vapour volume reduces
h- Ending of compression
i- Beginning of discharge stroke
j- Discharge continues
k- Most of vapour is discharge
l- Ending of discharge

Figure 15 how scroll compressor works

iv- Rotary compressor:

Two types of rotary compressor can be found as follow:

a- Rolling piston compressor:

Rolling piston, or fixed vane, rotary compressors are used in household refrigerators and air-conditioning units in sizes up to about 2 kW (Figure 16). This type of compressor uses a roller mounted on the eccentric of a shaft with a single vane or blade suitably positioned in the non-rotating cylindrical housing, generally called the cylinder block as shown in figure 17. The blade reciprocates in a slot machined in the cylinder block. This reciprocating motion is caused by the eccentrically moving roller. The drive motor stator and compressor are solidly mounted in the compressor housing. This design feature is possible due to low vibration associated with the rotary compression process, as opposed to reciprocating designs which employ spring isolation between the compressor parts and the housing. Suction gas is directly piped into the suction port of the compressor, and the compressed gas is discharged into the compressor housing shell.

This high-side shell design is used because of

- the simplicity of its lubrication system and
- the absence of oiling and compressor cooling problems.

Compressor performance is also improved because this arrangement minimizes heat transfer to the suction gas and reduces gas leakage area. Internal leakage is controlled through hydrodynamic sealing and selection of mating parts for optimum clearance. Hydrodynamic sealing depends on clearance, surface speed, surface finish, and oil viscosity. Close tolerance and low surface finish machining is necessary to support hydrodynamic sealing and to reduce gas leakage.

Figure 16 rotary compressor

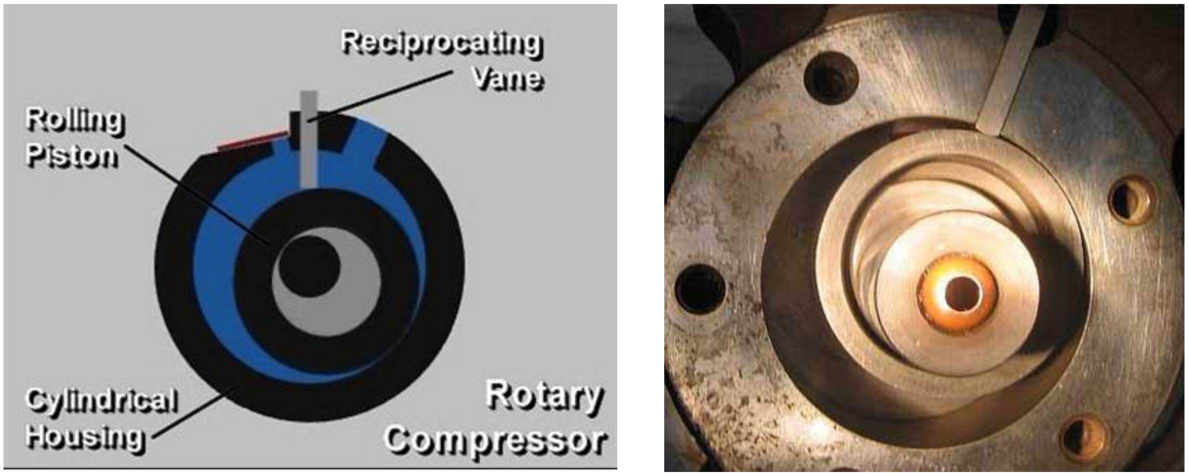
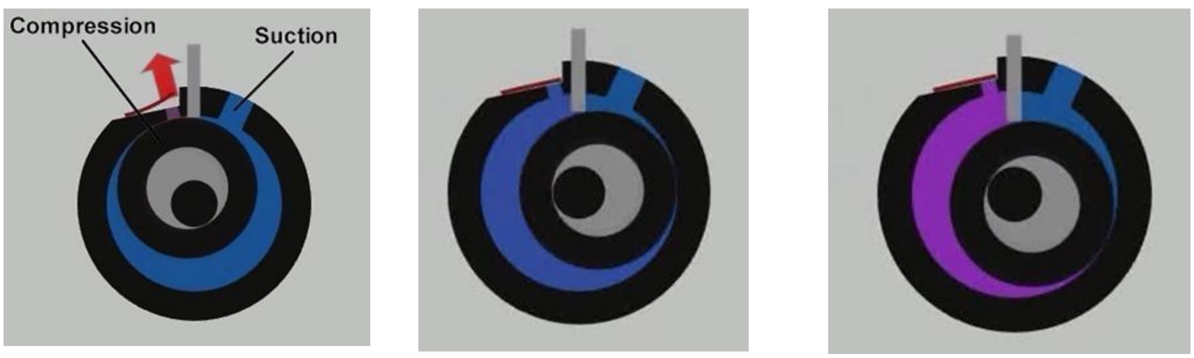
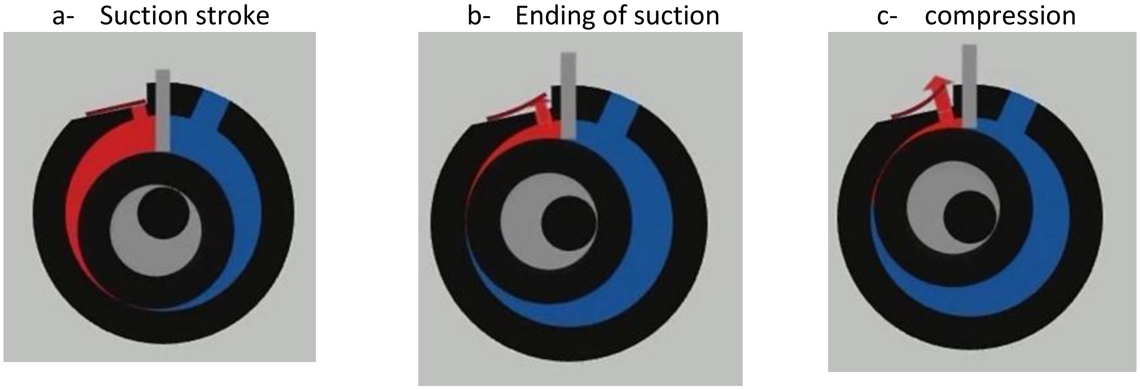


Figure 17 the rolling piston and cylindrical house



a- Suction stroke b- Ending of suction c- compression

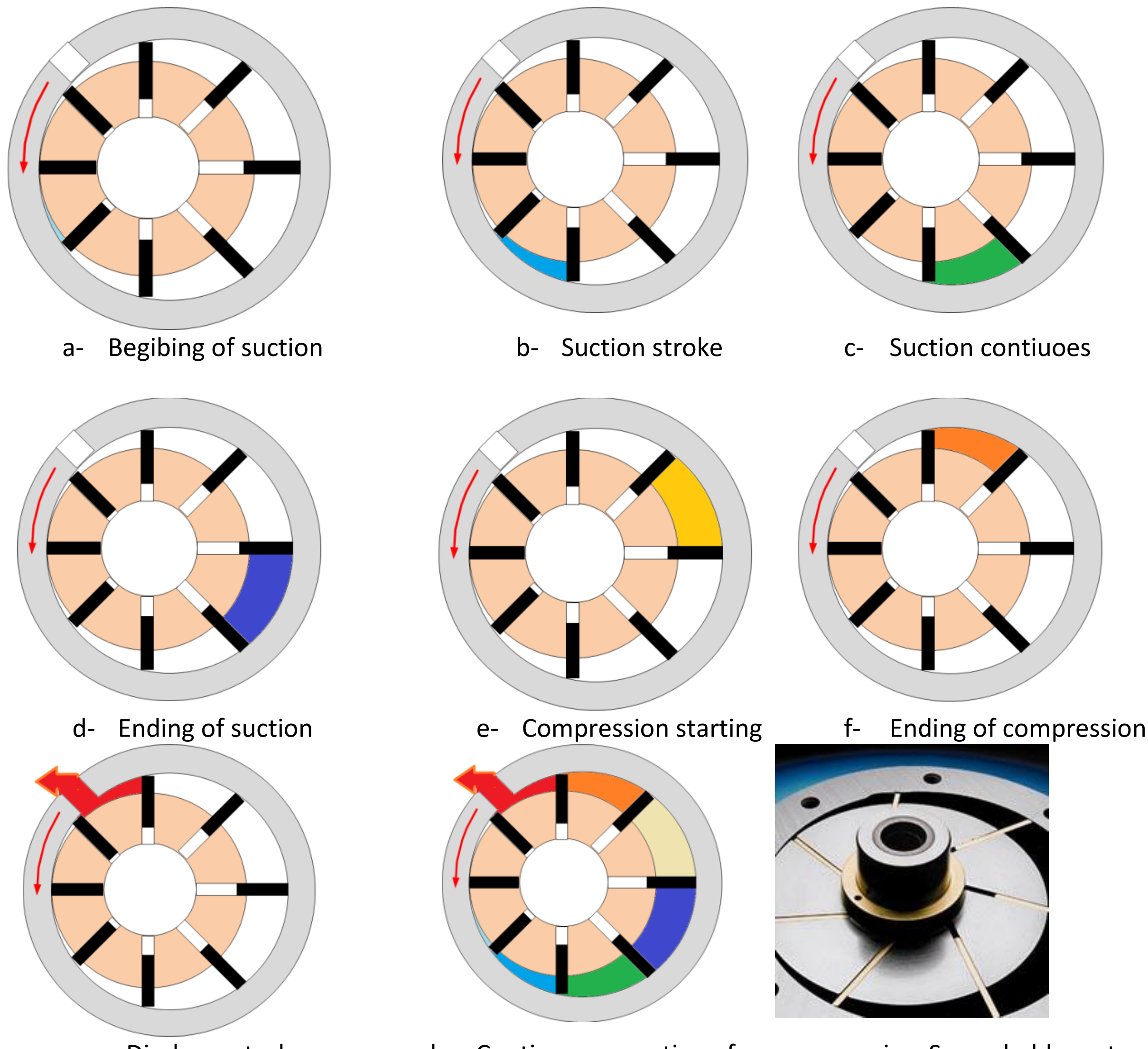


d- ending of compression e- beginning of discharge f- complete discharge

Figure 18 how rotary compressor works

b- Rotary Vane Compressor

Rotary vane compressors have a low mass-to-displacement ratio, which, in combination with compact size, makes them suitable for transport application. Small compressors in the 2 to 40 kW range are single-staged for a saturated suction temperature range of - 40 to 7°C at saturated condensing temperature up to 60°C. By employing a second stage, low-temperature applications down to -50°C are possible. Currently, R-22, R-404a, and R-717 refrigerants are used. Figure 19 is a cross-sectional view of an eight-bladed compressor and how it works. The eight discrete volumes are referred to as cells. A single shaft rotation produces eight distinct compression strokes. While conventional valves are not required for this compressor, suction and discharge check valves are recommended to prevent reverse rotation and oil logging during shutdown. Design of the compressor results in a fixed built-in volume ratio. While figure 19i shows the a rotary vane with seven blades.



a- Begibing of suction

b- Suction stroke

c- Suction contiuoes

d- Ending of suction

e- Compression starting

f- Ending of compression

g- Dischage storke

h- Contioues operating of compressor

i- Seven baldes rotar

Figure 19 how rotary eight vanes compressor works b- **CENTRIFUGAL COMPRESSORS**

Centrifugal compressors, sometimes called turbo compressors, belong to a family of turbo machines that includes(figure 20):

- fans,
- propellers,
- and turbines.

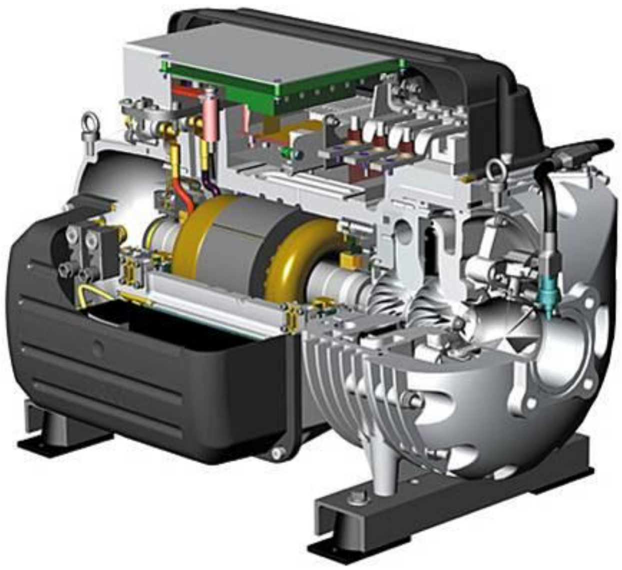
These machines continuously exchange angular momentum between a rotating mechanical element and a steadily flowing fluid. Because their flows are continuous, **turbo machines have greater volumetric capacities, size for size, than do positive displacement devices**. For effective momentum exchange, their rotating speeds must be higher, but little vibration or wear results because of the steadiness of the motion and the absence of contacting parts.

Centrifugal compressors are well suited for air-conditioning and refrigeration applications **because of their ability to produce a high pressure ratio**.

The suction flow enters the rotating element(figure 21) , or impeller, in the axial direction and is discharged radially at a higher velocity. **The change in diameter through the impeller increases the velocity of the gas flow. This dynamic pressure is then converted to static pressure**, through a diffusion process, which generally begins within the impeller and ends in a radial diffuser and scroll outboard of the impeller, as shown in figure 22.

Centrifugal compressors are used in a variety of refrigeration and air-conditioning installations. Suction flow ranges between 0.03 and 15 m^3/s, with rotational speeds between 1800 and 90 000 rpm.

However, the high angular velocity associated with a low volumetric flow establishes a minimum practical capacity for most centrifugal applications. The upper capacity limit is determined by physical size, a 15 m^3/s compressor being about 1.8 to 2.1 m in diameter. Suction temperature is usually between 10 and - 100°C, with a suction pressure between 14 and 700 kPa and discharge pressure up to 2 MPa. Pressure ratios range between 2 and 30. Almost any refrigerant can be used.



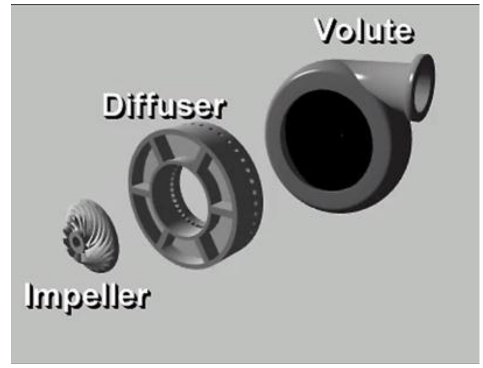
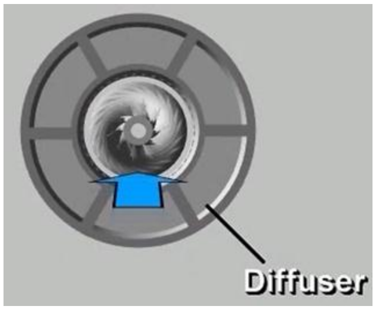
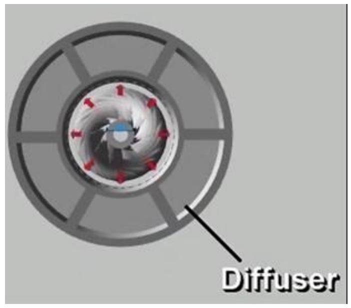


Figure 20 centrifugal compressor

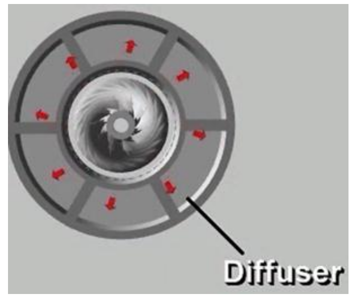
Figure 21 Centrifugal compressor impeller



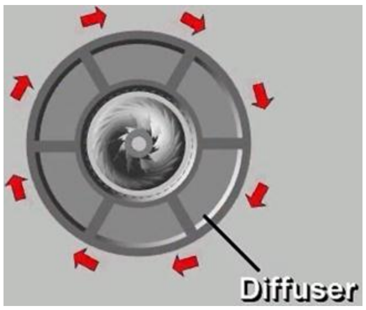
a- suction vapour



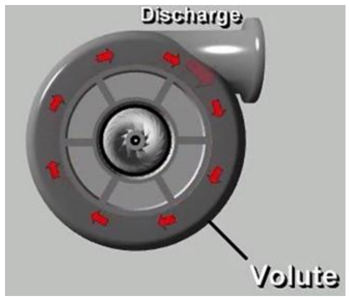
b- inlet vapour to impelle



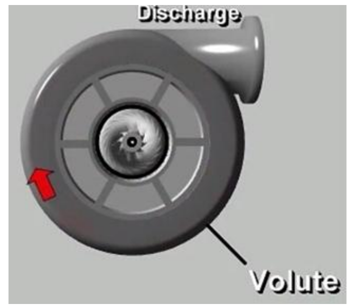
c- change direction fron axial to radial



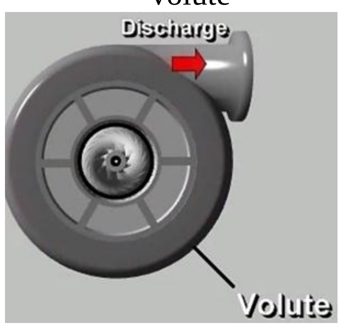
d- exit vapour from diffuser



e- enclosed vapour in volute



f- vapour flow through volute



g- discharge stroke

Figure 22 how centrifugal compressor work

Condensers and Cooling Towers

The specific objectives of this lesson are to:

1. Discuss general aspects of condensers used in refrigeration
2. Introduce refrigerant condensers
3. Classify refrigerant condensers based on the external fluid used, based on constructional details etc.
4. Compare air cooled condensers with water cooled condensers
5. Discuss briefly the effect of presence of air and other non-condensible gases in refrigerant condensers
7. Discuss briefly the concept of optimum condensing pressure for lowest running cost of a refrigeration system

At the end of the lecture, the student should be able to:

1. Classify and describe refrigerant condensers based on the external fluid used, based on the external fluid flow and based on constructional aspects
2. Compare air-cooled condensers with water-cooled condensers
3. Perform condenser design calculations using various correlations presented for estimating heat transfer coefficients on external fluid and refrigerant side and estimate the required condenser area for a given refrigeration system
4. Explain the effect of presence of non-condensable

1- INTRODUCTION

The purpose of the condenser in a vapour compression cycle is to accept the hot, high-pressure gas from the compressor and cool it to remove first the superheat and then the latent heat, so that the refrigerant will condense back to a liquid. In addition, the liquid is usually slightly sub-cooled. In nearly all cases, the cooling medium will be air or water

2- HEAT TO BE REMOVED

The heat to be removed in the condenser is shown in the p–h diagram (Figure 1) and, apart from comparatively small heat losses and gains through the circuit, will be

Heat taken in by evaporator + heat of compression.

This latter, again ignoring small heat gains and losses, will be the power input to the compressor, giving

Evaporator load + compressor input power = condenser load

Condenser load is stated as the rate of heat rejection. Some manufacturers give ratings in terms of the evaporator load, together with a ‘ de-rating ’ factor, which depends on the evaporating and condensing temperatures.

Evaporator × load-factor = condenser load

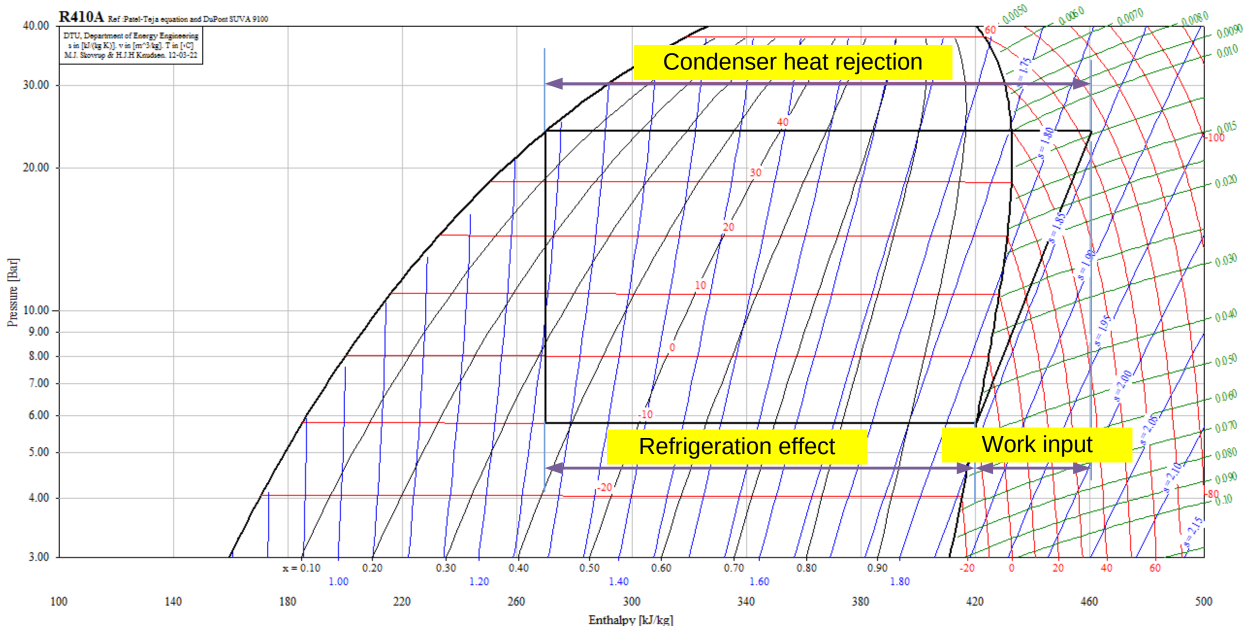


Figure 1 Condenser load on p-h diagram

The provision of a separate oil cooler will reduce condenser load by the amount of heat lost to the oil and removed in the oil cooler. This is of special note with oil-injected screw compressors, where a high proportion of the compressor energy is taken away in the oil. This proportion varies with the exact method of oil cooling, and figures should be obtained from the compressor manufacturer for a particular application.

3- Condenser classifications:
Based on the external fluid, condensers can be classified as:

1. Air cooled condensers
   a. Natural convection type
   b. Forced convection type
2. Water cooled condensers
   a. Shell and tube
   b. Shell and coil
   c. Tube in tube
3. Evaporative condensers

Figure 2 shows the types on condenser

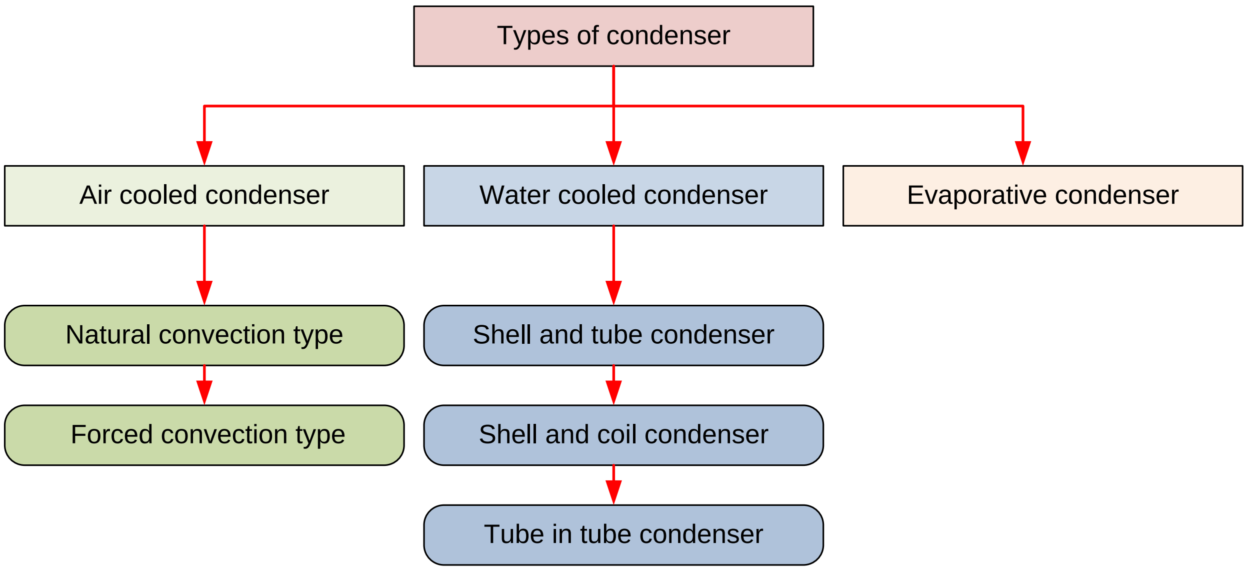


Figure 2 types on condenser

3-1 Air cooled condenser:

3-1-a Natural convection type

In natural convection type, **heat transfer from the condenser is by buoyancy induced natural convection and radiation**. Since the flow rate of air is small and the radiation heat transfer is also not very high, **the combined heat transfer coefficient in these condensers is small**. As a result a relatively **large condensing surface is required to reject a given amount of heat**. Hence these condensers are used for small capacity refrigeration systems like household refrigerators and freezers. The natural convection type condensers are either

- plate surface type or
- finned tube type.

Plate surface condenser:

In plate surface type condensers used in small refrigerators and freezers, the refrigerant carrying tubes are attached to the outer walls of the refrigerator, as shown in figure 3. The whole body of the refrigerator (except the door) acts like a fin. Insulation is provided between the outer cover that acts like fin and the inner plastic cover of the refrigerator. It is for this reason that outer body of the refrigerator is always warm. Since the surface is warm, the problem of moisture condensation on the walls of the refrigerator does not arise in these systems. These condensers are sometimes called as flat back condensers.

- **finned tube condenser:**

  The finned type condensers are mounted either below the refrigerator at an angle or on the backside of the refrigerator. In case, it is mounted below, then the warm air rises up and to assist it an air envelope is formed by providing a jacket on backside of the refrigerator. The fin spacing is kept large to minimize the effect of fouling by dust and to allow air to flow freely with little resistance. In the older designs, the condenser tube (in serpentine form) was attached to a plate and the plate was mounted on the backside of the refrigerator, as shown in figure 4. The plate acted like a fin and warm air rose up along it. In another common design, thin wires are welded to the serpentine tube coil. The wires act like fins for increased heat transfer area. Figure 5 shows the schematic of a wire-and-tube type condenser commonly used in domestic refrigerators. Regardless of the type, refrigerators employing natural convection condenser should be located in such a way that air can flow freely over the condenser surface.

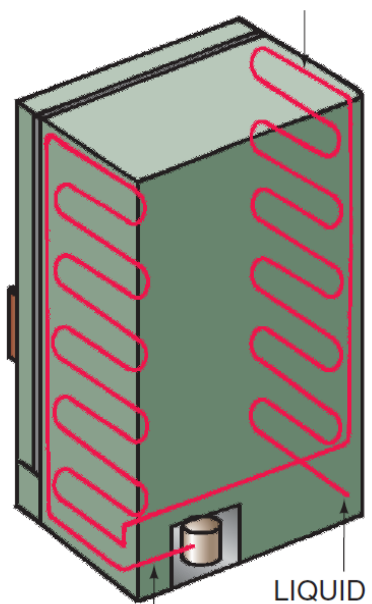


Figure 3 plate condenser

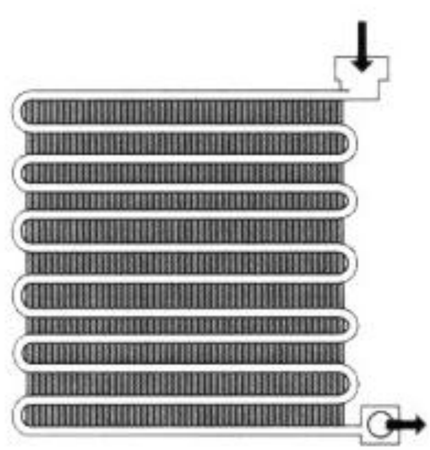


Figure 4 serpentine condenser (Old design)

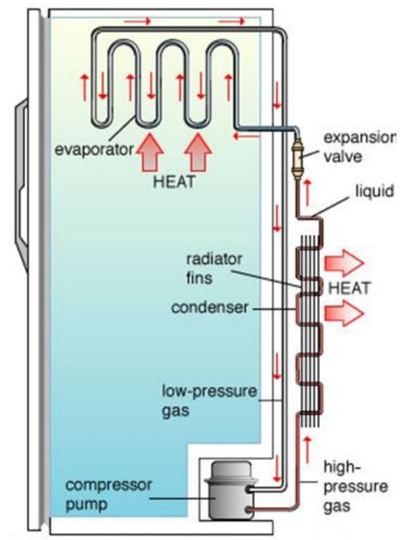


Figure 5 Finned tube condenser

b- Forced convection type

In forced convection type condensers, the circulation of air over the condenser surface is maintained by using a fan or a blower.

These condensers normally use fins on air-side for good heat transfer. The fins can be either plate type or annular type. Forced convection type condensers are commonly used in window air conditioners, water coolers and packaged air conditioning plants.

These are either chassis mounted or remote mounted. In chassis mounted type (figure 6), the compressor, induction motor, condenser with condenser fan, accumulator, HP/LP cut- out switch and pressure gauges are mounted on a single chassis. It is called condensing unit of rated capacity.

The remote mounted type, is either vertical or roof mounted horizontal type, as shown in figure 7.

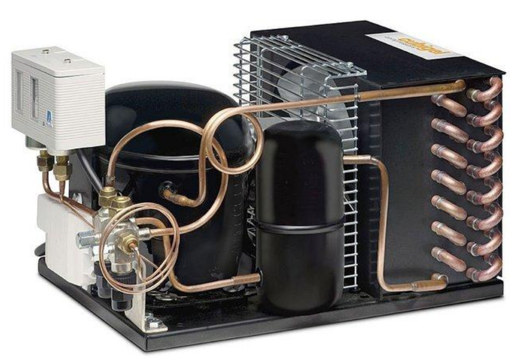


Figure 6 chassis mounted condenser

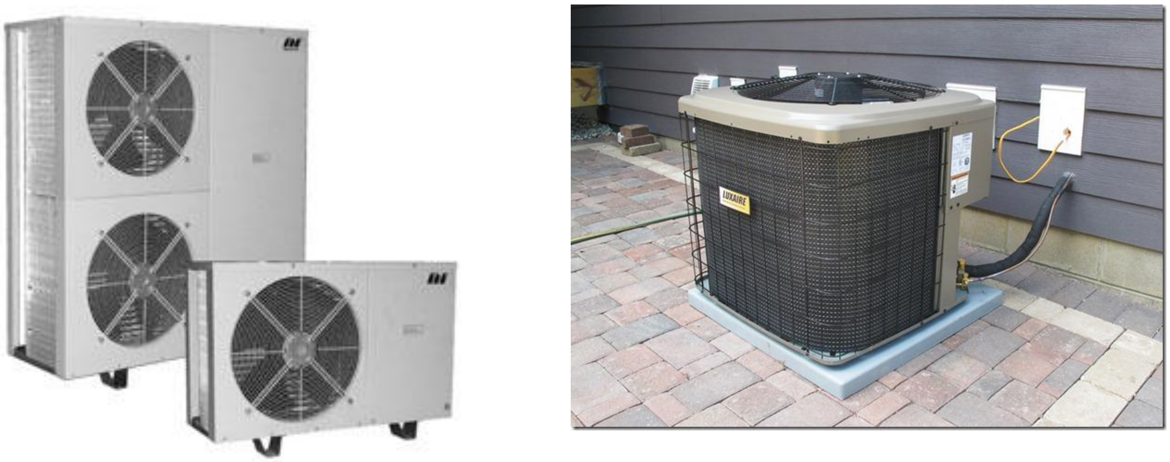


Figure 7 remote condenser

The area of the condenser seen from outside in the airflow direction is called **face area**. The velocity at the face is called **face velocity**. This is given by the volume flow rate divided by the face area. The face velocity is usually around 2 m/s to 3.5 m/s to limit the pressure drop due to frictional resistance. The coils of the tube in the flow direction are called rows. A condenser may have two to eight rows of the tubes carrying the refrigerant.

The air flows over the fins while the refrigerant flows inside the tubes. The fins are usually of aluminum and tubes are made of copper as shown in figure 8.

For ammonia condensers mild steel tubes with mild steel fins are used. In this case the fins are either welded or galvanizing is done to make a good thermal contact between fin and tube. In case of ammonia, annular crimpled spiral fins are also used over individual tubes instead of flat-plate fins as shown in figure 9. In finned tube heat exchangers the fin spacing may vary from 3 to 7 fins per cm. The secondary surface area is 10 to 30 times the bare pipe area hence; the finned coils are very compact and have smaller weigh

Figure 8 fins are usually of aluminum and tubes are made of copper

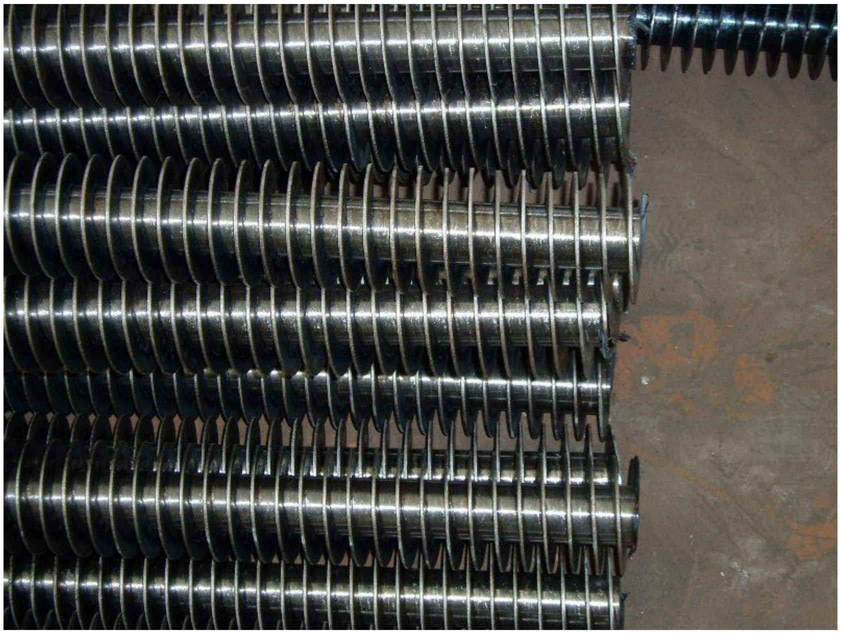


Figure 9 annular crimpled spiral fins

2- Water Cooled Condensers:

In water cooled condensers water is the external fluid. Depending upon the construction, water cooled condensers can be further classified into:

a. Shell-and-tube type
b. Shell-and-coil type
c. tube-in-tube type

a- shell and tube condenser

This is the most common type of condenser used in systems from 2 TR up-to thousands of TR capacity as shown in figure 10. In these condensers the refrigerant flows through the shell while water flows through the tubes in single to four passes as shown in figure 11. The condensed refrigerant collects at the bottom of the shell as shown in figure 12. The coldest water contacts the liquid refrigerant so that some sub-cooling can also be obtained. The liquid refrigerant is drained from the bottom to the receiver. There might be a vent connecting the receiver to the condenser for smooth drainage of liquid refrigerant. The shell also acts as a receiver. Further the refrigerant also rejects heat to the surroundings from the shell. The most common type is horizontal shell type as shown in figure 13. Vertical shell-and-tube type condensers are usually used with ammonia in large capacity systems so that cleaning of the tubes is possible from top while the plant is running

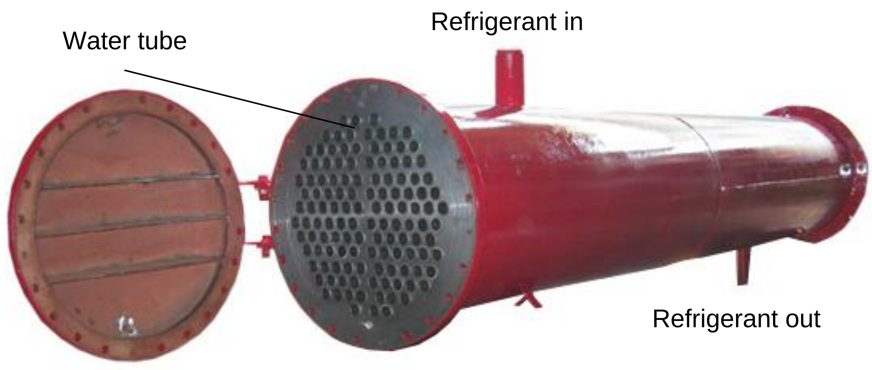


Figure 10 shell and tube condenser showing the water tube

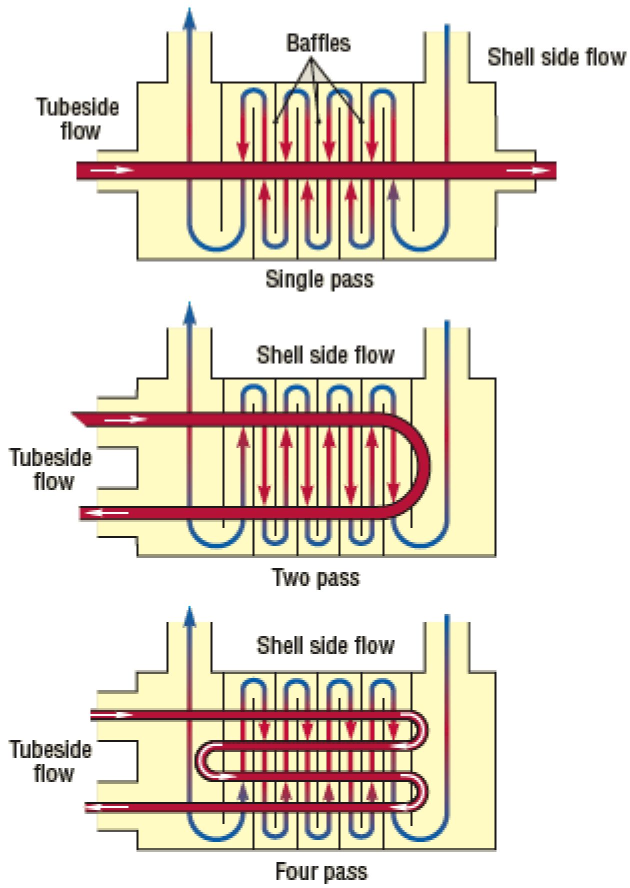


Figure 11 water flows through the tubes in single to four passes

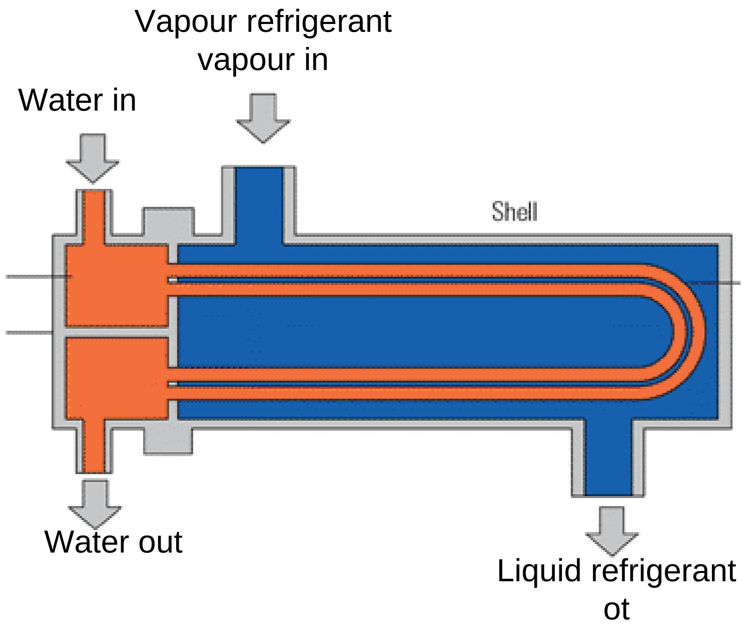


Figure 12 passage of water and refrigerant

Figure 13 vertical shell and tube condenser b- Shell-and-coil water-cooled condenser (figure 14) is simply a continuous copper coil mounted inside a steel shell. Water flows through the coil, and the refrigerant vapor from the compressor is discharged inside the shell to condense on the outside of the cold tubes. In many designs, the shell also serves as a liquid receiver. The shell-and-coil condenser has a low manufacturing cost, but this advantage is offset by the disadvantage that this type of condenser is difficult to service in the field. If a leak develops in the coil, the head from the shell must be removed and the entire coil pulled from the shell to find and repair the leak. A continuous coil is a nuisance to clean, whereas straight tubes are easy to clean with mechanical tube cleaners. In summary, with some types of cooling water, it may be difficult to maintain a high rate of heat transfer with a shell-and-coil condenser These condensers are used in systems up to 50 TR capacity

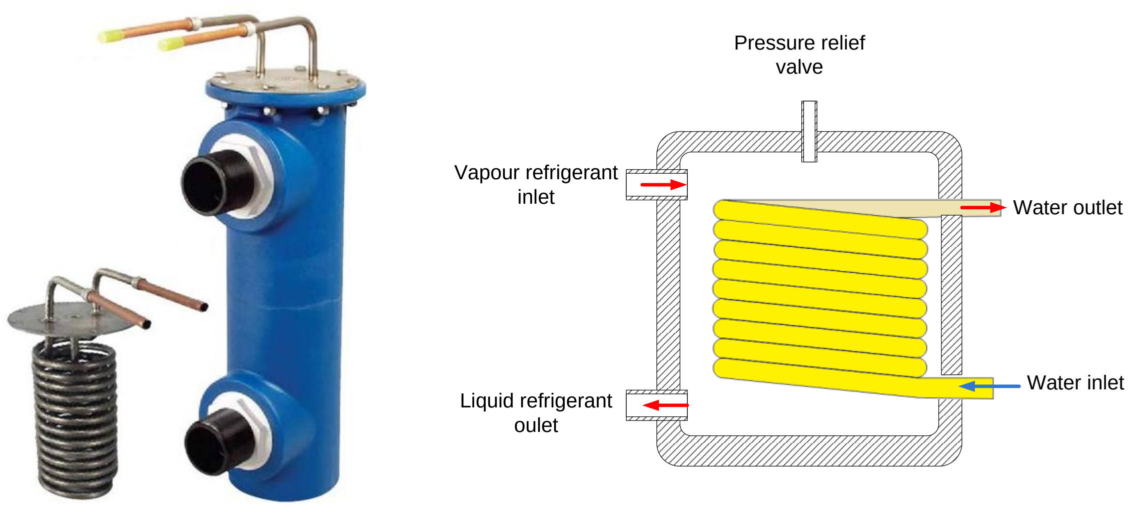


Figure 14 shell and coil condenser

c- Tube in tube condenser

The use of tube-within-a-tube for condensing purposes is popular because it is easy to make. Water passing through the inner tube along with the exterior air condenses (fig 15) the refrigerant in the outer tube. This “double cooling” improves efficiency of the condenser.

Water enters the condenser at the point where the refrigerant leaves the condenser. It leaves the condenser at the point where the hot vapor from the compressor enters the condenser. This arrangement is called counter-flow design. The rectangular type of tube-within-a-tube condenser uses a straight, hard copper pipe with manifolds on the ends. When the manifolds are removed, the water pipes can be cleaned mechanically. Figure 16 shows a condensing unit with tube in tube condenser

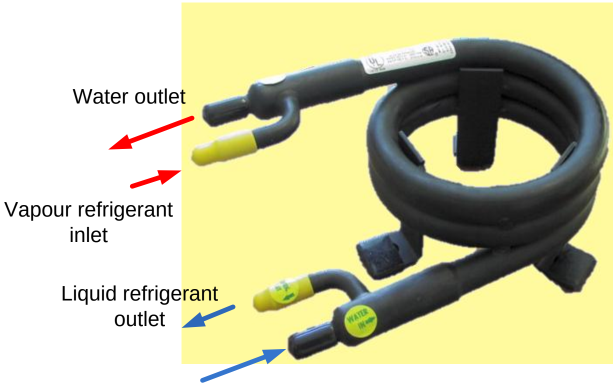


Figure 14 Tube in tube water cooled condenser

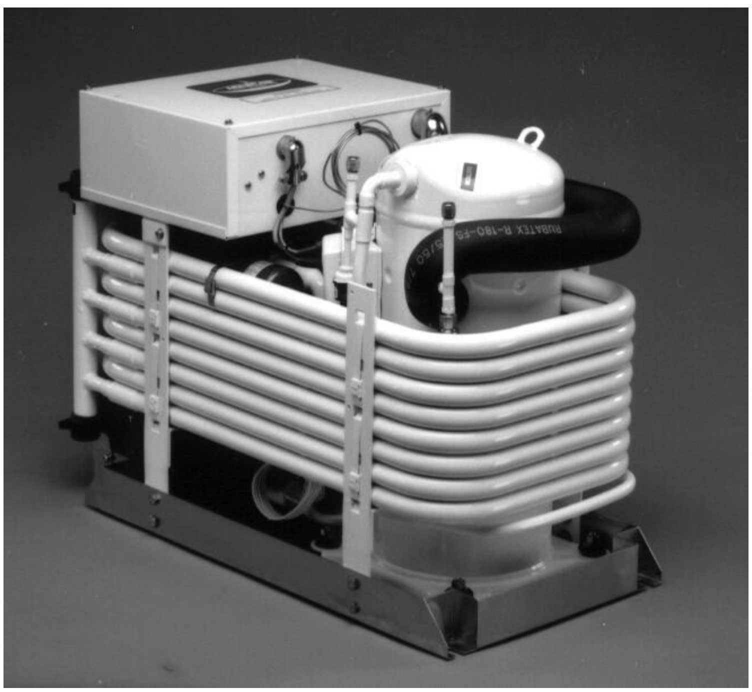


Figure 15 Condensing unit uses tube in tube condenser

3. Evaporative condensers

In evaporative condensers, both air and water are used to extract heat from the condensing refrigerant. Figure 16 shows the evaporative condenser. Evaporative condensers combine the features of a cooling tower and water-cooled condenser in a single unit. In these condensers, the water is sprayed from top part on a bank of tubes carrying the refrigerant and air is induced upwards. There is a thin water film around the condenser tubes from which evaporative cooling takes place. The heat transfer coefficient for evaporative cooling is very large. Hence, the refrigeration system can be operated at low condensing temperatures (about 11 to 13 K above the wet bulb temperature of air). The water spray countercurrent to the airflow acts as cooling tower. The role of air is primarily to increase the rate of evaporation of water.

- **The required air flow rates are in the range of 350 to 500 m^3/h per TR of refrigeration capacity.**
- **Evaporative condensers are used in medium to large capacity systems.**
- **These are normally cheaper compared to water cooled condensers, which require a separate cooling tower.**
- **Evaporative condensers are used in places where water is scarce. Since water is used in a closed loop, only a small part of the water evaporates.**
- **Make-up water is supplied to take care of the evaporative loss.**
- **Since the condenser is kept outside, to prevent the water from freezing, when outside temperatures are very low, a heater is placed in the water tank.**
- **When outside temperatures are very low it is possible to switch-off the water pump and run only the blowers, so that the condenser acts as an air cooled condenser.**

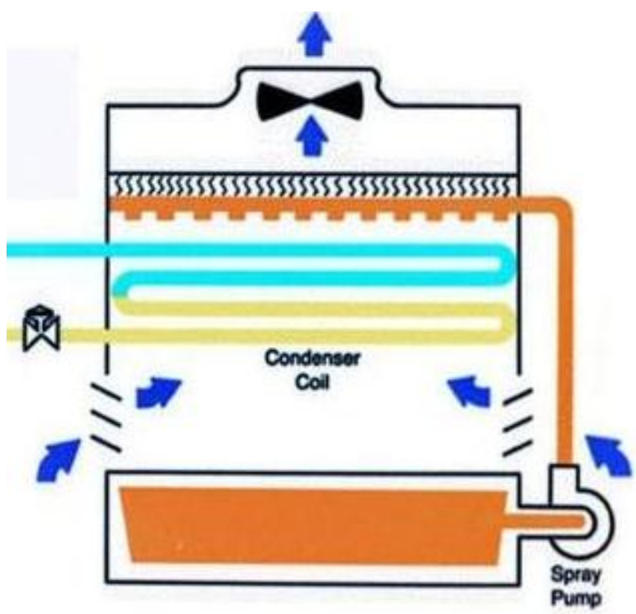


Figure 16 Evaporative condenser

Figure 17 Evaporative condenser

1-COOLING TOWERS

1-1 PRINCIPLE OF OPERATION

A cooling tower cools water by a combination of heat and mass transfer. The water to be cooled is distributed in the tower by spray nozzles, splash bars, or film-type fill, which exposes a very large water surface area to atmospheric air. Atmospheric air is circulated by

- fans,
- convective currents
- natural wind currents
- induction effect from sprays

A portion of the water absorbs heat to change from a liquid to a vapor at constant pressure. This heat of vaporization at atmospheric pressure is transferred from the water remaining in the liquid state into the air stream.

Figure 18 shows the temperature relationship between water and air as they pass through a counter flow cooling tower. The curves indicate the drop in water temperature (Point A to Point B) and the rise in the air wet-bulb temperature (Point C to Point D) in their respective passages through the tower. The temperature difference between the water entering and leaving the cooling tower (A minus B) is the range. The difference between the leaving water temperature and the entering air wet-bulb temperature (B minus C) in Figure 18 is the approach to the wet bulb or simply the approach of the cooling tower.

The approach is a function of:

- cooling tower capability,
- and a larger cooling tower produces a closer approach (colder leaving water) for a given heat load, flow rate, and entering air condition.

The thermal performance of a cooling tower depends principally on:

- the entering air wet-bulb temperature.
- *The entering air dry-bulb temperature and relative humidity, taken independently, have an insignificant effect on thermal performance of mechanical-draft cooling towers, but they do affect the rate of water evaporation within the cooling tower.*

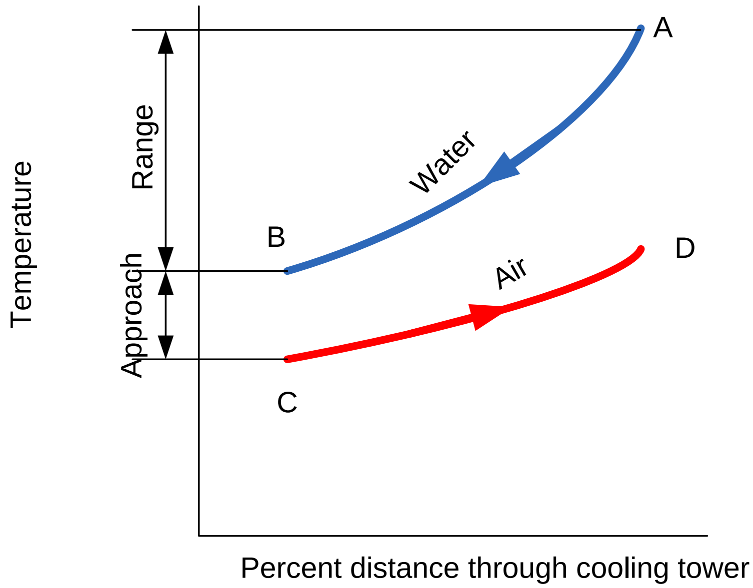


Figure 18 Temperature relationships between water and air as they pass through a counter flow cooling tower

1-2 DESIGN CONDITIONS

The thermal capability of any cooling tower may be defined by the following parameters:

1. Entering and leaving water temperatures

2. Entering air wet-bulb or entering air wet-bulb and dry-bulb temperatures

3. Water flow rate

- **The entering air dry-bulb** temperature affects the amount of water evaporated from any evaporative-type cooling tower. It also affects airflow through towers.
- **The thermal capability of a cooling tower** used for air-conditioning applications may be expressed in nominal capacity, **which is based on heat dissipation of 1.25 kW per kilowatt of evaporator cooling**.
- **Nominal cooling capacity** is defined as cooling 54 mL/s of water from 35°C to 29.4°C at a 25.6°C entering air wet-bulb temperature.
- For every kilowatt of heat picked up in the evaporator, the cooling tower must dissipate an additional 0.25 kW of compressor heat.

1-3 TYPES OF COOLING TOWERS

Two basic types of evaporative cooling devices are used

1-3-1 direct-contact or **open cooling tower** (see Figure 19), exposes water directly to the cooling atmosphere, thereby transferring the source heat load directly to the condenser
1-3-2 closed-circuit cooling tower, involves **indirect contact** between heated fluid and atmosphere (Figure 20). (evaporative condenser)

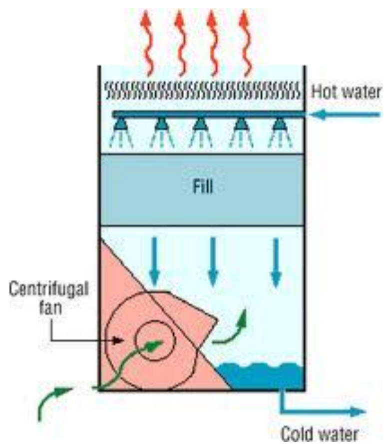


Figure 19 Direct contact cooling tow

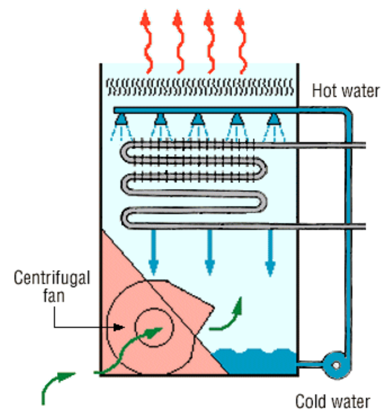


Figure 20 indirect contact cooling tow

The direct-contact devices, the most rudimentary is:

- **spray filled tower** that exposes water to the air without any heat transfer medium or fill. In this device, the amount of water surface exposed to the air depends on the efficiency of the sprays, and the time of contact depends on the elevation and pressure of the water distribution system see figure 21.
- To increase contact surfaces, as well as time of exposure, a heat transfer medium, or **fill**, is installed below the water distribution system, in the path of the air see figure 22. The two types of fill in use are splash type and film-type are.
    - **Splash-type fill** maximizes contact area and time by forcing the water to cascade through successive elevations of splash bars arranged in staggered rows see figure 23.
    - **Film-type fill** achieves the same effect by causing the water to flow in a thin layer over closely spaced sheets, principally polyvinyl chloride (PVC), that are arranged vertically see figure 24.

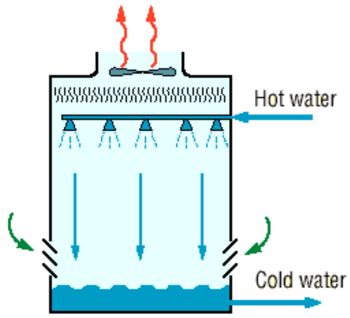


Figure 21 **spray filled tower**

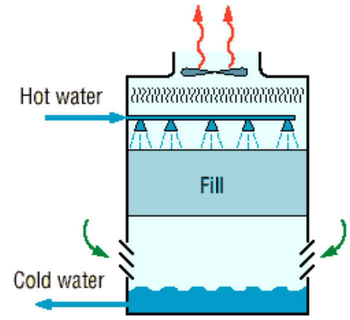


Figure 22 filled type tower

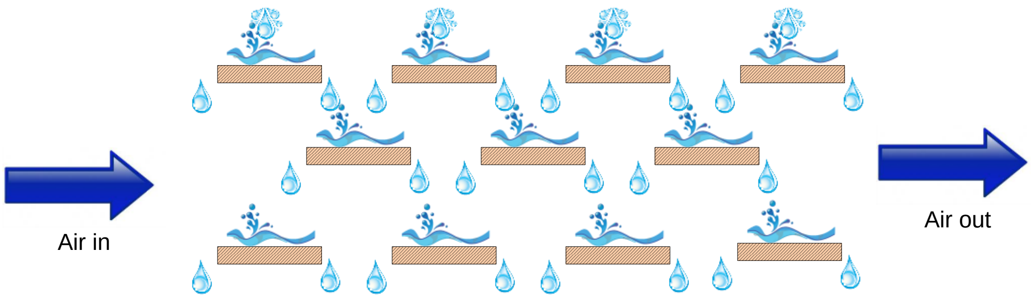


Figure 23 Splash type fill

Figure 24 film type fill

1-3-1 Types of Direct-Contact Cooling Towers

1. **Chimney (hyperbolic) towers** have been used primarily for large power installations, (Figure 25). The heat transfer mode may be counter flow, cross-flow, or parallel flow. Air is induced through the tower by the air density differentials that exist between the

lighter, heat-humidified chimney air and the outside atmosphere. Fill can be splash or film type. Materials used in chimney construction have been primarily steel-reinforced concrete; early day timber structures had limitations of size.

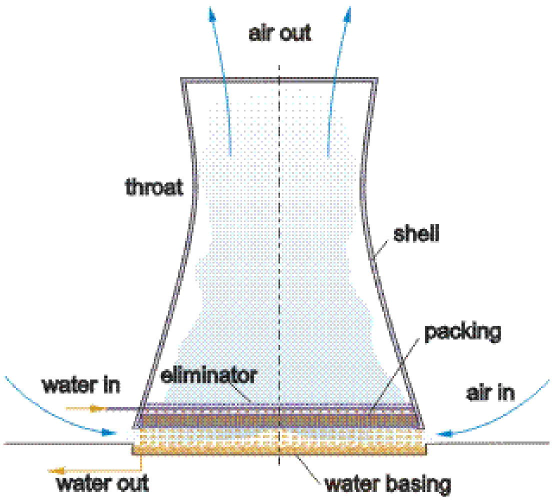
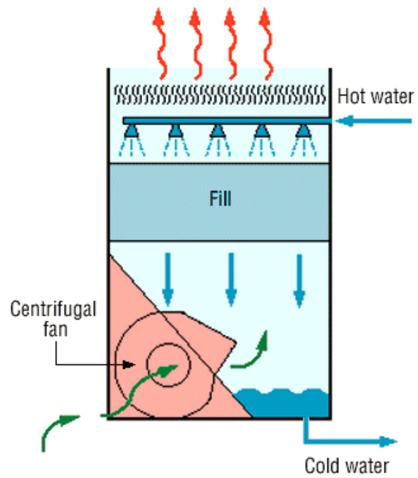


Figure 25 **Chimney (hyperbolic) towers**

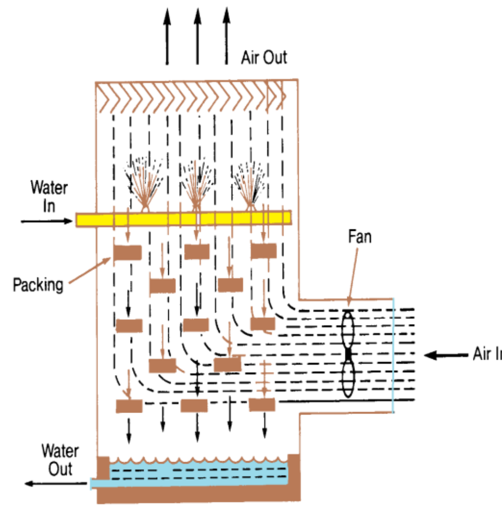
2. **Mechanical-Draft Towers.** Figure 26 shows four different designs for mechanical-draft towers (conventional towers).

- Forced – draft cooling tower: in this type the fan is on the inlet air ide, Forced draft cooling tower divide into two types:
  - Forced draft counter flow
  - Forced draft cross flow
- Induced – draft cooling tower: in this type the fan is on exit air side. Induced draft cooling towers divided into two types:
  - Induced draft counter-flow
  - Induced draft cross flow

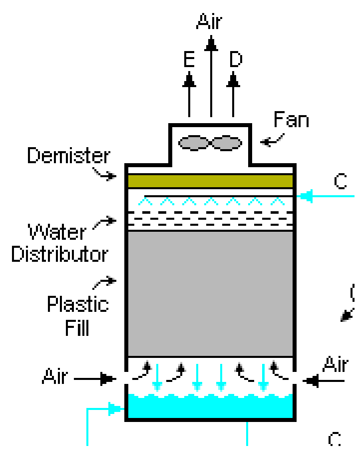
The type of fan selected, either centrifugal or propeller, depends on external pressure needs.



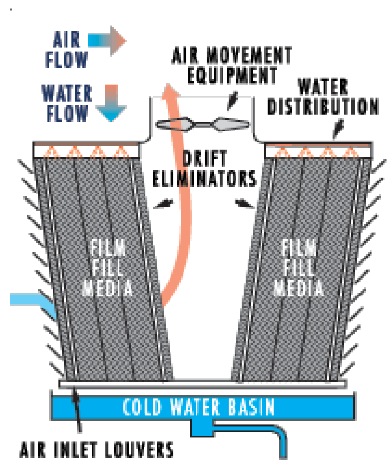
a- Counter flow – Forced draft



b-Cross flow- Forced draft



c- counter flow- Induced draft



d- cross flow - Induced draft

Figure 26 types of cooling tower

1-4 APPLICATION

Sitting

When a cooling tower can be located in an open space with free air motion and unimpeded air supply, sitting is normally not an obstacle to obtaining a satisfactory installation.

However, towers are often situated indoors, against walls, or in enclosures. In such cases, the following factors must be considered:

1. Sufficient free and unobstructed space should be provided around the unit to ensure an adequate air supply to the fans and to allow proper servicing.
2. Tower discharge air should not be deflected in any way that might promote recirculation [a portion of the warm, moist discharge air reentering the tower] . Recirculation raises the entering wet-bulb temperature, causing increased hot water and cold water temperatures, and, during cold weather operation, can promote the icing of air intake areas

Evaporators:

1- Introduction:

The purpose of the evaporator is to receive low-pressure, low-temperature refrigerant from the expansion valve and to bring it in close thermal contact with the load.

The refrigerant takes up its latent heat from the load and leaves the evaporator as a dry gas.

The function of the evaporator is to cool air or liquid in almost all cases. The air or liquid then cools the load. For example in a refrigerated display cabinet, the air is cooled and circulated to keep the contents at the required temperature; in a water chiller system, the water is circulated to individual fan-coil units to provide air conditioning. In heat pumps, the function can be described as recovering heat from air or liquid, but the evaporator construction will be very similar.

2- Classifications:

There are several ways of classifying the evaporators depending upon the:

2-1 heat transfer process

2-2 refrigerant flow

2-3 or condition of heat transfer surface.

2-1 Classifications according to heat transfer process:

The evaporator may be classified as natural convection type or forced convection type.

a- Forced convection type:

In Forced convection type a fan or a pump is used to circulate the fluid being refrigerated and make it flow over the heat transfer surface, which is cooled by evaporation of refrigerant, as shown in figure 1.

b- Natural convection type:

In natural convection type the fluid being cooled flows due to natural convection currents arising out of density difference caused by temperature difference. The refrigerant boils inside tubes and evaporator is located at the top. The temperature of fluid, which is cooled by it, decreases and its density increases. It moves downwards due to its higher density and the warm fluid rises up to replace it, as shown in figure 2.

Figure 1 forced convection type

Figure 2 Natural convection type

2-2 Classifications according to refrigerant flow:

The heat transfer phenomenon during boiling inside and outside tubes is very different; hence, evaporators are classified as those with flow inside and outside tubes.

a- The refrigerant is confined and boils inside the tubes

In natural convection type evaporators and some other evaporators, the refrigerant is confined and boils inside the tubes while the fluid being refrigerated flows over the tubes. The direct expansion coil where the air is directly cooled in contact with the tubes cooled by refrigerant boiling inside is an example of forced convection type of evaporator where refrigerant is confined inside the tubes, figure 3.

b- The refrigerant is kept in a shell

In many forced convection type evaporators, the refrigerant is kept in a shell and the fluid being chilled is carried in tubes, which are immersed in refrigerant. Shell and tube type brine and water chillers are mainly of this kind, figure 4.

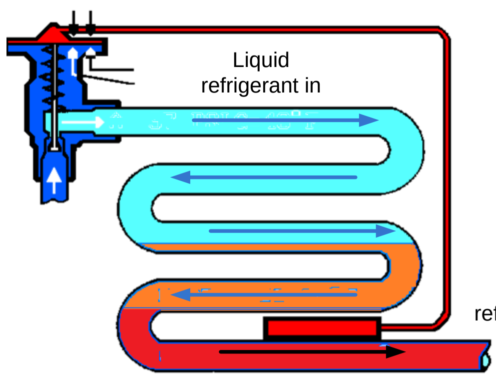


Figure 3 The refrigerant is confined and boils inside the tubes

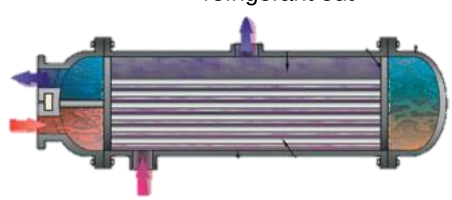


Figure 4 The refrigerant is kept in a shell

2-3. Flooded and Dry Type

The third classification is flooded type and dry type.

a- Flooded type: evaporator is said to be flooded type if liquid refrigerant covers the entire heat transfer surface. This type of evaporator uses a float type of expansion valve.

b- Dry type: an evaporator is called dry type when a portion of the evaporator is used for superheating the refrigerant vapour after its evaporation.

2-1-b Natural Convection type evaporator coils:

- Works:

These are mainly used in:

1. domestic refrigerators
2. and cold storages.

When used in cold storages, long lengths of bare or finned pipes are mounted near the ceiling or along the high sidewalls of the cold storages.

The high-density air flows downwards through the product in the cold storage. The air becomes warm by the time it reaches the floor as heat is transferred from the product to air. Some free area like a passage is provided for warm air to rise up. The same passage is used for loading and unloading the product into the cold storage.

- advantages

The advantages of such natural convection coils are that:

1. The coil takes no floor space and it also requires low maintenance cost.
2. It can operate for long periods without defrosting the ice formed on it and it does not require special skill to fabricate it.
3. Defrosting can be done easily (e.g. by scraping) even when the plant is running.

- Disadvantage

Disadvantage is that

1. Natural convection heat transfer coefficient is very small hence very long lengths are required which may cause excessive refrigerant side pressure drops unless parallel paths are used.
2. The large length requires a larger quantity of refrigerant than the forced convection coils.
3. The large quantity of refrigerant increases the time required for defrosting, since before the defrosting can start all the liquid refrigerant has to be pumped out of the evaporator tubes.
4. The pressure balancing also takes long time if the system trips or is to be restarted after load shedding.

2-4 Shell-and-Tube Liquid Chillers

The shell-and-tube type evaporators are very efficient and require minimum floor space and headspace. These are easy to maintain, hence they are very widely used in medium to large capacity refrigeration systems.

The shell-and-tube evaporators can be either

- Dry type : In dry expansion type, the refrigerant flows through the tubes while in flooded type the refrigerant is in the shell.
- or Flooded type.

Figure 5 shows a flooded type of shell and tube type liquid chiller where the liquid (usually brine or water) to be chilled flows through the tubes in double pass just like that in shell and tube condenser. The refrigerant is fed through a float valve, which maintains a constant level of liquid refrigerant in the shell. The shell is not filled entirely with tubes. This is done to maintain liquid refrigerant level below the top of the shell so that liquid droplets settle down due to gravity and are not carried by the vapour leaving the shell.

If the shell is completely filled with tubes, then a surge drum is provided after the evaporator to collect the liquid refrigerant Shell-and-tube evaporators can be either single pass type or multi-pass type shown in figure 6 . In multi-pass type, the chilled liquid changes direction in the heads. Shell-and-tube evaporators are available in vertical design also. Compared to horizontal type, vertical shell-and-tube type evaporators require less floor area. The chilled water enters from the top and flows downwards due to gravity and is then taken to a pump, which circulates it to the refrigeration load. At the inlet to tubes at the top a special arrangement introduces swirling action to increase the heat transfer coefficient.

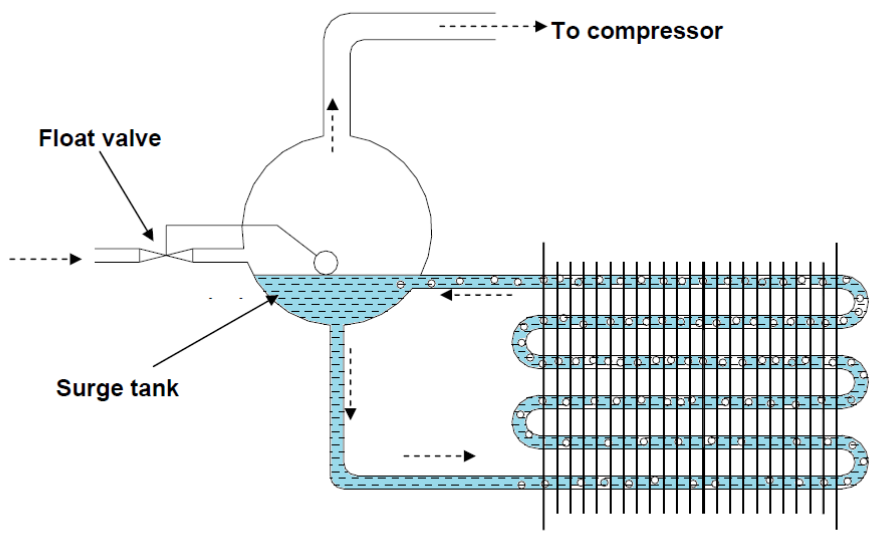


Figure 5 flooded evaporator

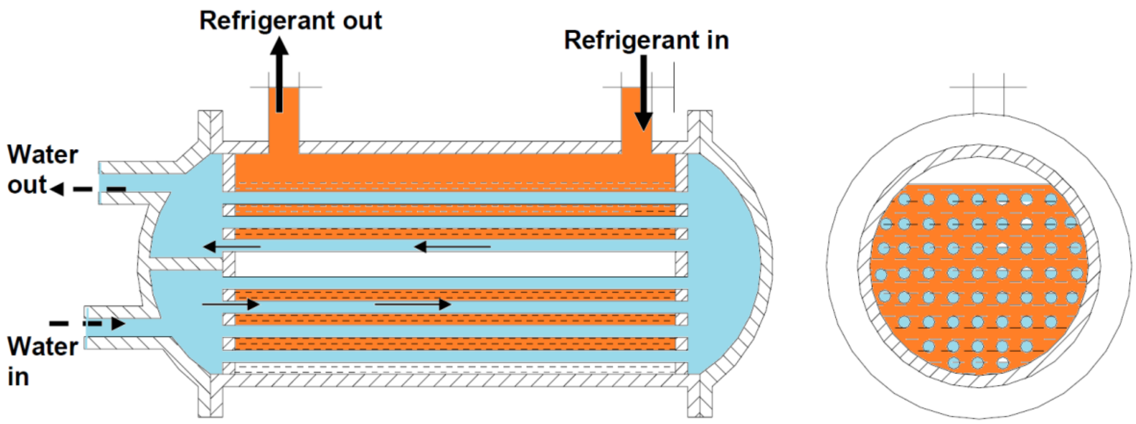


Figure 6 shell and tube flooded evaporator

Figure 7 shows a liquid chiller with refrigerant flowing through the tubes and water flowing through the shell. A thermostatic expansion valve feeds the refrigerant into the tubes through the cover on the left. The liquid to be chilled flows through the shell around the baffles. Baffles prevents the short-circuiting of the fluid flowing in the shell. This evaporator is of dry type since some of the tubes superheat the vapour. Some times more than one circuit is also provided. Changing the heads can change the number of passes. It depends upon the chiller load and the refrigerant velocity to be maintained in the heat exchanger.

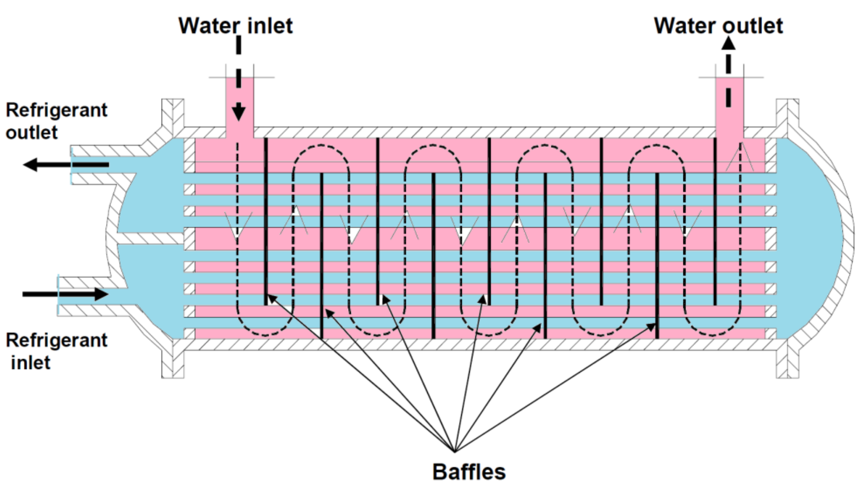


Figure 7 direct expansion type, Shell-and-Tube evaporator

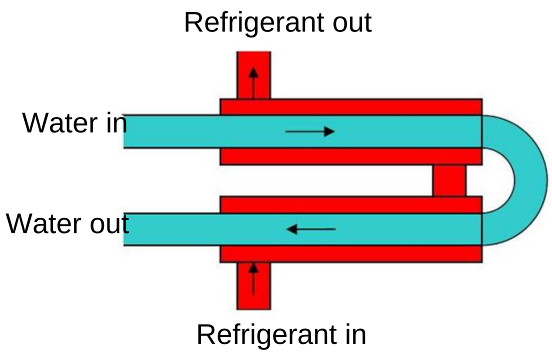
2-7 Shell-and-Coil type evaporator

These are of smaller capacity than the shell and tube chillers. These are made of one or more spiral shaped bare tube coils enclosed in a welded steel shell. It is usually dry-expansion type with the refrigerant flowing in the tube and chilled liquid in the shell. The water in the shell gives a large amount of thermal storage capacity called *hold-up capacity*. This type is good for small but highly infrequent peak loads. It is used for cooling drinking water in stainless steel tanks to maintain sanitary conditions. It is also used in bakeries and photographic laboratories. The flooded types are not recommended for any application where the temperature of chilled liquid may be below 3 ^{o}C. figure 8 shows the shell and coil evaporator.

2-8 Double pipe type evaporator

This consists of two concentric tubes, the refrigerant flows through the annular passage while the liquid being chilled flows through the inner tube in counter flow. The refrigerant side is welded hence there is minimum possibility of leakage of refrigerant. These may be used in flooded as well as dry mode. This requires more space than other designs. Shorter tubes and counter flow gives good heat transfer coefficient

Figure 8 shell and coil evaporator.



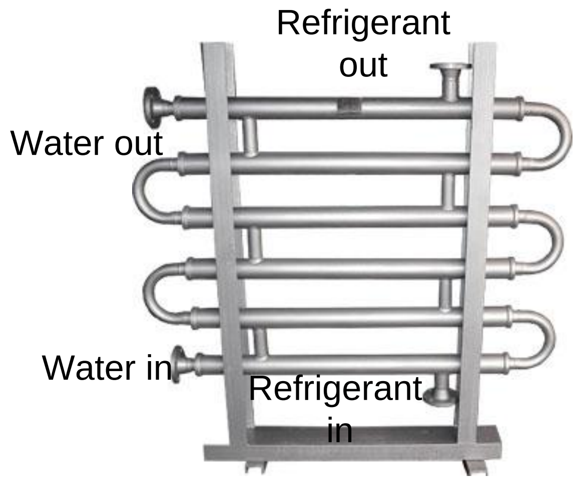


Figure 9 Tube in tube evaporator

2-9 Direct expansion fin-and-tube type

These evaporators are used for cooling and dehumidifying the air directly by the refrigerant flowing in the tubes. Similar to fin-and-tube type condensers, these evaporator consists of coils placed in a number of rows with fins mounted on it to increase the heat transfer area. Various fin arrangements are used. Tubes with individual spiral straight fins or crimpled fins welded to it are used in some applications like ammonia. Plate fins accommodating a number of rows are used in air conditioning applications with ammonia as well as synthetic refrigerants such as fluorocarbon based refrigerants.

The liquid refrigerant enters from top through a thermostatic expansion valve as shown in Fig. 10. This arrangement makes the oil return to compressor better rather than feeding refrigerant from the bottom of the coil.

Figure 10 Direct expansion fin-and-tube type

Expansion Devices

1- General

The purpose of the expansion valve is to control the flow of refrigerant from the high-pressure condensing side of the system into the low-pressure evaporator. In most cases, the pressure reduction is achieved through a variable flow orifice, either modulating or two-position. Expansion valves may be classified according to the method of control.

2- Capillary Tube

A capillary tube is a long, narrow tube of constant diameter. The word “capillary” is a misnomer since surface tension is not important in refrigeration application of capillary tubes. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1.0 m to 6 m, as shown in figure 1. The pressure reduction in a capillary tube occurs due to the following two factors:

a- The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop, and

b- The liquid refrigerant flashes (evaporates) into mixture of liquid and vapour as its pressure reduces. The density of vapour is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases. The increase in velocity or acceleration of the refrigerant also requires pressure drop.

2-1 Selection of Capillary Tube

For any new system, the diameter and the length of capillary tube have to be selected by the designer such that the compressor and the capillary tube achieve the balanced point at the desired evaporator temperature. There are :

1. analytical and
2. graphical methods to select the capillary tube.

The fine-tuning of the length is finally done by *cut-and-try* method. A tube longer than the design (calculated) value is installed with the expected result that evaporating temperature will be lower than expected. The tube is shortened until the desired balance point is achieved. This is done for mass production. If a single system is to be designed then tube of slightly shorter length than the design length is chosen. The tube will usually result in higher temperature than the design value. The tube is pinched at a few spots to obtain the required pressure and temperature.

3- Short-tube restrictors

Short-tube restrictors are widely used in residential air conditioners and heat pumps. They offer low cost, high reliability, ease of inspection and replacement, and potential elimination of check valves in the design of a heat pump. Because of their pressure equalizing characteristics, short-tube restrictors allow the use of a low-starting-torque compressor motor. Short-tube restrictors, as used in residential systems, are typically 10 to 13 mm in length, with a length-to-diameter (L/D) ratio greater than 3 and less than 20. Short-tube restrictors are also called plug orifices or orifices, although the latter is reserved for restrictors with an L/D ratio less than 3. Capillary tubes have an L/D ratio much greater than 20. An orifice tube, that shown in figures 2 and 3, a type of short-tube restrictor, is commonly used in automotive air conditioners. Most automotive applications use orifice tubes with L/D ratios between 21 and 35 and inside diameters from 1 to 2 mm. An orifice tube allows the evaporator to operate in a flooded condition, which improves performance. To prevent liquid from flooding the compressor, an accumulator/dehydrator is installed to separate liquid from vapor and to meter a small amount of lubricant-rich refrigerant to the compressor. However, the accumulator/dehydrator does cause a pressure drop penalty on the suction side. While table 1 shows the color and size of orifice tube.

Figure 1 the capillary tube

Figure 2 orifice tube

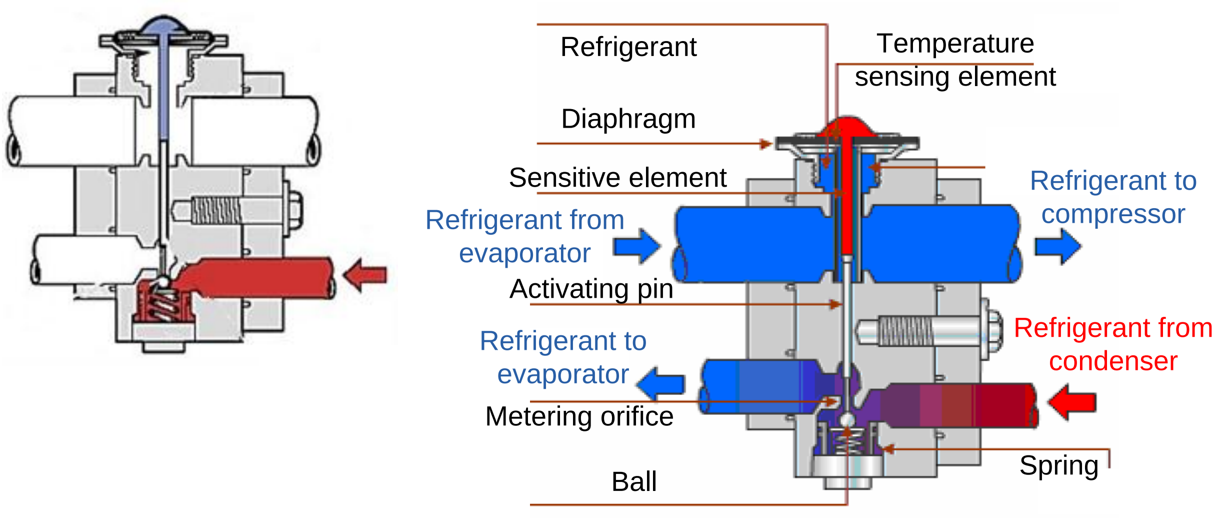
Figure 3 orifice tube details

Table 1 Color and size of orifice tube

FOT Color	**Orifice Size**
Blue	.067"
Red	.062"
Orange	.057"
Green	.052"
Brown	.047"

4- Expansion block:

The expansion block is an expansion device; it is use in car air conditioner. The block valve differs from the previously mentioned expansion valve in that it has four passages, although the basic operation is exactly the same. Figure 4 show the expansion block operation, it can be seen from the figure that when the expansion valve is closed it insulated the evaporator from the circuit. Thus block valve controls refrigerant flow using system opposing pressure. Figure 5 shows the location of expansion block.



Expansion block - closed Expansion block- open

Figure 4 expansion block operation

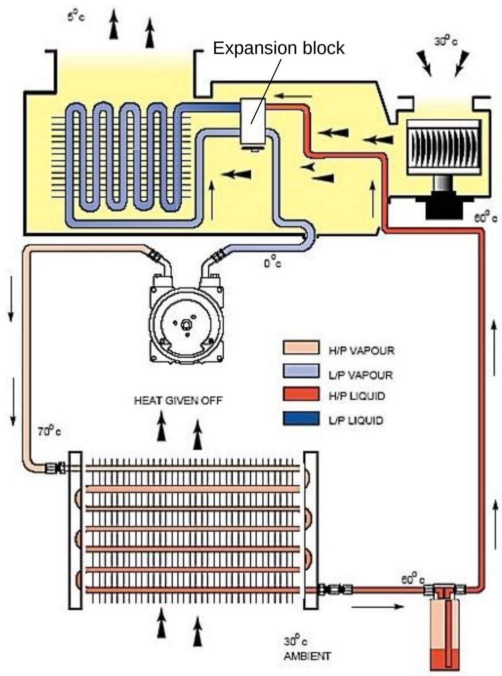
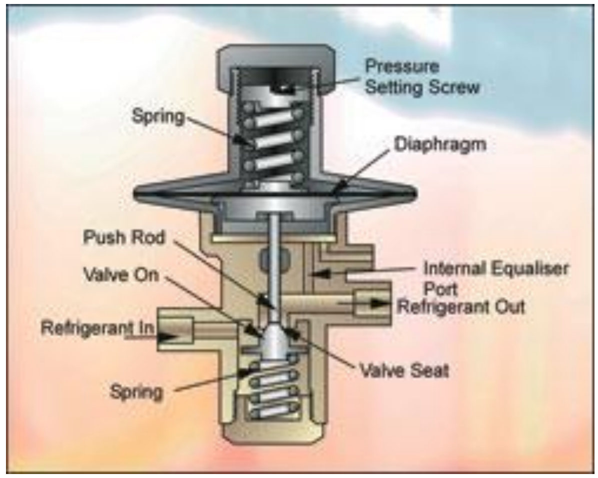


Figure 5 location of expansion block

5- Automatic Expansion Valve

An Automatic Expansion Valve (AEV) also known as a constant pressure expansion valve acts in such a manner so as to maintain a constant pressure and thereby a constant temperature in the evaporator. The automatic expansion valves are used wherever constant temperature is required, for example, milk chilling units and water coolers where freezing is disastrous. In air-conditioning systems it is used when humidity control is by DX coil temperature. Automatic expansion valves are simple in design and are economical. These are also used in home freezers and small commercial refrigeration systems where hermetic compressors are used. Normally the usage is limited to systems of less than 10 TR capacities with critical charge. Critical charge has to be used since the system using AEV is prone to flooding. Hence, no receivers are used in these systems. In some valves a diaphragm is used in place of bellows. Figure 6 shows the automatic expansion valve.



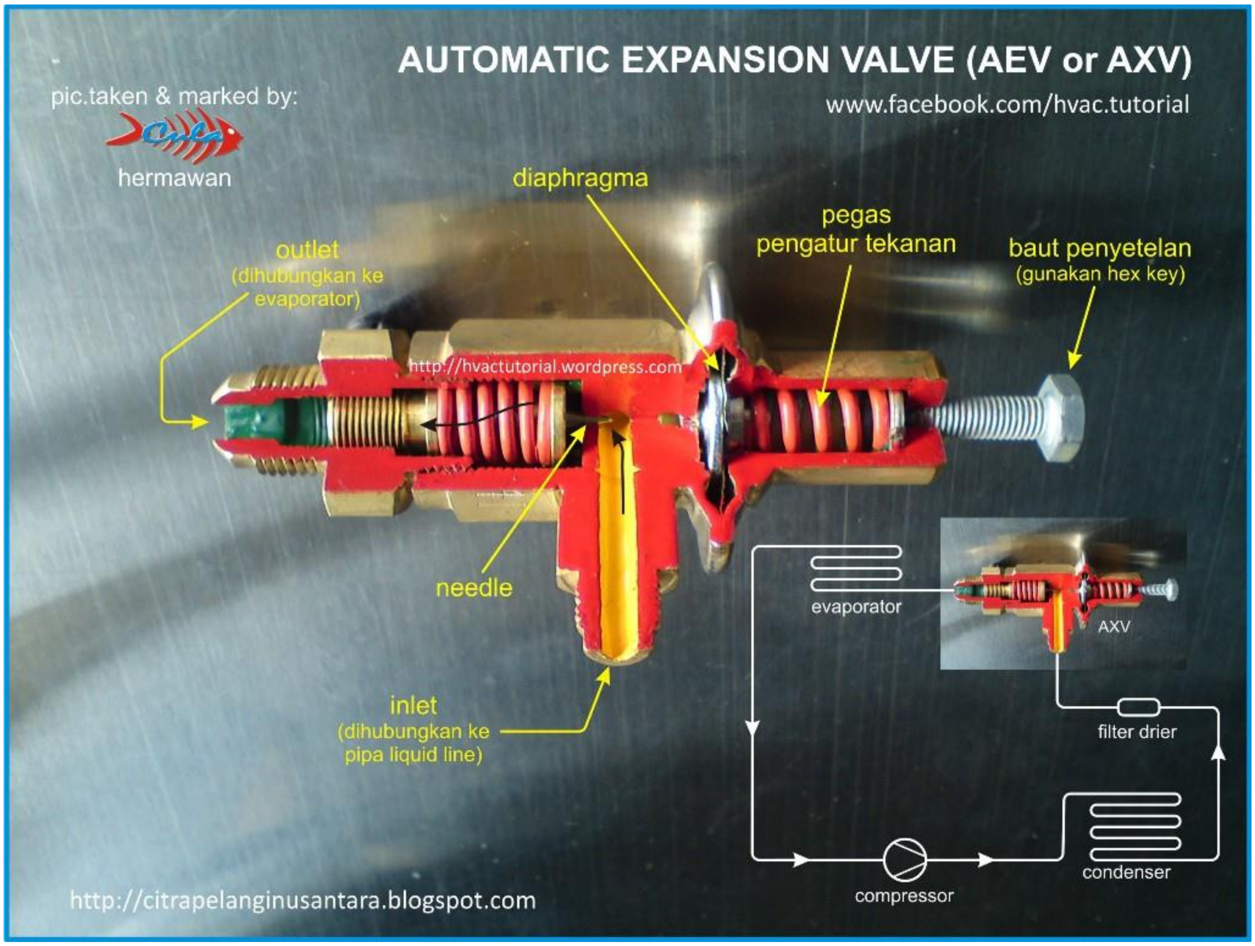


Figure 6 automatic expansion valve

6- Thermostatic expansion valve

The thermostatic expansion valve controls the flow of liquid refrigerant entering the evaporator in response to the superheat of gas leaving the evaporator. It keeps the evaporator active without allowing liquid to return through the suction line to the compressor. This is done by controlling the mass flow of refrigerant entering the evaporator so it equals the rate at which it can be completely vaporized in the evaporator by heat absorption. Because this valve is operated by superheat and responds to changes in superheat, a portion of the evaporator must be used to superheat refrigerant gas. Unlike the constant-pressure valve, the thermostatic expansion valve is not limited to constant-load applications. It is used for controlling refrigerant flow to all types of direct-expansion evaporators in air-conditioning and in commercial (medium-temperature), low temperature, and ultralow-temperature refrigeration applications. The factory superheat setting (static superheat setting) of thermostatic expansion valves is made when the valve starts to open.

Valve manufacturers establish capacity ratings on the basis of opening superheat, typically from 2 to 4 K, depending on valve design, size, and application. Figure 7 shows a cross sectional in thermostatic expansion valve.

The simple *thermostatic expansion valve* relies on the pressure under the diaphragm being approximately the same as that at the coil outlet, and small coil pressure drops can be accommodated by adjustments to the spring setting. Figure 8 shows thermostatic expansion valve.

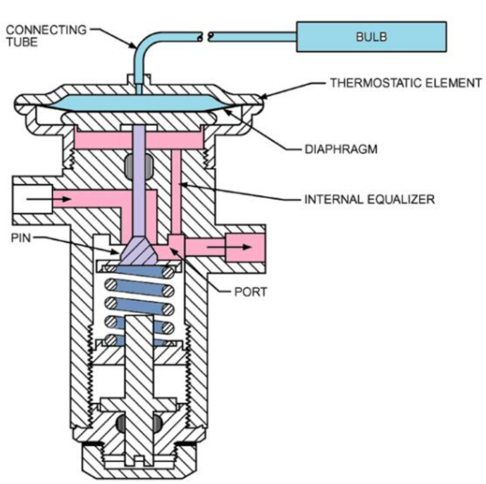


Figure 7 thermostatic expansion valve

Figure 8 thermostatic expansion valve with internal equalizer

Where an evaporator coil is divided into a number of parallel passes, a distribution device with a small pressure loss is used to ensure equal flow through each pass. Pressure drops of 1-2 bar are common. There will now be a much larger finite difference between the pressure under the diaphragm and that at the coil inlet. To correct for this, the body of the valve is modified to accommodate a middle chamber and an *equalizing connection* which is taken to the coil outlet, close to the phial position. Most thermostatic expansion valves will have provision for an external equalizer connection, figure 9 shows thermostatic expansion valve with external equalizer.

Figure 9 shows thermostatic expansion valve with external equalizer

7- Float valve:

Float type expansion valves are normally used with flooded evaporators in large capacity refrigeration systems. A float type valve opens or closes depending upon the liquid level as sensed by a buoyant member, called as float. The float could take the form of a hollow metal or plastic ball, a hollow cylinder or a pan. Thus the float valve always maintains a constant liquid level in a chamber called as float chamber. Depending upon the location of the float chamber, a float type expansion valve can be either a low-side float valve or a high-side float valve.

7-1 Low-side float valves:

A low-side float valve maintains a constant liquid level in a flooded evaporator or a float chamber attached to the evaporator. When the load on the system increases, more amount of refrigerant evaporates from the evaporator. As a result, the refrigerant liquid level in the evaporator or the low-side float chamber drops momentarily. The float then moves in such a way that the valve opening is increased and more amount of refrigerant flows into the evaporator to take care of the increased load and the liquid level is restored. Figure 10 shows a low side float valve.

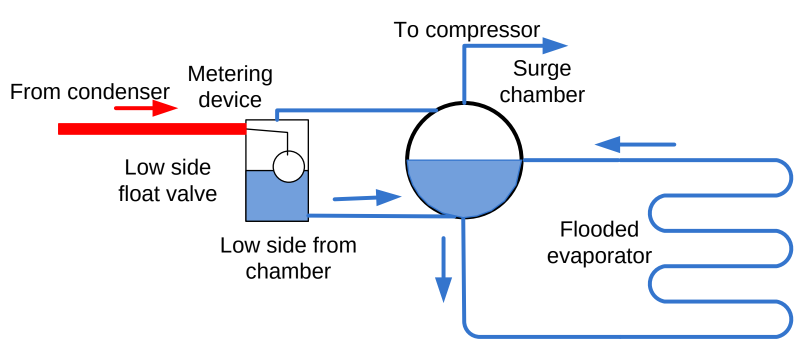
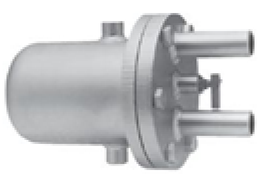


Figure 10 low side float valve.

7-2 High-side float valves:

Figure 11 shows the schematic of a high-side float valve. As shown in the figure, a high-side float valve maintains the liquid level constant in a float chamber that is connected to the condenser on the high pressure side. When the load increases, more amount of refrigerant evaporates and condenses. As a result, the liquid level in the float chamber rises momentarily. The float then opens the valve more to allow a higher amount of refrigerant flow to cater to the increased load, as a result the liquid level drops back to the original level. The reverse happens when the load drops. Since a high-side float valve allows only a fixed amount of refrigerant on the high pressure side, the bulk of the refrigerant is stored in the low-pressure side (evaporator). Hence there is a possibility of flooding of evaporator followed by compressor slugging. However, unlike low side float valves, a high-side float valve can be used with both flooded as well as direct expansion type evaporators.



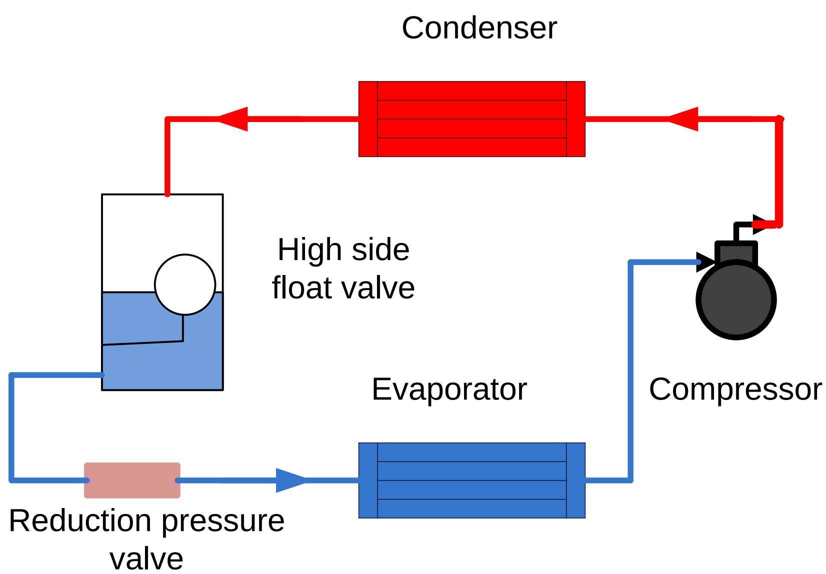


Figure 11 high-side float valve

8- Electronic Expansion Valve

The schematic diagram of an electric expansion valve is shown in 12. As shown in the figure, an electronic expansion valve consists of an orifice and a needle in front it. The needle moves up and down in response to magnitude of current in the heating element. A small resistance allows more current to flow through the heater of the expansion valve, as a result the valve opens wider. A small negative coefficient thermistor is used if superheat control is desired.

The thermistor is placed in series with the heater of the expansion valve. The heater current depends upon the thermistor resistance that depends upon the refrigerant condition. Exposure of thermistor to superheated vapour permits thermistor to self heat thereby lowering its resistance and increasing the heater current. This opens the valve wider and increases the mass flow rate of refrigerant. This process continues until the vapour becomes saturated and some liquid refrigerant droplets appear. The liquid refrigerant will cool the thermistor and increase its resistance. Hence in presence of liquid droplets the thermistor offers a large resistance, which allows a small current to flow through the heater making the valve opening narrower. Figure 13 shows the electronic expansion valve.

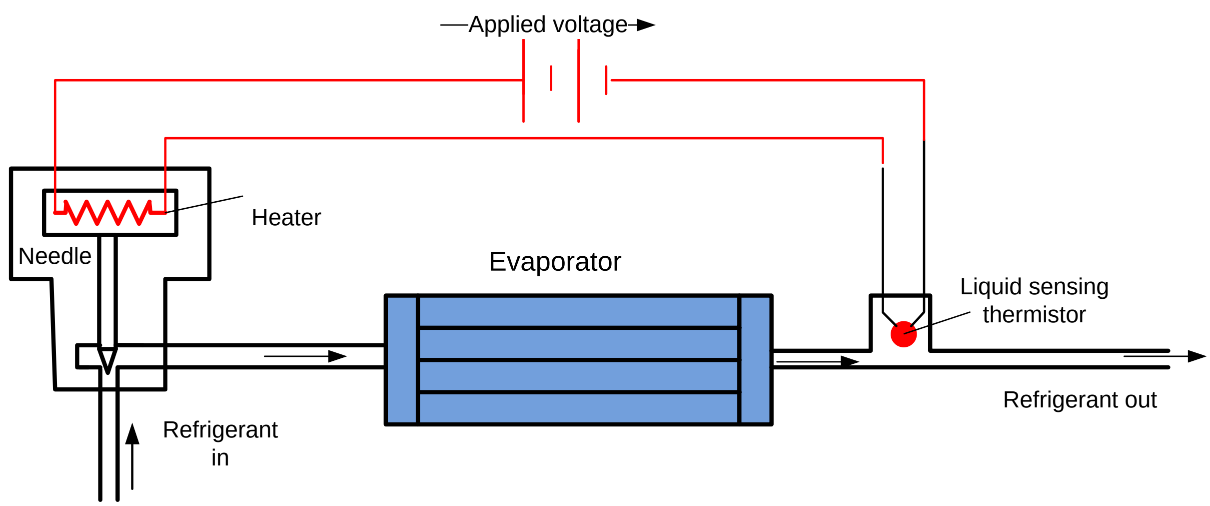


Figure 12 schematic diagram of an electric expansion valve

Figure 13 electric expansion valve

1. Which of the following statements are TRUE?
   a) A capillary tube is a variable opening area type expansion device
   b) In a capillary tube pressure drop takes place due to fluid friction
   c) In a capillary tube pressure drop takes place due to fluid acceleration
   d) In a capillary tube pressure drop takes place due to fluid friction and acceleration

Ans.: d)

2. Which of the following statements are TRUE?
   a) An automatic expansion valve maintains a constant pressure in the condenser
   b) An automatic expansion valve maintains a constant pressure in the evaporator
   c) In an automatic expansion valve, the mass flow rate of refrigerant increases as the refrigeration load increases
   d) Automatic expansion valve based systems are critically charged

Ans.: b) and d)

3. A thermostatic expansion valve:
   a) Maintains constant evaporator temperature
   b) Maintains a constant degree of superheat
   c) Increases the mass flow rate of refrigerant as the refrigeration load increases
   d) Prevents slugging of compressor

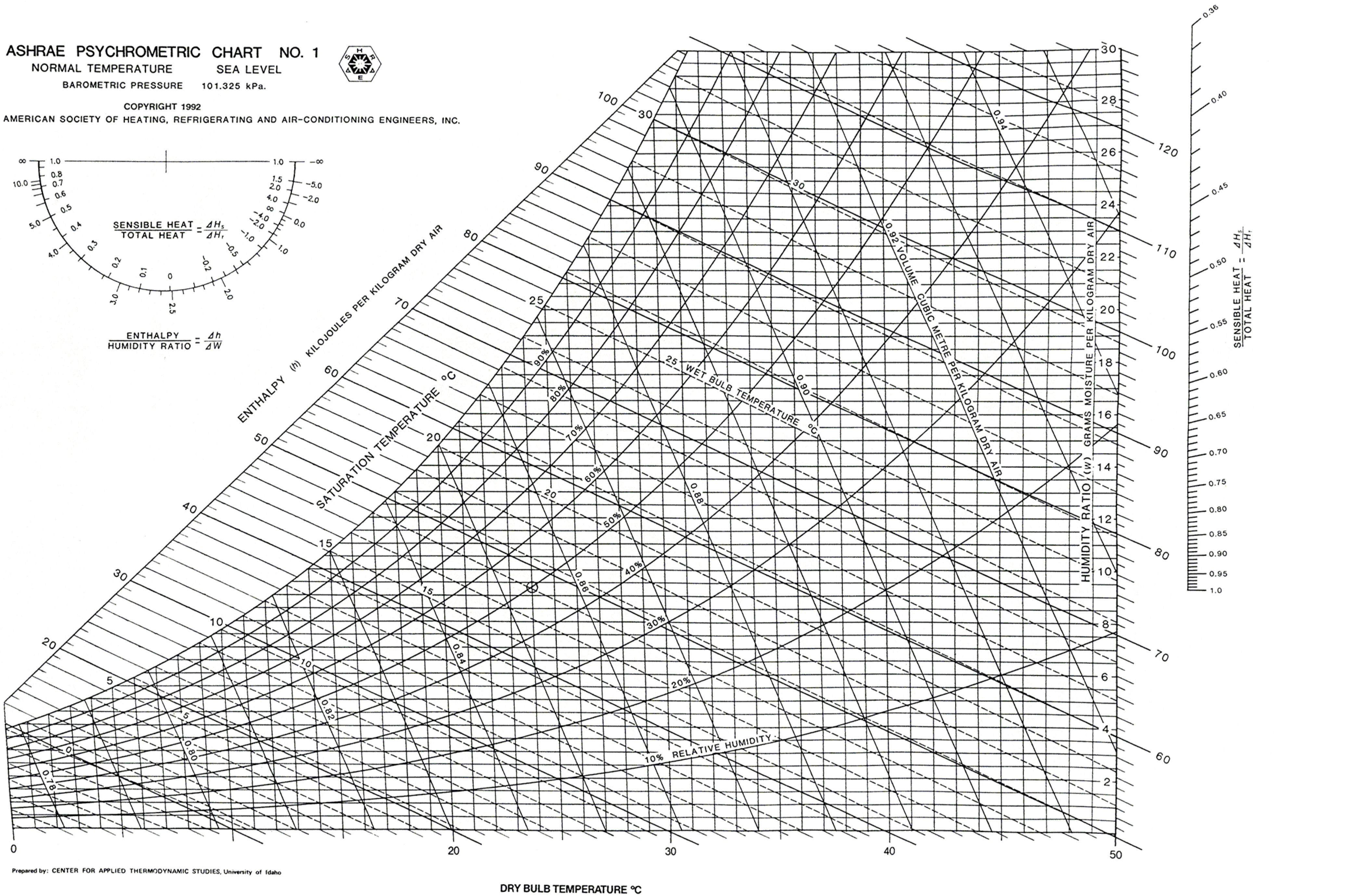
Ans.: b), c) and d)

4. Which of the following statements are TRUE?
   a) A float valve maintains a constant level of liquid in the float chamber
   b) A float valve maintains a constant pressure in the float chamber
   c) Low-side float valves are used with direct expansion type evaporators
   d) High-side float valves are used in flooded type evaporators

Ans.: a)

Table 3 Thermodynamic Properties of Water at Saturation (*Continued*)

Temp., °C	Absolute Pressure	Specific Volume, m³/kg$_w$			Specific Enthalpy, kJ/kg$_w$			Specific Entropy, kJ/(kg$_w$·K)			Temp., °C
		Sat. Solid	Evap.	Sat. Vapor	Sat. Solid	Evap.	Sat. Vapor	Sat. Solid	Evap.	Sat. Vapor	
t	p_{ws}, kPa	v_i/v_f	v_{ig}/v_{fg}	v_g	h_i/h_f	h_{ig}/h_{fg}	h_g	s_i/s_f	s_{ig}/s_{fg}	s_g	t
0	0.6112	0.001000	206.139	206.140	–0.04	2500.93	2500.89	–0.0002	9.1559	9.1558	0
1	0.6571	0.001000	192.444	192.445	4.18	2498.55	2502.73	0.0153	9.1138	9.1291	1
2	0.7060	0.001000	179.763	179.764	8.39	2496.17	2504.57	0.0306	9.0721	9.1027	2
3	0.7581	0.001000	168.013	168.014	12.60	2493.80	2506.40	0.0459	9.0306	9.0765	3
4	0.8135	0.001000	157.120	157.121	16.81	2491.42	2508.24	0.0611	8.9895	9.0506	4
5	0.8726	0.001000	147.016	147.017	21.02	2489.05	2510.07	0.0763	8.9486	9.0249	5
6	0.9354	0.001000	137.637	137.638	25.22	2486.68	2511.91	0.0913	8.9081	8.9994	6
7	1.0021	0.001000	128.927	128.928	29.43	2484.31	2513.74	0.1064	8.8678	8.9742	7
8	1.0730	0.001000	120.833	120.834	33.63	2481.94	2515.57	0.1213	8.8278	8.9492	8
9	1.1483	0.001000	113.308	113.309	37.82	2479.58	2517.40	0.1362	8.7882	8.9244	9
10	1.2282	0.001000	106.308	106.309	42.02	2477.21	2519.23	0.1511	8.7488	8.8998	10
11	1.3129	0.001000	99.792	99.793	46.22	2474.84	2521.06	0.1659	8.7096	8.8755	11
12	1.4028	0.001001	93.723	93.724	50.41	2472.48	2522.89	0.1806	8.6708	8.8514	12
13	1.4981	0.001001	88.069	88.070	54.60	2470.11	2524.71	0.1953	8.6322	8.8275	13
14	1.5989	0.001001	82.797	82.798	58.79	2467.75	2526.54	0.2099	8.5939	8.8038	14
15	1.7057	0.001001	77.880	77.881	62.98	2465.38	2528.36	0.2245	8.5559	8.7804	15
16	1.8188	0.001001	73.290	73.291	67.17	2463.01	2530.19	0.2390	8.5181	8.7571	16
17	1.9383	0.001001	69.005	69.006	71.36	2460.65	2532.01	0.2534	8.4806	8.7341	17
18	2.0647	0.001001	65.002	65.003	75.55	2458.28	2533.83	0.2678	8.4434	8.7112	18
19	2.1982	0.001002	61.260	61.261	79.73	2455.92	2535.65	0.2822	8.4064	8.6886	19
20	2.3392	0.001002	57.760	57.761	83.92	2453.55	2537.47	0.2965	8.3696	8.6661	20
21	2.4881	0.001002	54.486	54.487	88.10	2451.18	2539.29	0.3108	8.3331	8.6439	21
22	2.6452	0.001002	51.421	51.422	92.29	2448.81	2541.10	0.3250	8.2969	8.6218	22
23	2.8109	0.001003	48.551	48.552	96.47	2446.45	2542.92	0.3391	8.2609	8.6000	23
24	2.9856	0.001003	45.862	45.863	100.66	2444.08	2544.73	0.3532	8.2251	8.5783	24
25	3.1697	0.001003	43.340	43.341	104.84	2441.71	2546.54	0.3673	8.1895	8.5568	25
26	3.3637	0.001003	40.976	40.977	109.02	2439.33	2548.35	0.3813	8.1542	8.5355	26
27	3.5679	0.001004	38.757	38.758	113.20	2436.96	2550.16	0.3952	8.1192	8.5144	27
28	3.7828	0.001004	36.674	36.675	117.38	2434.59	2551.97	0.4091	8.0843	8.4934	28
29	4.0089	0.001004	34.718	34.719	121.56	2432.21	2553.78	0.4230	8.0497	8.4727	29
30	4.2467	0.001004	32.881	32.882	125.75	2429.84	2555.58	0.4368	8.0153	8.4521	30
31	4.4966	0.001005	31.153	31.154	129.93	2427.46	2557.39	0.4506	7.9812	8.4317	31
32	4.7592	0.001005	29.528	29.529	134.11	2425.08	2559.19	0.4643	7.9472	8.4115	32
33	5.0351	0.001005	28.000	28.001	138.29	2422.70	2560.99	0.4780	7.9135	8.3914	33
34	5.3247	0.001006	26.561	26.562	142.47	2420.32	2562.79	0.4916	7.8800	8.3715	34
35	5.6286	0.001006	25.207	25.208	146.64	2417.94	2564.58	0.5052	7.8467	8.3518	35
36	5.9475	0.001006	23.931	23.932	150.82	2415.56	2566.38	0.5187	7.8136	8.3323	36
37	6.2818	0.001007	22.728	22.729	155.00	2413.17	2568.17	0.5322	7.7807	8.3129	37
38	6.6324	0.001007	21.594	21.595	159.18	2410.78	2569.96	0.5457	7.7480	8.2936	38
39	6.9997	0.001007	20.525	20.526	163.36	2408.39	2571.75	0.5591	7.7155	8.2746	39
40	7.3844	0.001008	19.516	19.517	167.54	2406.00	2573.54	0.5724	7.6832	8.2557	40
41	7.7873	0.001008	18.564	18.565	171.72	2403.61	2575.33	0.5858	7.6512	8.2369	41
42	8.2090	0.001009	17.664	17.665	175.90	2401.21	2577.11	0.5990	7.6193	8.2183	42
43	8.6503	0.001009	16.815	16.816	180.08	2398.82	2578.89	0.6123	7.5876	8.1999	43
44	9.1118	0.001009	16.012	16.013	184.26	2396.42	2580.67	0.6255	7.5561	8.1816	44
45	9.5944	0.001010	15.252	15.253	188.44	2394.02	2582.45	0.6386	7.5248	8.1634	45
46	10.0988	0.001010	14.534	14.535	192.62	2391.61	2584.23	0.6517	7.4937	8.1454	46
47	10.6259	0.001011	13.855	13.856	196.80	2389.21	2586.00	0.6648	7.4628	8.1276	47
48	11.1764	0.001011	13.212	13.213	200.98	2386.80	2587.77	0.6778	7.4320	8.1099	48
49	11.7512	0.001012	12.603	12.604	205.16	2384.39	2589.54	0.6908	7.4015	8.0923	49
50	12.3513	0.001012	12.027	12.028	209.34	2381.97	2591.31	0.7038	7.3711	8.0749	50
51	12.9774	0.001013	11.481	11.482	213.52	2379.56	2593.08	0.7167	7.3409	8.0576	51
52	13.6305	0.001013	10.963	10.964	217.70	2377.14	2594.84	0.7296	7.3109	8.0405	52
53	14.3116	0.001014	10.472	10.473	221.88	2374.72	2596.60	0.7424	7.2811	8.0235	53
54	15.0215	0.001014	10.006	10.007	226.06	2372.30	2598.35	0.7552	7.2514	8.0066	54
55	15.7614	0.001015	9.5639	9.5649	230.24	2369.87	2600.11	0.7680	7.2219	7.9899	55
56	16.5322	0.001015	9.1444	9.1454	234.42	2367.44	2601.86	0.7807	7.1926	7.9733	56
57	17.3350	0.001016	8.7461	8.7471	238.61	2365.01	2603.61	0.7934	7.1634	7.9568	57
58	18.1708	0.001016	8.3678	8.3688	242.79	2362.57	2605.36	0.8060	7.1344	7.9405	58
59	19.0407	0.001017	8.0083	8.0093	246.97	2360.13	2607.10	0.8186	7.1056	7.9243	59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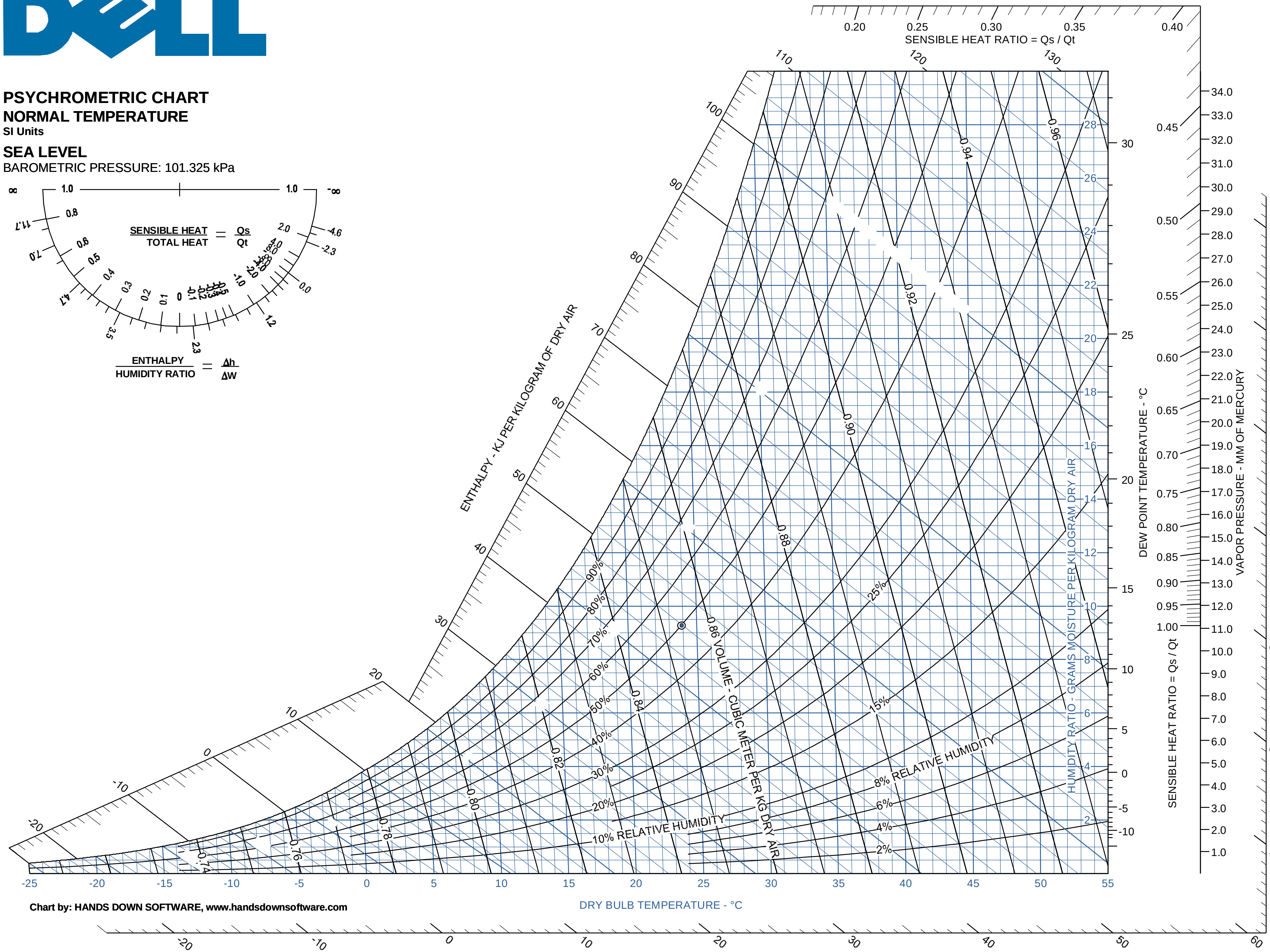
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔHt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY (h) KILOJOULES PER KILOGRAM DRY AIR
SATURATION TEMPERATURE °C
25 WET BULB TEMPERATURE °C
0.92 VOLUME CUBIC METRE PER KILOGRAM DRY AIR
HUMIDITY RATIO (W) GRAMS MOISTURE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
20%
30%
40%
50%
60%
70%
80%
90%
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔHt
DRY BULB TEMPERATURE °C
Prepared by: CENTER FOR APPLIED THERMODYNAMIC STUDIES, University of Idaho

PSYCHROMETRIC CHART
NORMAL TEMPERATURE
SI Units

SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa



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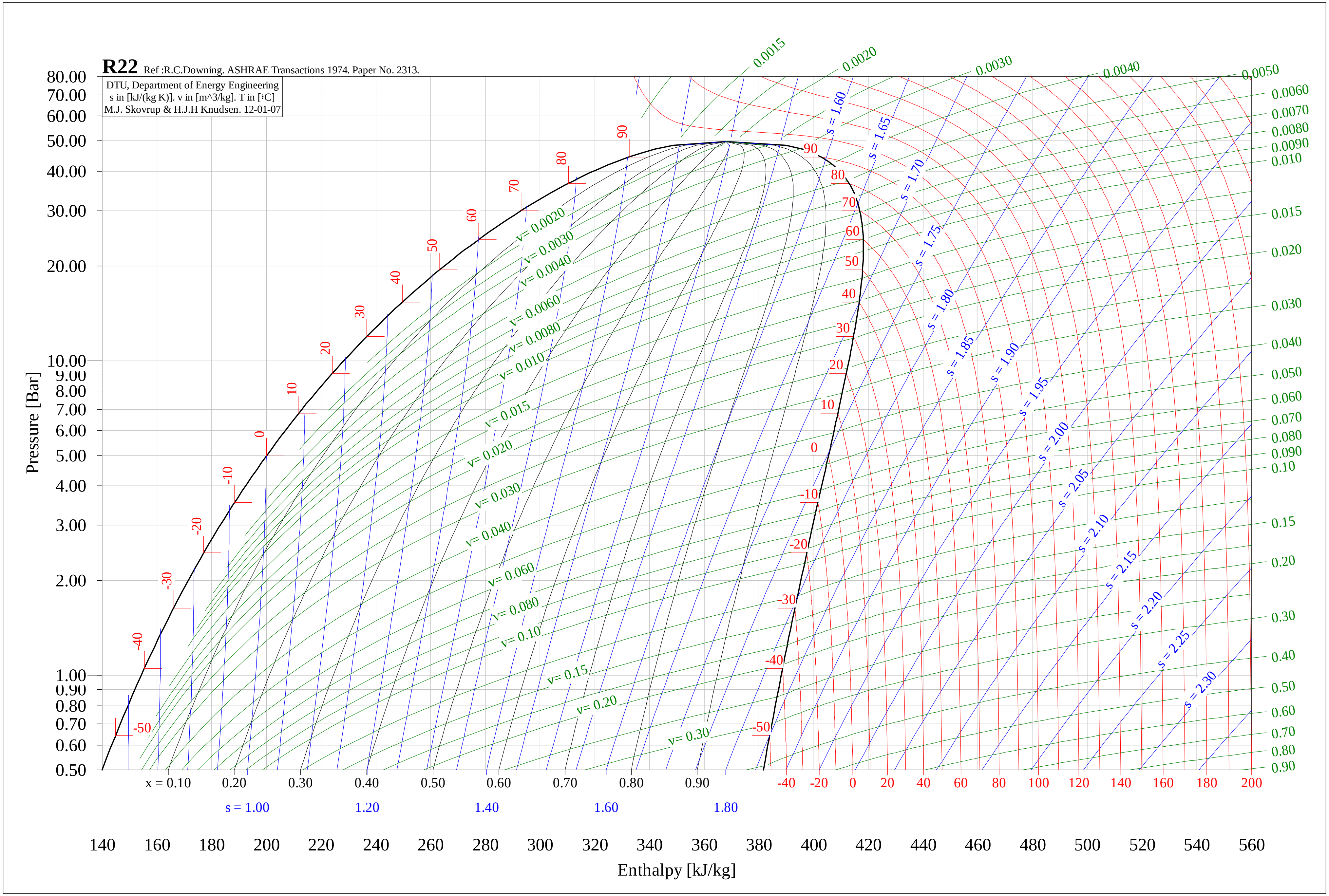
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DTU, Department of Energy Engineering
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M.J. Skovrup & H.J.H Knudsen. 12-01-07
Pressure [Bar]
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60.00
50.00
40.00
30.00
20.00
10.00
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.90
0.80
0.70
0.60
0.50
x = 0.10
0.20
0.30
0.40
0.50
0.60
0.70
0.80
0.90
s = 1.00
1.20
1.40
1.60
1.80
140
160
180
200
220
240
260
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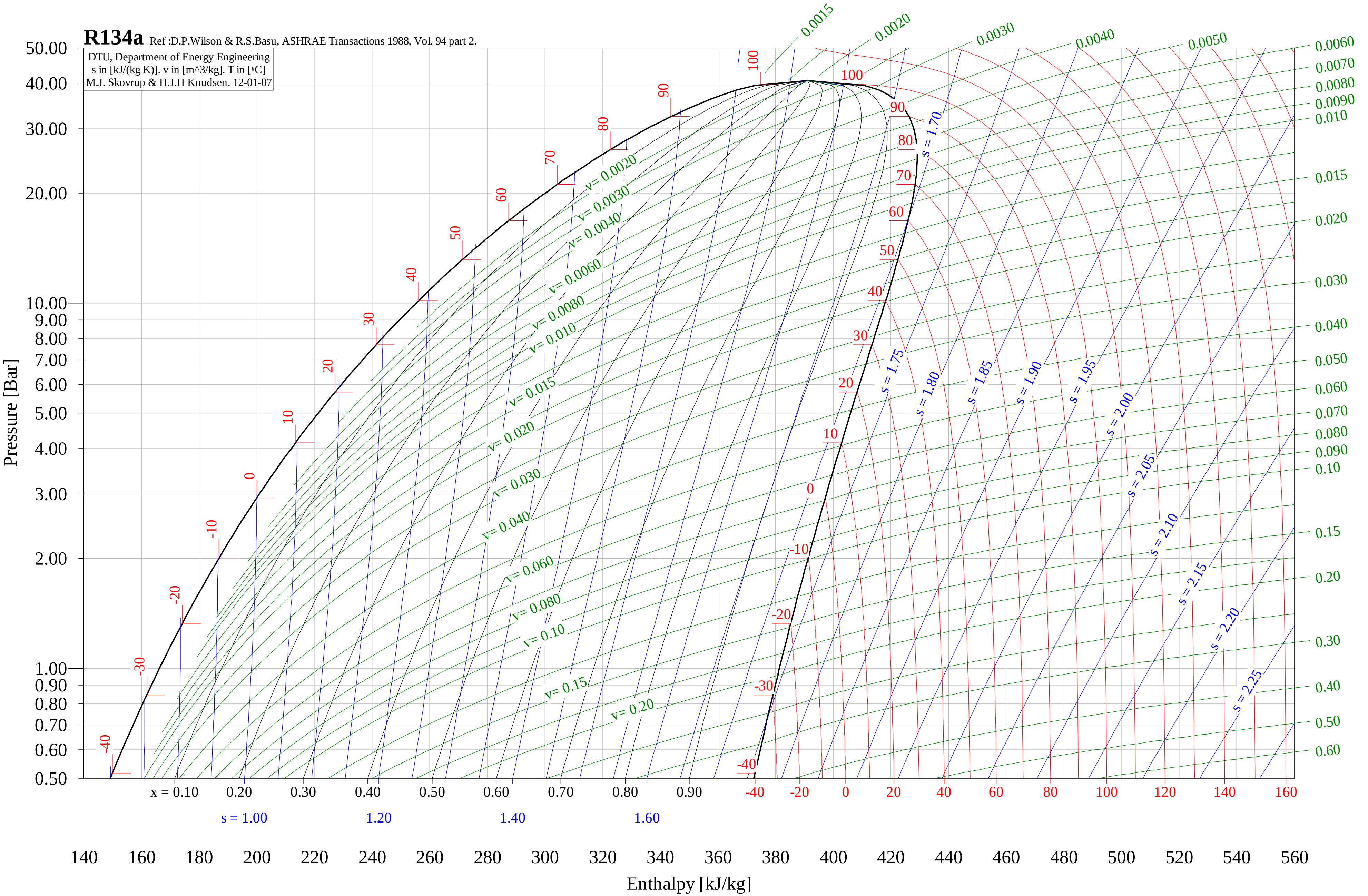
320
340
360
380
400
420
440
460
480
500
520
540
560
Enthalpy [kJ/kg]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DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [°C]
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Enthalpy [kJ/kg]
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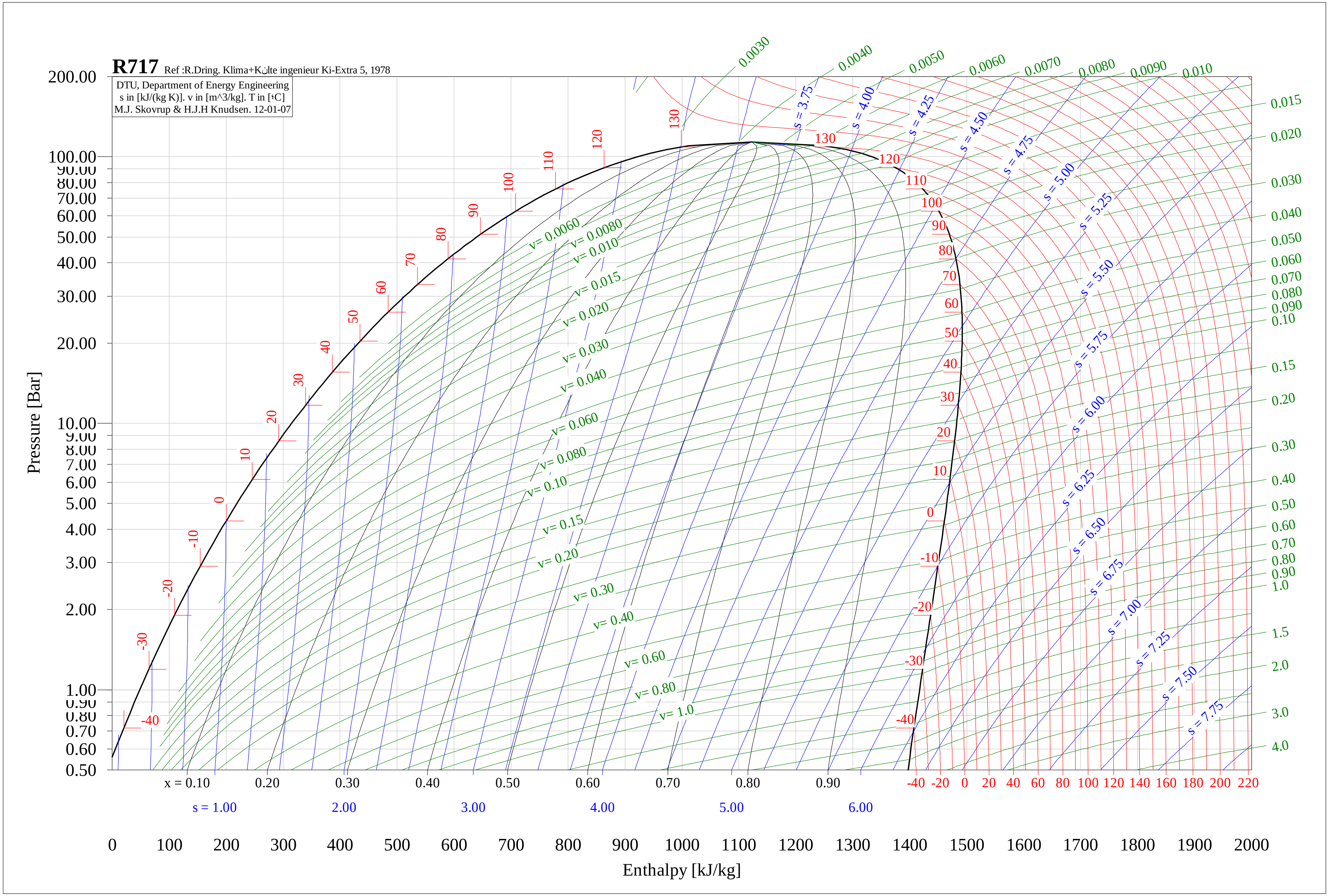
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Enthalpy [kJ/kg]
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7.00
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5.00
4.00
3.00
2.00
1.00
0.90
0.80
0.70
0.60
0.50
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300
1400
1500
1600
1700
1800
1900
2000
x = 0.10
0.20
0.30
0.40
0.50
0.60
0.70
0.80
0.90
s = 1.00
2.00
3.00
4.00
5.00
6.00
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-20
0
20
40
60
80
100
120
140
160
180
200
220
-40
-30
-20
-10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
v= 0.0060
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1.0
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s = 3.75
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s = 4.25
s = 4.50
s = 4.75
s = 5.00
s = 5.25
s = 5.50
s = 5.75
s = 6.00
s = 6.25
s = 6.50
s = 6.75
s = 7.00
s = 7.25
s = 7.50
s = 7.75