Lectures of
Air conditioning and Refrigeration

By

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2019-2020

References:

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L1 (Introduction to A.C.)

-Air Conditioning:

Is the process of treating air to control simultaneously its temperature, humidity, cleanliness and distribution to meet the comfort requirements of the occupants of the conditioned space.

Historically air conditioning has implied cooling or otherwise improving the indoor environment during the warm months of the year. In modern times the term has taken on a more literal meaning and can be applied to year –round environment situations. That is air conditioning refers to the control of temperature, moisture content, cleanliness, air quality and air circulation as required by occupants, a process, or a product in the space. This definition was first proposed by “Willis Carrier" an early pioneer in air conditioning.

-Applications of air conditioning and Refrigeration systems:

1- Residential air conditioning.

2- Industrial Air conditioning.

3- Air conditioning of vehicles.

4- Food storage and distribution.

5- Food processing.

6- Chemical and process industries.

7- Special applications of refrigeration.

-Fundamental Concepts:

Generally the background preparation for a study of air conditioning systems design is covered in thermodynamics, fluid mechanics and heat transfer curriculums.

The most important concept in this area is the first law of thermodynamics which leads to the important idea of the energy balance.

The principles dealing with the behavior of liquid and gases flowing in pipes and ducts are important especially the relationship between flow and pressure loss. The concepts are closely related and used in conjunction with thermodynamics concepts.

Generally the simplest concepts of heat transfer dealing with conduction, convection and radiation are used in system design.

-Definition of some of the more important processes required to air conditioning:

- ***Heating:***

  Is the transfer of energy to a space or to the air in a space by virtue of a difference in temperature between the source and the space or air. This process may take by different forms such as direct radiation and free convection to the space.

- ***Cooling:***

  Is the transfer of energy from a space or air supplied to a space by virtue of difference in temperature between the source and the space or air. In the usual cooling process air is circulated over a surface maintained at a low temperature.

- ***Humidification:***

  Is the transfer of water vapor to atmospheric air. Heat transfer is associated with mass transfer; however the transfer of mass and energy are manifested in an increase in the concentration of water in the air-water vapor mixture.

- ***Dehumidification:***

  Is the transfer of water vapor from atmospheric air. The transfer of energy is from the air as a consequence the concentration of water in the air- water vapor mixture is lowered. This process is most often accomplished by circulating the air over a surface maintained at a sufficiently low temperature to cause the condensation of water vapor from the mixture. It is also possible to dehumidify by spraying cold water into the air stream.

- ***Cleaning:***

  The cleaning of air usually implies filtering additionally it may be necessary to remove contaminant gases from air. Filtering is most often done by a process in which solid particles are captured in a porous medium.

- ***Air motion:***

  The motion of air in the vicinity of the occupant should be sufficiently strong to create uniform comfort conditions in the space, but gentle enough to be unnoticed.

-*ASHRAE:*

American society for heating, refrigeration & Air conditioning Engineers.

-*HVAC:*

Heating, ventilating, and air conditioning.

-Measuring units:

Hear, power& Energy:

1 BTU = 1055.06 J =1.055 KJ

BTU: British thermal unit

1 KW= 3413 Btu/hr

1 KW= 0.2844 TR

TR: Ton of refrigeration

TR = 12000 Btu/hr = 3.5168 KW

Temperature:

$T_F = 1.8\ Tc + 32$

$T_C = \frac{TF-32}{1.8}$

$\Delta Tc = 1.8\ \Delta Tf$

$$\Delta Tf = \frac{\Delta Tc}{1.8}$$

Absolute temperature:

$T = T_c + 273.16 \approx T_c + 273$ (K)

$T = T_F + 459.69 \approx T_F + 460$ (R)

L2 (Properties of moist air)

-Atmospheric air:

The atmospheric air contains many gaseous components as well as water vapor and miscellaneous contaminants (e.g. smoke& pollen).

-Dry air:

Exists when all water vapor & contaminants have been removed from atmospheric air. The composition of dry air is relatively constant, but small variations in the amounts of individual components occur with time, geographical location and altitude. The percentage composition of dry air by volume as: Nitrogen 78%, Oxygen 21% & 1% of different gases such as carbon dioxide, neon, helium, methane and hydrogen,….

-Moist air:

Is a mixture of dry air and water vapor. The amount of water vapor in moist air varies from zero (dry air) to a maximum value (saturation) .

-Properties of moist air:

The most exact calculation of thermodynamic properties of moist air is based on the formulations developed by hyland & Wexler. The psychometric chart & tables of a ASHRAE are calculated & constructed from these Formulations.

-Dry- bulb Temperature (DBT):

Is the temperature reading of the dry bulb.

-Wet- bulb Temperature (WBT):

When unsaturated moist air flow over the wet bulb of psychometer liquid water on the surface of the cotton wick evaporates and as a result. The temperature of the cotton wick & the wet bulb drop. This depressed wet bulb reading is called wet- bulb temperature

-Dew- point Temperature (T_{dew}):

Is the temperature of saturated moist air of the same moist air sample having the same humidity ratio & the same atmospheric pressure of the mixture Pat.

-Sensible heat:

Is that heat energy associated with the change of air temperature between two state points.

-Latent heat:

Is the heat energy associated with the change of the state of material. The latent heat of vaporization denotes the latent heat required to vaporize liquid water into water vapor.

Humidity calculations:

-Dalton's law:

Dalton's law shows that for a mixture of gases occupying a given volume at a certain temperature, the total pressure of mixture is equal to the sum of the partial pressure of the constituents of the mixture:-

$Pm= P_1 + P_2 + \ldots\ldots\ldots$

Where:

Pm: total pressure of mixture (pa)

P_1, P_2,.: partial pressure of constituents 1,2,…..(pa)

$m_m= m_a+ m_w$

Where:

m_m : mass of moist air (kg)

m_a: mass of dry air (kg)

m_w: mass of water vapor (kg)

$v_m = v_a = v_w$

$T_m = T_a = T_w$

$H_m = Ha + H_W$

The pressure of moist air: $Pat = Pa + Pw$

Where:

Pat: atmospheric Pressure or pressure of outdoor moist air (pa)

Pa & Pw: pressure of dry air & water vapor (pa)

-Humidity:

- humidity ratio (moisture content):

The humidity ratio of moist air (w) is the ratio of the mass of water vapor m_w to the mass of dry air m_a contained in the mixture of the moist air (kg/kg).

$$w = \frac{mw}{ma}$$

From Dalton's law & ideal gas law

$$P\,V = mRT$$

$$V = V_a = V_w \;,\; T = T_w = T_a$$

$$m = \frac{PV}{RT}$$

$$\therefore w = \frac{Pw\ \ Vw/Rw\,Tw}{Pa\ \ Va/Ra\,Ta} = \frac{Pw\,V\,Ra\,T}{Pa\,V\,Rw\ \,T} = \frac{Ra}{Rw}\frac{Pw}{Pa}$$

$$\therefore w = 0.62198\,\frac{Pw}{Pat-Pw}$$

Where Ra: dry air gas constant = 287 J\kg. K

Rw : water vapor gas constant = 461 J\kg. K

For moist air at saturation

$$W_S = 0.62198\,\frac{Pws}{Pat-Pws}$$

Where: P_{ws} : saturation pressure of water vapor of moist air

-Relative humidity (∅):

Is defined as the ratio of the mole fraction of water vapor xw in a moist air sample to the mole fraction of the water vapor in a saturated moist air sample xws at the same temperature & pressure .

$$\emptyset = \frac{xw}{xws} \quad /T, \text{P}$$

Where: $x_w = \frac{nw}{na+nw}$ & $xws = \frac{nws}{na+nws}$

n_a: No. of moles of dry air (mol)

n_w: No. of moles of water vapor in moist air sample (mol).

n_{ws}: No. of moles of water vapor in saturated moist air sample (mol).

The moist air is a binary mixture of dry air and water vapor so:

$$x_a + x_w = 1$$

From I.G. equation:

$$\emptyset = \frac{Pw}{PwS} /T, P$$

$P_w V = n_w R_o T_R$

$P_a V = n_a R_o T_R$

-Degree of saturation (μ):

Is the ratio of the humidity ratio of moist air w to the humidity ratio of the saturated moist air w_s.

$$\mu = \frac{w}{wS} /T, P$$

$$\emptyset = \frac{\mu}{1-(1-\mu)xws} = \frac{\mu}{1-(1-\mu)(Pws/Pat)}$$

-specific humidity q:

Is the ratio of the mass of water vapor to the to total mass of moist air.

$$q = \frac{mw}{(mw+ma)}$$

-Density of moist air (ρ):

It is the ratio of total mass of moist air to the volume:

$\rho = (m_a + m_w)/ V$

-Enthalpy of mixture of perfect gases:

$h = h_a + w\ hg$

ha: specific enthalpy for dry air

hg: specific enthalpy for saturated water vapor.

h = 1.022 td + w (hfg + 2.3 tdp)

Where hfg at tdp

L3 (Psychometric chart)

-Psychometric chart:

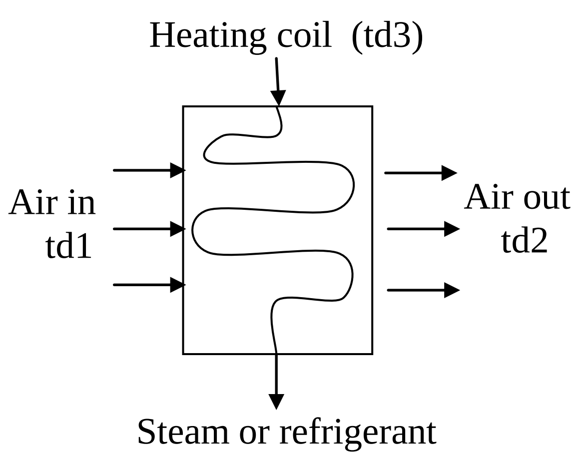
It is a graphical representation of the various thermodynamic properties of moist air. The psychometric chart is very useful for finding out the properties of air which are required in the field of air conditioning.

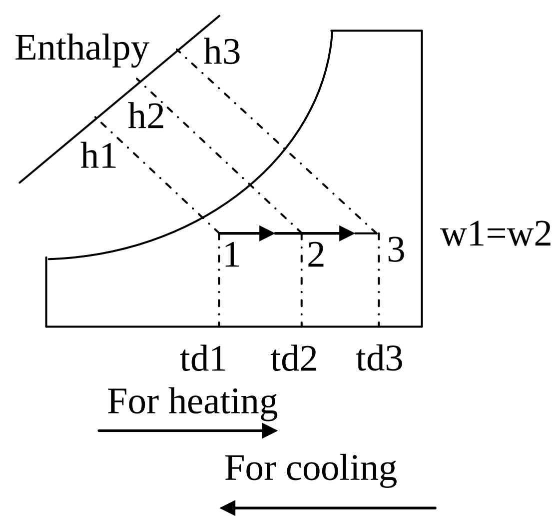
Ex.1

-Psychometric processes:

-Sensible heating and cooling:

Is the heating or cooling of air without any change in its humidity ratio.





$q = h_2 - h_1$

$= Cp_a\,(td_2 - td_1) + w\,Cp_w\,(td_2 - td_1)$

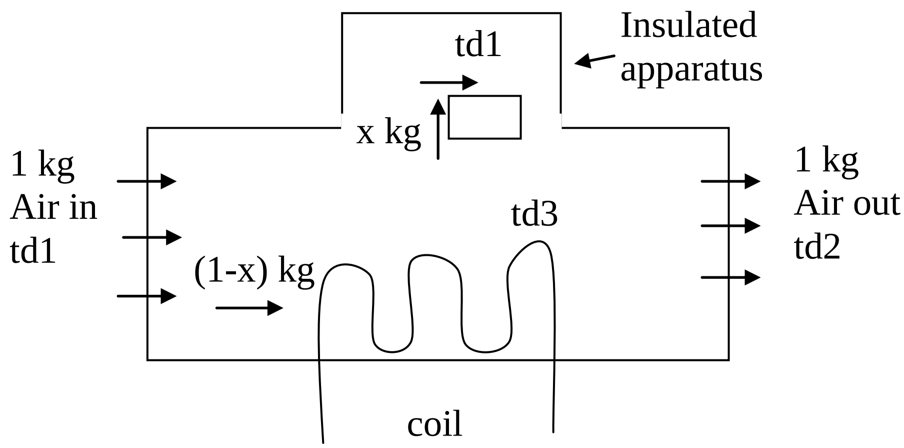
$= (Cp_a + w\,Cp_w)\,(td_2 - td_1)$

$q_{heating} = 1.022\,(td_2 - td_1)$ kJ/kg

$q_{cooling} = 1.022\,(td_1 - td_2)$ kJ/kg

where: Cp_m : humid specific heat $= (Cp_a + w\,Cp_w) = 1.022$ kJ/kg.K

-By-pass factor of heating and cooling coils:



-Enthalpy balance:

$x\,Cp_m * td_1 + (1-x)\,Cp_m * td_3 = 1 * Cp_m * td_2$

$x\,(td_3 - td_1) = td_3 - td_2$

$$x = \frac{td3 - td2}{td3 - td1}$$

Where x is the by-pass factor

For heating coil:

$$BPF = \frac{td3 - td2}{td3 - td1}$$

For cooling coil:

$$BPF = \frac{td2 - td3}{td1 - td3}$$

The by-pass factor depends upon the following:

1- The No. of fins provided in unit length.
2- The No. of rows in a coil in the direction of flow.
3- The velocity of flow of air.

Ex.2

-Efficiency of heating and cooling coils:

Efficiency of coil (η) =(1- BPF)   Also it called (Contact factor)

Efficiency of heating coil:

$$\eta_h = (1 - \text{BPF}) = 1 - \frac{\text{td3} - \text{td2}}{\text{td3} - \text{td1}} = \frac{\text{td2} - \text{td1}}{\text{td3} - \text{td1}}$$

Efficiency of cooling coil:

$$\eta_c = (1 - \text{BPF}) = 1 - \frac{\text{td2} - \text{td3}}{\text{td1} - \text{td3}} = \frac{\text{td1} - \text{td2}}{\text{td1} - \text{td3}}$$

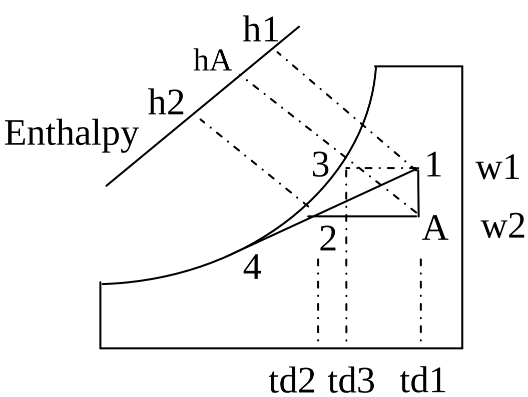
-Sensible heat factor:

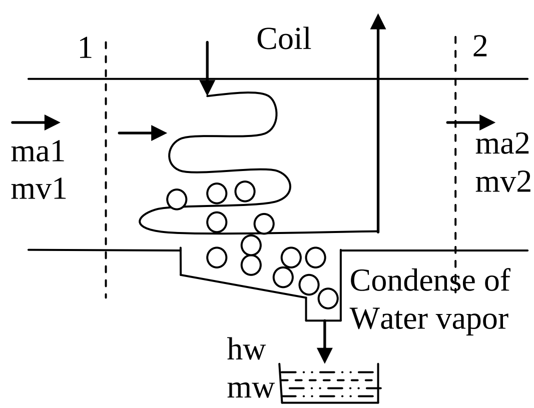
Is the ratio of the sensible heat to the total heat or also known as sensible heat ratio (SHR)

$$SHF = \frac{sensible\ heat}{total\ heat} = \frac{SH}{LH + SH}$$

-Cooling and Dehumidification:

This process is generally used in summer air conditioning to cool and dehumidify the air. The air is passed over a cooling coil or through a cold water spray. In this process the DBT and w of air decreases.





$m_{a2} = m_{a1}$, $m_{v2} \neq m_{v1}$

$m_{v1} = m_{v2} + m_w$ (mass balance)

$m_w = m_{v1} - m_{v2}$

$w = m_v / m_a$

$m_v = w\, m_a$

$m_a = m_{a1} = m_{a2}$, $m_w = w_1\, m_{a1} - w_2\, m_{a2}$

$m_w = m_a (w_1 - w_2)$

Energy equation:

$ma1\ h1 = q + m_{a2}\, h_2 + m_w\, h_w$

$q = m_{a1}\, h_1 - m_{a2}\, h_2 - m_w\, h_w$

$q = m_a (h_1 - h_2) - m_w\, h_w$

Where: q: coil load or refrigeration load.

m_w: mass of water vapor which condensate.

h_m: water enthalpy at temperature of coil surface.

td1: dry bulb temperature of air entering the coil.

tdp1: dew point temperature of air entering the coil = td3.

td4: effective surface temperature of coil also is known as apparatus dew point (ADP).

$$(\mathrm{BPF}) = \frac{\mathrm{td2} - \mathrm{td4}}{\mathrm{td1} - \mathrm{td4}} = \frac{\mathrm{td2} - \mathrm{ADP}}{\mathrm{td1} - \mathrm{ADP}}$$

Also

$$(\mathrm{BPF}) = \frac{\mathrm{w2} - \mathrm{w4}}{\mathrm{w1} - \mathrm{w4}} = \frac{\mathrm{h2} - \mathrm{h4}}{\mathrm{h1} - \mathrm{h4}}$$

q = LH + SH

q = (h1 – h2) = (h1 –hA) +(hA – h2)

LH: latent heat removed due to condensation of vapor = h1 – hA

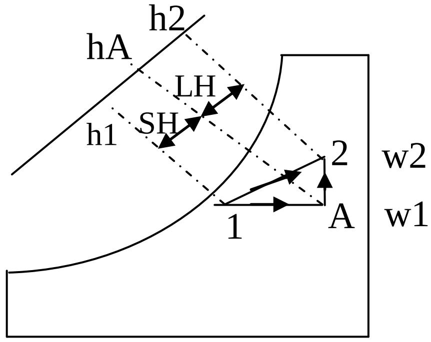
SH: sensible heat removed (cooling) = hA – h2

$$SHF = \frac{SH}{LH + SH} = \frac{hA - h2}{h1 - h2}$$

Ex.3

-Heating and humidification:

This process is generally used in winter air conditioning to warm and humidify the air. It is the reverse process of cooling and dehumidification. The process of heating and humidification is shown by line 1-2 on the psychometric chart.



$q = LH + SH$

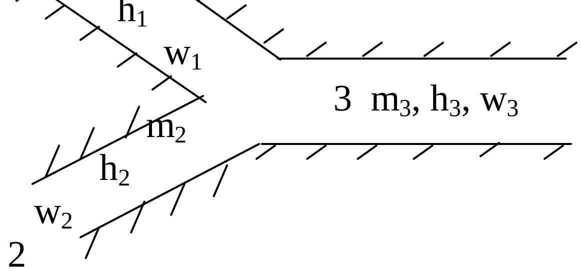
$q = h_2 - h_1 = (h_2 - h_A) + (h_A - h_1)$

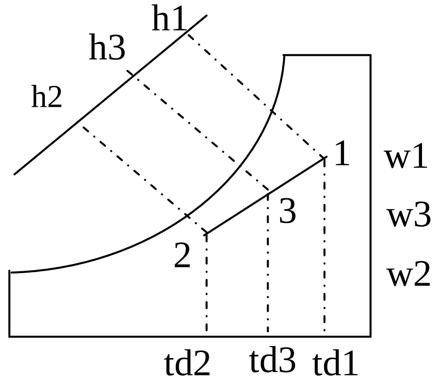
Where: SH: sensible heat added $(h_A - h_1)$

LH: latenet heat of vaporization $(h_2 - h_A)$

-Adiabatic mixing of two air streams:

A common process in air conditioning systems is the adiabatic mixing of two moist air as shown in figure bellow:





Mass balance:

m1 + m2 =m3

Energy balance:

m1 h1 +m2 h2 = m3 h3

Mass balance of water vapor

m1 w1 + m2 w2 = m3 w3

substitution of mass balance equation in energy balance equation:

m1 (h1 – h3) = m2 (h3 –h2)

$$\frac{m1}{m2} = \frac{(h3 - h2)}{(h1 - h3)}$$

Substitution equation of mass balance in equation of vapor mass balance:

$$\frac{m1}{m2} = \frac{(w3 - w2)}{(w1 - w3)}$$

$$\frac{m1}{m2} = \frac{(h3 - h2)}{(h1 - h3)} = \frac{(w3 - w2)}{(w1 - w3)}$$

$$T_{3} = \frac{m1\,T1 + m2\,T2}{m1 + m2}$$

$$h_{3} = \frac{m1\,h1 + m2\,h2}{m1 + m2}$$

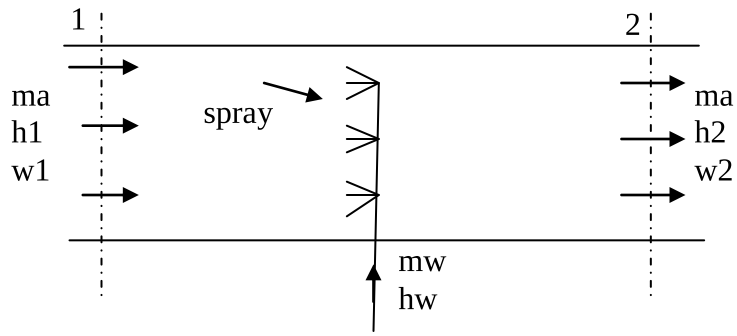
$$w_{3} = \frac{m1\,w1 + m2\,w2}{m1 + m2}$$

The final conditions of the mixture (point 3) lie on the straight line 1-2.

<u>Ex.4</u>

-Adiabatic mixing of water injected into moist air (humidification):

Steam or liquid water can be injected into a moist air stream to raise its humidity as shown in figure bellow:



$m_a\, h_1 + m_w\, h_w = m_a\, h_2$

$m_a\, w_1 + m_w = ma\, w_2$

$$hw = \frac{(h2 - h1)}{(w2 - w1)}$$

<u>Ex.5</u>

L4 (Heat transfer principles)

Heat transfer:

Whenever a temperature gradient exists within a system or when two systems at different temperature are brought into contact, energy is transferred. The process by which the energy transport takes place is known as heat transfer.

Therefore heat transfer can be defined as the transmission of energy from one region to another as a result of a temperature difference between them.

Modes of heat transfer:

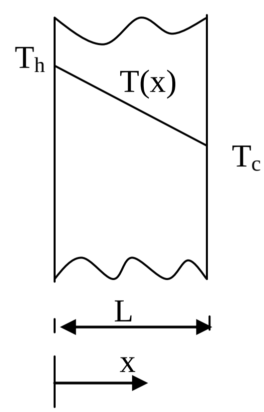
Heat transfer generally recognizes three distinct modes: conduction, convection & radiation.

Conduction:

$$q = - k A \frac{dT}{dX}$$

$$q\, dx = - k A\, dT$$

$$\frac{q}{A}\int_0^L dx = -\int_{Tn}^{Tc} kdT$$



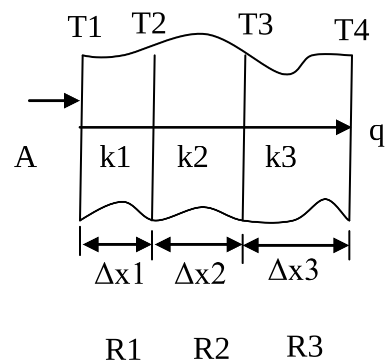
When k is constant

$$q = \frac{AK}{L}(Th - Tc) = \frac{\Delta T}{Rcond.}$$

Where R_{cond}: conduction resistance

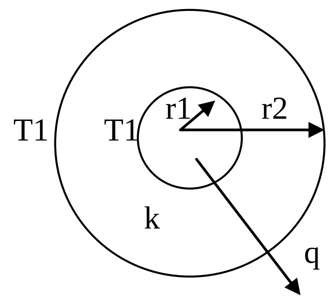
$$R_{cond} = \frac{L}{AK}$$

Heat conduction in a composite plan wall:



$$q = \frac{\Delta T}{R1+R2+R3} = \frac{\Delta T}{\sum_{i=1}^{n} Ri}$$

Conduction through hollow cylinders:



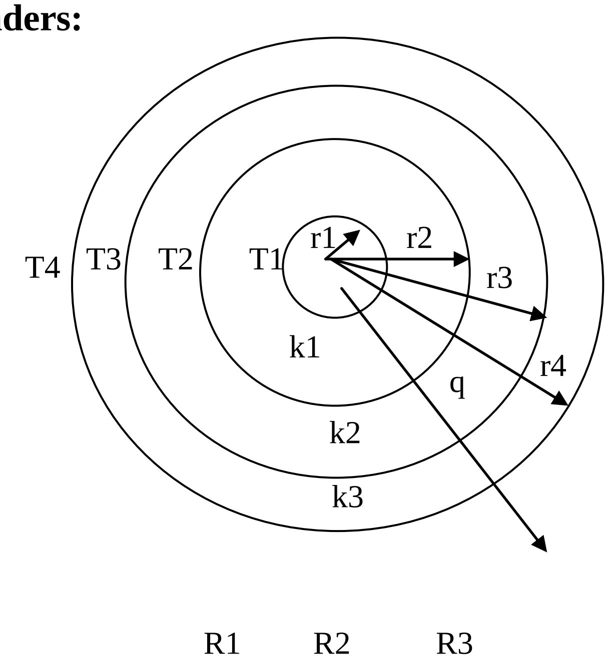
$$q = - k\ A\ \frac{dT}{dr}$$

where $A = 2\pi r1$

$$q = \frac{2\pi kL\ \Delta T}{Lnr2/r1} = \frac{\Delta T}{\left(\frac{Lnr2/r1}{2\pi kL}\right)} = \frac{\Delta T}{\text{Rcond}}$$

where $R_{cond.} = \frac{Ln^{r2}/_{r1}}{2\pi kL}$

Conduction through composite cylinders:



$$q = \frac{\Delta T}{R1+R2+R3}$$

where

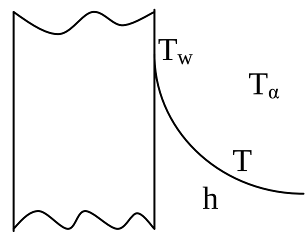
$$R_1 = \frac{Ln^{r2}/_{r1}}{2\pi k1L}$$

$$R_2 = \frac{Ln^{\ r3}/_{r2}}{2\pi k2L}$$

$$R_3 = \frac{Ln^{\ r4}/_{r3}}{2\pi k3L}$$

$$q = \frac{\Delta T}{\sum R}$$

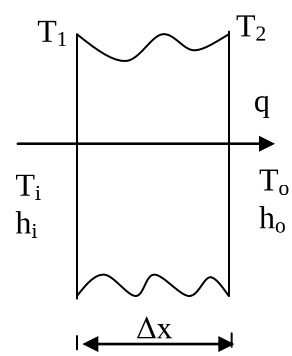
convection:



$$q = h\,A\,(Tw - T\infty) = \frac{Tw - T\infty}{\frac{1}{hA}} = \frac{Tw - T\infty}{Rcouv}$$

R_{conv}: convection resistance $= \frac{1}{hA}$

Heat transfer through wall with internal & external convection:



$$q = \frac{Ti - T1}{Rconvi} = \frac{T1 - T2}{Rcond} = \frac{T2 - To}{Rconvo}$$

$$q = \frac{Ti - To}{Rconvi + Rcond + Rconvo}$$

where $R_{convi} = \frac{1}{hiA}$, $R_{cond} = \frac{\Delta X}{KA}$, $R_{cnvo} = \frac{1}{hoA}$

Overall heat transfer coefficient:

$$q = U\,A\,\Delta T = \frac{\Delta T}{R}$$

$$R = \frac{1}{UA}$$

where:

U: overall heat transfer coefficient.

A: heat transfer area

ΔT: temperature difference

$$UA = \frac{1}{Rconvi + Rcond + Rconvo}$$

$UA = \frac{1}{\frac{1}{hiAi}+\frac{\Delta x}{KA}+\frac{I}{hoAo}}$ for plan wall

$Q = \frac{Ti-To}{\frac{1}{hiA}+\frac{\Delta x}{KA}+\frac{1}{hOA}}$

$UA = \frac{1}{\frac{1}{hiAi}+\frac{Ln(ro/ri)}{2\pi kL}+\frac{1}{hoAo}}$ for pipe

$Ui = \frac{1}{\frac{1}{hi}+\frac{Ai\,Ln(ro/ri)}{2\pi kL}+\frac{Ai}{hoAo}}$ based on A_i

$Uo = \frac{1}{\frac{Ao}{hiAi}+\frac{AoLn(ro/ri)}{2\pi kL}+\frac{1}{ho}}$ based on A_o

Calculation of heat transfer coefficient(h):

$Re = \frac{\rho uL}{\mu}$ For flow over plate

$Re = \frac{\rho uD}{\mu}$ For flow in pipe

For flow in channels D is replaced by Dh

$Dh = \frac{4A}{P}$

Dh: hydraulic diameter

A: flow area

p: wetted permiter

$Nu = \frac{hL}{K} = \frac{hD}{K} = \frac{hDh}{k}$

For flow over plan surface

If $Re < 5\times10^5$ Flow is laminar and $Nu = 0.664\ Re^{0.5}\ pr^{1/3}$

If $Re > 5\times10^5$ Flow is turbulent $Nu = 0.036\ Re^{0.8}\ pr^{1/3}$

For flow in pipes

If $Re < 2300$ Flow is laminar and

$Nu = 4.364$ for constant wall heat flux

$Nu = 3.66$ for constant wall temperature

If $Re > 2300$ Flow is turbulent and $Nu = 0.023\ Re^{0.8}\ pr^{0.4}$

Radiation heat transfer:

Black body radiation:

$Eb = \sigma T^4$ (w/m^2)

Where:

Eb: energy emitted by black body

σ: Stefan Boltzmann constant= 5.669×10^{-8} (w/m^2.k^4)

In general

$q = \varepsilon\, \sigma\, A\, T^4$ (W)

ε: emissivity of body ($0 \le \varepsilon \le 1$)

$\varepsilon = 1$ for black body

In air conditioning total resistance is

$$R = \frac{1}{fiA} + \frac{\Delta X1}{K1A} + \frac{\Delta X2}{K2A} + \cdots + \frac{1}{foA}$$

fi=9.37 w/m^2.c for inside (still air)

fo= 34.1 w/m^2.c for winter with wind speed 24 km/h

fo= 22.7 w/m^2.c for summer with wind speed 12 km/h fi: inside film or surface conductance

fo: outside film or surface conductance

$$U = \frac{1}{\frac{1}{fiA}+\frac{\Delta x1}{k1A}+\frac{\Delta x2}{k2A}.......+\frac{1}{foA}}$$

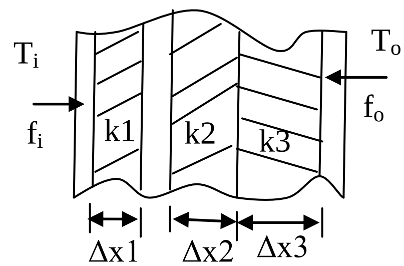
For plan walls (equal area)

$$U = \frac{1}{\frac{1}{fi}+\frac{\Delta x1}{k1}+\frac{\Delta x2}{k2}.......+\frac{1}{fo}}$$

When air space is provided between the materials then the overall coefficient of heat transmission is:

$$U = \frac{1}{\frac{1}{fi}+\frac{\Delta x1}{k1}+\frac{1}{ka}+\frac{\Delta x2}{k2}+\frac{\Delta x3}{k3}+\frac{1}{fo}}$$

Ka: thermal conductance of air space.

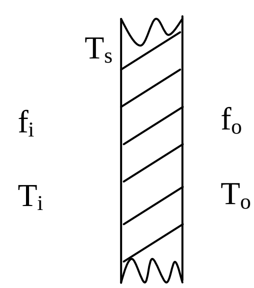


Wall surface temperature:

The inside wall temperature con not be assume to be equal to the inside air temperature. If the wall surface temperature is lower than the dew point temperature of air this cause a damage to the plaster & wood work due to condensation of the vapor in air. In winter must be decrease the humidity of air or decrease the film resistance by increase the velocity of air of surface to prevent the condensation. We can calculate wall surface temperature from the temperature of air at sides of wall.

$$q = \frac{A}{Rtotal}(Ti - To) = \frac{A}{Rf}(Ti - Ts)$$

$$T_{surface} = Ti - \frac{Rf}{Rtotal}(Ti - To)$$



L5 (comfort conditions)

- Thermal exchanges of body with environment:

The human body maintains its thermal equilibrium with environment by means of three modes of heat transfer (evaporation, radiation and convection)

A human body feels comfortable when the heat produced by metabolism is equal to the sum of the heat dissipated to the surroundings & the heat stored in human body by raising the temperature of body tissues

$Q_m - W = Q_E \pm QR \pm Qc \pm Qs$

Q_m: Metabolic heat produced in body.

W: Useful rate of working.

Q_m - W: Heat to be dissipated to the atmosphere.

Q_R: Heat lost or gained by radiation.

Q_E: Heat lost by evaporation.

Qc: Heat lost or gained by convection.

Qs: Heat stored in the body.

Metabolic heat produced Q_m depends upon the rate of food energy consumption.

The unit of metabolic is(met).

$$\text{Met} = 58.2 \text{ w/m}^2$$
$$= 50 \text{ k cal/hr. m}^2$$
$$= 18.4 \text{ Btu/hr. ft}^2$$

The value of metabolic is obtain from tables according to activity.

From the weight w (kg) & length (H) in (m) & age we can Calculate:

Surface area of human body

$A = 73.8\ W^{0.425}\ .\ H^{0.725}$ (m^2)

Metabolic of man

$MQ_m = 3.22 + 0.66\ W + 0.12\ H - 0.33\ age$ (W)

Metabolic of women's

$WQ_m = 3.15 + 0.47w + 0.09H - 0.22\ age$ (W)

Density of human body:

$\rho = 0.028\ w^{-0.3}.H^{0.725} + 0.75$ (Kg/m^3)

Human body efficiency

$$\eta = \frac{w}{Qm}$$

Heat loss by evaporation:

The heat loss by evaporation is always positive and it depends upon the vapor pressure difference between the skin surface and the surrounding air.

$Q_E = Cd\ A\ (P_s - P_v)\ hfg\ Cc$

Cd: Diffusion coefficient in kg of water evaporated per unit surface area and pressure difference per hour.

A: skin surface area = 1.8 m^2 for normal man.

P_s: saturation vapor pressure corresponding to skin Temperature.

P_v: vapor pressure of air.

Cc: Factor which a counts for clothing.

-Heat loss or gain by radiation (Q_R):

It depend on the mean radiant temperature which is the average surface temperature of the surrounding objects (MRT)

When MRT$< DBT \Rightarrow QR \quad is\ positive \&\ heat\ is\ loss$

When MRT $> DBT \Rightarrow$ QR is negative & heat is gain

-Heat loss by convection (Qc):

$Q_c = U\,A\,(T_B - T_S)$

Where:

U: Body film coefficient of heat transfer .

A: Body surface area = 1.8 m^2 for normal man.

T_B: Temperature of body.

T_S: Temperature of surrounding.

If Ts > TB ⇒ Q_c is (-) and the heat will gained by body.

If Ts < TB ⇒ Q_c is (+) and the heat will lost from body.

When Q_E,Q_R & Q_c are high & positive & (Q_E+Q_R+ Q_c) is greater than (Q_m -W) then the heat stored in the body Q_s will be negative i.e. body temperature falls down. Thus the sick, weak or old human feels more cold.

-Physiological hazards resulting from heat:

In summer T_S always > T_B Thus Q_R & Q_C are negative & the body will gain heat from surrounding.

Following are some of the physiological hazards which may result due to the rise of human temperature.

1-Heat exhaustion:

It is due to the failure of normal blood circulation. The symptoms include headache, dizziness.

2-Heat cramp:

It is results from loss of salt due to an excessive rate of perspiration.

3-Heat stroke:

When a body is exposed to excessive heat & work the body temperature may rise to 40.5 or higher at this temperature seveating ceases & the human may enter a coma.

-Factors affecting human comfort:

1-Effective Temperature:

The degree of warmth or cold felt by a human body depends mainly on:

a- DBT, b -Ø, c- Air velocity

It is also defined as the index which correlates the combined effects of air temperature, relative humidity& air velocity on the human body.

-Comfort chart: which is the practical application of the concept of effective Temperature fig.(2-1).

2-Heat production & regulation in human body:

Human body is like a heat engine which gets its energy from the combustion of food within it. The rate at which the body produces heat is termed as metabolic rate.

The human body attempts to maintain its temperature by withdrawal of blood& by increased rate of metabolism.

3-Heat & moisture losses from human body:

The heat is given off from the human body is either sensible or latent heat or both. In order to design any A.C. system it is necessary to know the rates at which these two forms of heat are given under different conditions of air temperature & body activity.

4-Moisture content of air:

In winter if the cold outside air having a low moisture content leak into conditioned space it will cause a low relative humidity unless moisture is adds to the air by the process of humidification. In summer the reverse will take place unless moisture is removed by dehumidification.

5-Quantity & quality of air:

The air in an occupied space should free from toxic, dust & odour.

6-Air motion:

The air motion which includes the distribution of air is very important to maintain uniform temperature in the conditioned space.

7-Cold& hot surfaces:

The cold or hot objects in a conditioned space may cause discomfort to the occupants.

8-Air stratification:

The movement of air to produce the temperature gradient from floor to ceiling. To achieve comfortable conditions the A.C. systems must be designed to reduce air stratification to minimum.

-Factors affecting optimum effective temperature.

1- Climatic & seasonal differences.
2- Clothing.
3- Age& sex.
4- Duration of stay.
5- Kind of activity.
6- Density of occupants.

L6 (Cooling load)

Cooling load calculation is normally made to size HVAC systems and their components. In principle, the loads are calculated to maintain the indoor design conditions.

In cooling load calculation, there are four heat flow terms:

1-Cooling load:

Is the total heat required to be removed from the space in order to bring it at the desired temperature by the A.C. equipments. The purpose of estimation of cooling load is to determine the size of the A.C. equipment.

2-Heat gain:

Is a simultaneous summation of all external heat flows plus the heat generated inside the building.

(heat gain$\neq$ $cooling\ load$)

3-Space heat extraction rate:

Is usually the same as the space cooling load but with an assumption that the space temperature remains constant.

4-Cooling coil load:

Is the summation of all the cooling loads of the various space served by the equipment plus any loads external to the spaces such as duct heat gain, duct leakage, fan heat

-Components of cooling load:

The components of cooling load are classified into (external sources) which are the heat gain sources coming from outside and (internal sources) which are the heat gain sources generated inside the conditioned space.

There is another classification for the cooling load components which are: sensible & latent heat gain:

A-Sensible heat gain:

The sensible heat gain is occur when there are a direct addition of heat to the enclosed space which caused by any or all of the following sources:

1- Conduction through exterior walls, floors, ceilings, doors & windows due to temperature difference.
2- The heat received from solar radiation
   a- Heat transmitted directly through glass of windows or doors.
   b- Heat absorbed by wall & roofs exposed to solar radiation.
3- Heat conducted from unconditioned places in the same building.
4- Heat given from light, motors, machinery, cooking operation ,etc.
5- Heat received from the occupants.
6- Heat carried by infiltrating air.
7- Heat gain through the walls of ducts carrying conditioned air through unconditioned spaces.
8- Heat gain from the fan work

B-Latent heat gain:

Which is occur due to addition of water vapor to the air of enclosed space. The latent heat gain may occur due to any or all of the following sources :

1- Heat gain due to moisture in the outside air entering by infiltration.
2- Heat gain due to vaporization of moisture from occupants.

3- Heat gain due to vaporization of moisture from any process such as cooking of food.
4- Heat gain due to moisture passing directly into the conditioned space through permeable walls or from outside where the water vapor pressure is higher

-Design conditions:

The amount of cooling load has to be accomplished to keep building comfortable depends on the desired conditions indoor & on the outdoor conditions in a given day.

-Indoor design conditions:

For most of the comfort systems, the recommended indoor conditions:

1- Summer: 23 to 26 C^o DBT & 50% RH
2- Winter : 21 to 23 C^o DBT & 20-30% RH

-Outdoor design conditions:

Outdoor design conditions are determined from published data for the specific location.

-Calculation of cooling load:

1-Heat gain through external walls and roofs:

$$Q = U\,A\,(T_i - T_O) = \frac{A\,(Ti - To)}{R_{total}}$$

Q:Sensible heat flow

A: Area of wall

U: Overall heat transfer coefficient ($w/m^2.\mathring{c}$).

ΔT ($Ti-T_o$): temperature difference for the inside & outdoor temperature.

For summer months the combined effect of convection, conduction, radiation & thermal (time lag) for (opaque surfaces) are to be considered. The time lag is the difference between the time of peak outside temperature & the time of the resulting indoor temperature. All of the transmitted solar radiation does not immediately act to increase the cooling load, some is absorbed by wall & is radiated back to indoor space even after the sunset. Therefore the time at which the space may realize the heat gain as a cooling load is considerably offset from the time the heat started to flow. This phenomenon is called «thermal storage effect».

$$Q = U\ A\ CLTD$$

Where :

CLTD : cooling load temperature difference for roofs, walls (from tables 6-2 and 6-4).

Not that the values in tables (6.2 & 6.4) assume a dark color, design outdoor temperature of 35 $\mathring{c}$, mean outdoor temperature (Tm = 29.4), design inside temperature (T_R = 25.5), a daily range (DR = 11.6 $\mathring{C}$), clear sky on the July 21& latitude = 40 ^{o} north.

When conditions are different, CLTD values from tables (6.2 & 6.4) must be corrected before being used.

<u>For walls:</u>

$$Q_W = U_W\ A_W\ CLTD_{corr.}$$

$$CLTD_{corr.} = [\ CLTD\ (tables\ 6.3 + 6.4) + LM\ (table\ 6.5)]*\ K + (25.5-T_R) + (T_O-29.4)$$

Where: LM: latitude month correction factor from table (6.5).

K: wall color correction factor

K = 1 for dark colored or light in an industrial area.

K= 0.83 for permanently medium - colored (rural area)

K= 0.65 for permanently light - colored (rural area)

$T_{O} = t_o - \left(\frac{Dr}{2} \right)$

Dr: Daily range

Ex.1

For roofs:

$Q_r = U_r \ A_r \ CLTD_{corr.}$

$CLTD_{corr.} = [(CLTD \ (table\ 6.2) + LM \ (table\ 6.5)) *K + (25.5-TR) + (T_O-29.4)] *f$

K: wall color correction factor

K= 1 for dark colored or light in an industrial area .

K= 0.5 for light colored roof

f: is a factor for attic fan & or ducts above ceiling.

f = 1 for no attic or ducts.

f = 0.75 for positive (mechanical) ventilation

(25.5 - TR): is the indoor design temperature correction.

(To - 29.4): Is the outdoor design temperature correction.

Ex.2

2-Heat gain thought glass:

Cooling load caused by solar radiation:

$Q_r = A \ \ Sc \ (tables \ (6.9 + 6.10 + 6.11)) \times SHG_{max}. \ table \ (6.6) \times CLF \ (tables \ (6.7 + 6.8))$

Cooling load caused by conduction:

$Q_c = U_g \ \ A \ \ CLTD_{corr}.$

$CLTD_{corr} = CLTD \ (table\ 6.12) + (25.5 - TR) + (To - 29.4)$

Where: SC: shading coefficient

CLF: cooling load factor = $\frac{sensible\ cooling\ load}{sensible\ heat\ gain}$

SHG_{max}: maximum solar heat gain transmission.

Total heat gain from windows:

$Q = A\ (U.\ CLTD_{corr}. + SC.\ SHG_{max}\ .CLF)$

Ex.3

3-Heat gain from adjacent unconditioned spaces:

Which is the heat gained from adjacent unconditioned spaces which have usually temperature higher than the temperature of conditioned space.

$$Q = U\ A\ (Tb - Ti)$$

Where: Tb: temperature of unconditioned space

Ti: temperature of conditioned space

4-Heat gain due to infiltration:

The infiltration air is the air that enters a conditioned space through window cracks & opening of doors.

$$Q_{i.s} = m_i.Cp\ (T_o - T_i) = 1.22\ V\ (T_o - T_i) \qquad (kW)$$

$$Q_{i.l} = m_i.\ hfg\ (w_o - w_i) = 2940\ V\ (w_o - w_i) \qquad (kW)$$

$$Q_{i.total} = m\ (h_o - h_i) = 1.2\ V\ (h_o - h_i) \qquad (kW)$$

$$L_{crack} = 2\ (H + w)$$

m_i: mass flow rate of infiltrated air (kg/s)

V: volumetric rate of infiltrated air (m^3/s) (from table (8)

$Cp = 1.017\ kJ/kg.\ C^o$

$\rho = 1.2\ kg/m^3$

For two walls exposed we must take the length of crack for the wall which has the greatest length.

Ex.4, Ex.5

5-Heat gain due to ventilation:

The ventilation (i.e. supply of outside air) is provided to the conditioned space in order to minimize odors, concentration of smoke, carbon dioxide and other undesirable gases so that freshness of air could be maintained. If the infiltration air quantity is larger than the ventilation quantity then the latter should be increased to at least equal to the infiltration air. The outside air adds sensible as well as latent heat.

$$Q_{v.s} = m_v \, Cp \, (T_o - T_i) \qquad (kW)$$

$$Q_{v.L} = m_v \, hfg \, (w_o - w_i) \qquad (kW)$$

$$Q_{v.total} = m_v \, (h_o - h_i) \qquad (kW)$$

Where:

m_v: mass flow rate of ventilated air

w_i , wo: humidity ratio inside & outside the room

hfg: latent heat of vaporization of water ≈ 2450 kJ/kg

Ventilated air flow rate for persons:

$$V_v = (V_v \,/\, person) \text{ No. of persons}$$

$$m_v = V_v \, \rho = 1.2 \, V_v$$

Where:

(V_v / persons): quantity of ventilating air per person gated from table (10)

<u>Ex.6</u>

6-Heat gain from occupants:

The human body in a cooled space constitutes a cooling load of sensible & latent heat. The heat gain from occupants is based on the average number of people that are expected to be present in conditioned space and depends upon the activity of the person.

$$Q_{o.s} = (Q_s / \text{person}) \text{ No. of persons. CLF}$$

$$Q_{o.L} = (Q_L / \text{person}) \text{ No. of persons}$$

Where:

Q_{os} & $Q_{o.L}$ are the total sensible and latent heat gain from occupants.

(Q_s / person) & (Q_L / person): Amount of sensible & latent heat gain from one person from table (11)

CLF: cooling load factor from table (12)

Ex.7

7-Heat gain from appliances:

The appliances frequently used in air conditioned space may be electrical, gas fired or steam heated. Table (7) gives most of the commonly used appliances.

8-Heat gain due to processes:

The exist of processes in the conditioned space add the heat to space and lead to increase the humidity of air.

$$Q_p = Q_{ps} + Q_{p.L}$$

$$Q_{p.L} = m_v \; hfg$$

$Q_{p.L}$: Latent heat added

m_v: vapor generation rate (kg/s)

hfg: latent heat of vaporization (kJ/kg)

9-Heat gain from lighting equipments:

The heat gain from electric lights depends upon the rating of lights in watts, use factor & allowance factor.

Q_{lights} = power of lights * use factor * allowance factor

Use factor = 1 in residence, commercial stores and shops.

= 0.5 in industrial workshops.

Allowance factor = 1.25 (for fluorescent light)

Ex.8

10-Heat gain from power equipments:

The power equipments such as fan, motor or any other equipments of this type add heat in the air conditioned space.

$$Q_{eq} = \frac{power\ of\ motor(kw) * load\ factor * use\ factor}{motor\ efficiency}$$

Load factor: is the fraction of the total load at which the motor is working.

Load factor and use factor always taken as 1.

If the fan is located before the air conditioner the heat energy must be added to the load. If the fan is located after the air conditioner the energy is added to the room sensible heat load.

11-Heat gain through ducts:

The heat gain due to supply duct depends upon the temperature of air in the duct & temperature of the surrounding the duct.

$$Q_D = U\ A_D\ (Ta - Ts)$$

U: overall heat transfer coefficient.

A_D: surface area of duct.

Ta: Temperature of ambient air.

Ts : Temperature of supply air.

L7 (Heating Load)

❖ Heating load calculation:

Heating load includes the following main sources:

1- Walls & roofs.
2- Glass.
3- Infiltration.
4- Heat loss from adjacent unconditioned places.
5- Other heat required for humidification purposes or safety factor.

1-Walls & roofs:

Can be estimated from:

$$Q_{wall} = U_{wall\ or\ roof.}\ A_{wall\ or\ roof.}\ \Delta T$$

EX.1

2-Glass (windows):

The heat loss through the glass may be obtained from the following equation:

$$Q_{glass} = U_g\ A_g\ (T_i - T_o)$$

The windows area corresponds to the whole area of all windows including sashes and door. The value of T_i in the above equation correspond to the temperature at the mean high of the windows

EX.2

3- Infiltration:

Include the load required to worm the interring air through the cracks and clearance around windows and doors. It depends on tightness of the construction and wind velocity.

$$Q_{i.s} = m_i C_p (T_i - T_o) \quad (w)$$

$$Q_{i.L} = m_i hfg (w_i - w_o) \quad (w)$$

EX.3

4-heat loss from adjacent unconditioned spaces:

$$Q = U A (T_i - T_p)$$

$$T_p = T_i - \frac{1}{2} (T_i - T_o)$$

- في حالة الغرف المجاورة التي تحتوي على مصادر حرارية غير معتادة مثل وجود الطباخات أو لوندري يضاف حوالي (5 – 10) درجات على درجة حرارة الهواء الخارجي للحصول على درجة حرارة الغرفة المجاورة. $T_p = T_o + (5 \text{ to } 10)$
- بالنسبة للأرضيات التي تقع مباشرة على التربة أو أرضيات السراديب تفرض درجة حرارة التربة مابين (10 إلى 14) درجة.

5-Other heat required for humidification purposes or safety factor:

إن مكونات الجزء الخامس من الحمل الحراري صعبة التحديد بصورة دقيقة إذ لا تجرى اعتياديا حسابات معينة للمفقودات الحرارية المتنوعة ولكن هناك بعض الأمور الظاهرة الواجب اعتبارها والتي تدخل ضمن هذا الحقل مثل معامل الأمان والحرارة اللازمة للترطيب. فمثلا إذا كانت البناية تقع في منطقة معرضة لتيارات هوائية دائمية يضاف من 10 إلى 20 بالمائة من حمل التدفئة المحسوب كمعامل أمان لتغطية ما هو غير منظور من فقدان حراري نتيجة التيارات الهوائية أو تعرض المنطقة لتغيرات جوية سيئة غير متوقعة.

❖ **Space heating**:

The load for heating the space is:

$$Q_{sens.} = \dot{m}\, Cp\, \Delta T$$

This equation is used to calculate air flow rate required for heating also used to estimate the amount of steam, hot water, warm air or electric energy of offset the heat losses.

L8 (Fluid flow through ducts and air distribution)

Classification of ducts:

1- Supply air duct
2- Return air duct
3- Fresh air duct
4- Low pressure duct
5- Medium pressure duct
6- High pressure duct
7- Low velocity duct
8- High velocity duct

Duct material:

The duct are usually made from galvanized iron sheet metal or aluminum sheet metal. Now the use of non metal ducts has increased.

Duct shape:

The ducts may be made in circular, rectangular or square shapes. From an economical point of view, the circular ducts are preferred because the circular shape can carry more air in less space.

Pressure in ducts:

The flow of air within a duct system is produced by the pressure difference existing between the different locations. The types of pressure in ducts are:

1- Static pressure.
2- Dynamic or velocity pressure.
3- Total pressure.

$P_T = P_s + P_d$

Continuity equation for ducts:

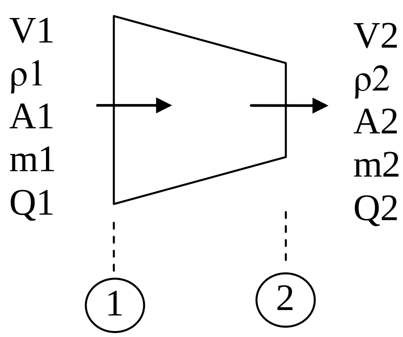


Fig.1

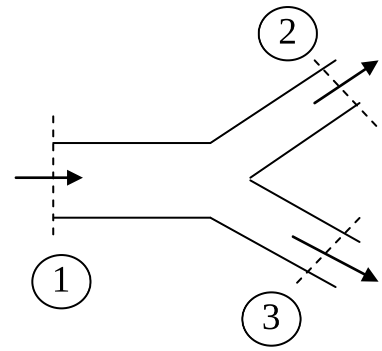


Fig.2

Mass flow rate of air through section (1)

$m_1 = \rho_1 Q_1 = \rho_1 A_1 V_1$

$Q_1 = A_1 V_1$

Mass flow rate of air through section (2)

$m_2 = \rho_2 Q_2 = \rho_2 A_2 V_2$

$Q_2 = A_2 V_2$

From continuity for fig. (1)

$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$

For air conditioning purposes the density of air is assumed constant (1.2 kg/m^2)

$\therefore Q1 = Q2$ (continuity equation)

$A1V1 = A2V2$

For branched duct as in fig. (2)

$m_1 = m_2 + m_3$

$\rho_1 Q_1 = \rho_2 Q_2 + \rho_3 Q_3$

$\therefore Q_1 = Q_2 + Q_3$

Bernoulli's Eq. for duct:

For flow between section (1) & (2) in duct.

$Ps_1 + \frac{\rho 1 v_1^2}{2} + \rho_1 g z_1 = Ps_2 + \frac{\rho 2 v_2^2}{2} + \rho_2 g z_2$

For $\rho 1 = \rho 2$ & $z_1 = z_2$ (at same level)

$Ps_1 + \frac{\rho v_1^2}{2} = Ps_2 + \frac{\rho v_2^2}{2}$

Or $Ps_1 + Pd_1 = Ps_2 + Pd_2$

P_s: static gauge pressure.

For frictionless flow there is no pressure drop and

$P_{T1} = P_{T2}$

Where $P_{T1} = Ps_1 + Pd_1$

$P_{T2} = Ps_2 + Pd_2$

In actual practice there is always pressure drop in duct due to friction & other causes such as sudden changes in the cross- section & direction.

$P_{s1} + P_{d1} = P_{s2} + P_{d2} + \Delta p$

Where Δp: is the total pressure drop between two sections

Pressure loss in ducts:

The pressure is lost due to friction between the moving particles of fluid (air) and interior surfaces of duct. This loss is the friction loss. The pressure is also lost dynamically at changes of direction and cross section (dynamic loss).

$\Delta P = \Delta P_f + \Delta Pd$

Pressure Loss due to friction:

Pressure Loss due to friction may be obtained by Darcy's equation:

$\Delta P_f = f\frac{L}{Dh}\frac{\rho v2}{2}$

$= K_f\ \frac{\rho v2}{2}$

ΔP_f: pressure Loss due to friction (N/m^2)   (pa)

f: friction factor depend upon the surface (dimensionless) .

L: length of duct.(m)

v: mean velocity of air (m/s)

Dh: hydraulic diameter.

For circular duct ⇒ Dh = D

For noncircular duct ⇒ Dh = $\frac{4 * cross\ sectional\ area}{wetted\ permeter} = \frac{4\ A}{P}$

Kf: resistance coefficient or pressure Loss coefficient = $f\frac{L}{Dh}$

In air conditioning the pressure loss due to friction is expressed in mm of water.

$\Delta P_f = f\ \frac{L}{Dh} *\ P_d$ (in mm of water)

1 Pa /m = 0.1019 (mm of H2O /m)

<u>Friction factor (f):</u>

1- For laminar flow (Re < 2000)

$f = \frac{64}{Re}$

2- For turbulent flow and relative roughness < 0.001

$f = \frac{0.3164}{(Re)^{0.25}}$

where relative roughness = $\frac{\varepsilon}{D}$ (roughness factor)

ε:Absolute roughness of surface

D: Diameter of duct

3- for turbulent flow & rough ducts $\frac{\varepsilon}{D}$ > 0.001

f = $\frac{1}{\left[1.74 - 2\ \log\left(\frac{2\varepsilon}{D}\right)\right]^{\ 2}}$

Or f may be read directly from moody chart for different Re & $\frac{\varepsilon}{D}$

Table 16 shows absolute roughness values

Figures (6) & (7) show the standard chart for estimating frictional pressure drop in circular ducts made of GI at standard air conditions (T=20C̊ & PB =1 atm). When the conditions differ from standard conditions, the pressure drop must be corrected.

Δp_{corr} = Δp (figures 6 and 7) × KT (figure8) × KE (figure 8)

KT: correction factor for temperature

KE: correction factor for elevation.

Charts (6) & (7) is valid only for circular ducts for other shapes an equivalent diameter has to be used to estimate the frictional pressure drop.

Rectangular ducts:

Equivalent diameter: A rectangular duct is said to be equivalent to a circular duct if the volumetric flow rate Q & frictional pressure drop per unit length (ΔPf/L) are same for both.

$$D_{eq} = 1.3\ \frac{(a.b)^{0.625}}{(a+b)^{0.25}}$$

a,b: Two sides of rectangle

$\frac{a}{b}$: aspect ratio

Ex.1, Ex.2, Ex.3

Dynamic pressure Losses in ducts:

Dynamic pressure loss takes place whenever there is a change in either the velocity or direction of air flow due to the use of a variety of bends & fittings in A.C ducts.

$$\Delta P_d = c\ \frac{\rho v2}{2}$$

C: dynamic loss coefficient.

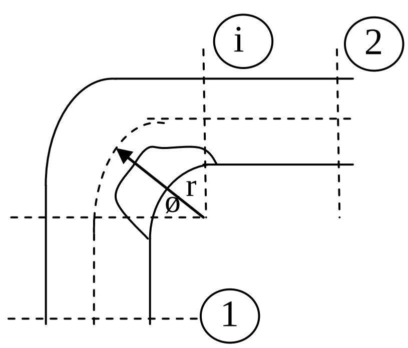
Sometimes an equivalent length L_{eq} is defined to estimate the dynamic pressure loss through bends & fittings.

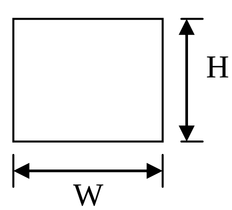
$$\Delta P_d = c\,\frac{\rho v2}{2} = f\ \frac{\mathrm{leq}}{D}\,\frac{\rho v2}{2}$$

$$L_{eq} = \frac{C.D}{f} \quad \text{or } c = \frac{f\ leq}{D}$$

Evaluation of dynamic pressure loss through various fittings:

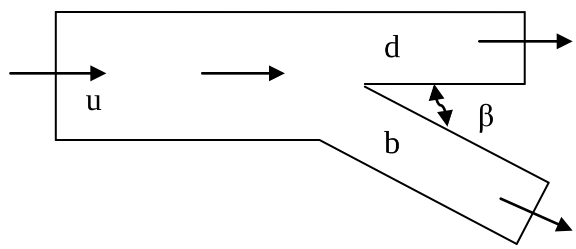
a-Turns, bends or elbows:





$$\Delta P_{d,b} = Cb\ (\frac{\rho v^2}{2}) = f\ ((H/w),\ r)\ \ (\frac{\rho v^2}{2})$$

b-Branch take- offs:



The dynamic pressure drop from the upstream (u) to downstream (d), $\Delta P_{u\text{-}d}$ is given by:

$$\Delta P_{u\text{-}d} = 0.4\ (\frac{\rho v_d^2}{2})\ (1-\frac{vd}{vu})^2 = C_{u\text{-}d}\ (\frac{\rho v_u^2}{2}))$$

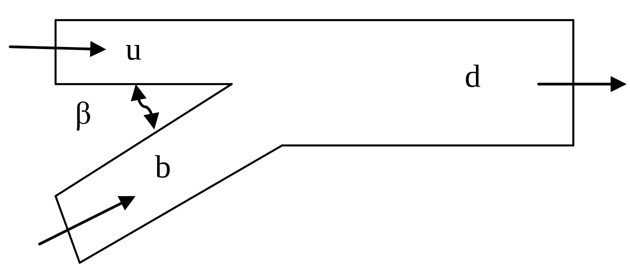
The dynamic pressure drop from the upstream (u) to branch (b)

$$\Delta P_{u\text{-}b} = C_{u\text{-}b}\ (\frac{\rho v_u^2}{2}))$$

Where v_d: air velocity in the down stream

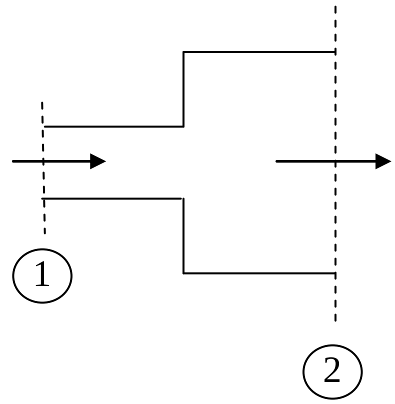
v_u: air velocity in the up stream

c-Branch entries:



Equations & parameters as in branch take- offs

d-Sudden enlargement:

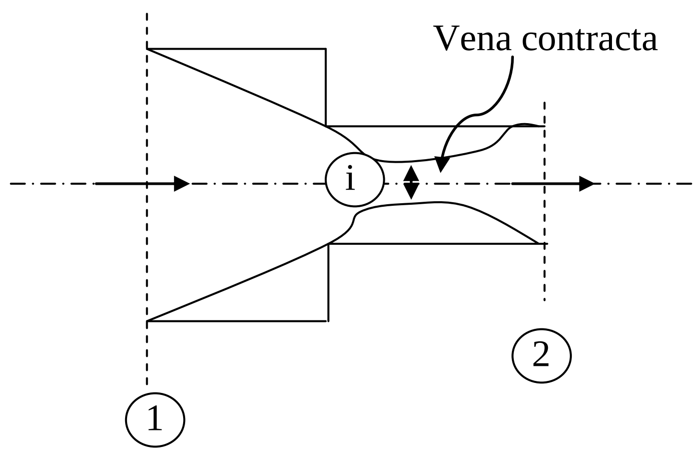


$\Delta P_{d,enl} = \left(\frac{\rho\, v_1^2}{2}\right)\left(1 - \frac{A1}{A2}\right)^2$

Where: v_1: velocity before enlargement

A_1,A_2: areas before & after enlargement

e-Sudden contraction:



$$\Delta P_{d,con} = \left(\frac{\rho v_2^2}{2}\right)\left(\frac{A2}{A_1'} - 1\right)^2 = \left(\frac{\rho\, v_2^2}{2}\right)\left(\frac{1}{Cc} - 1\right)$$

The coefficient Cc is known as contraction coefficient and is equal to A'1/A2 and is found to be a function of the area ratio A2/A1.

A2/A1	Cc
0.1	0.624
0.5	0.681
0.8	0.813
1	1

To over-come the fluid friction & the resulting head a fan is required in A.C. systems. When a fan is introduced into the duct the static & total pressure at the section where the fan is located rise. This rise is called fan total pressure (FT P).

Duct design:

The purpose of the duct design is to select suitable dimensions of duct for each run and then to select a fan which can provide the required supply air flow rate to each conditioned zones

Duct design methods:

1- velocity method.
2- Equal pressure drop method (equal friction loss method)
3- Static regain method.

1-Velocity method:

The velocity method is one of the simplest ways of design the duct system for both supply & return air. However the application of this method requires selection of suitable velocities in different duct runs, which requires experience .

The various steps involved in this method are:

1- Select suitable velocities in the main & branch ducts

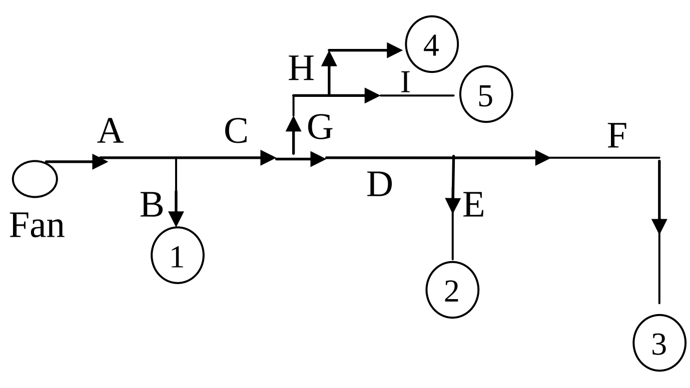
2- Find the diameter of main & branch ducts from air flow rates and velocity for circular ducts. For rectangular ducts find the cross-sectional area from flow rate & velocity and then be fixing the aspect ratio. Find the two sides of the rectangular duct Q = V.A
3- From the velocities & duct dimensions find the frictional pressure drop for ducts using friction chart or equation.
4- From the duct layout, dimensions and air flow rates find the dynamic pressure losses for all the bends & fittings
5- Select a fan that can provide sufficient FT P.

Ex.4

2-Equal friction

Is a simple and most widely used conventional method. This method usually yields a better design than the velocity and it is suitable when the duct is not too long and used for both supply & return ducts.

In this method the frictional pressure drop per unit length in the main & branch ducts (ΔP_f / L) are kept same for example.



The steps are as following:

1- Select a suitable frictional pressure drop per unit length (ΔP_f/L)

$(\frac{\Delta Pf}{L})A = (\frac{\Delta Pf}{L})B = (\frac{\Delta Pf}{L})C$……..

2- Then the equivalent diameter of the main duct (A) is obtained from the selected value of ($\Delta P_f/L$) and the air flow rate. Air flow rate in main duct QA is equal to the sum of air flow rates to all the conditioned zones

QA = $\sum_{i=1}^{N} Qi$

From the air flow rate & ($\Delta P_f/L$) the equivalent diameter of the main duct ($D_{eq.A}$) can be obtained either from the friction chart or using the equation.

$$D_{eq.A} = \left[\frac{0.022243\, Q_A^{1.852}}{(\frac{\Delta Pf}{L})_A}\right]^{(\frac{1}{4.973})}$$

$D_{eq.A} = \frac{0.02243qa}{\left(\frac{\Delta pf}{L}\right)A}$

3- Since the frictional pressure drop per unit length is same for all the duct runs, the equivalent diameter of the duct runs obtained from equation

$(\frac{Q^{1.852}}{Deq^{4.973}})_A = (\frac{Q^{1.852}}{Deq^{4.973}})_B = \ldots\ldots\ldots.$

4- If the duct is rectangular then two sides of the rectangular duct of each run are obtained from the equivalent diameter of that run and by fixing aspect ratio. Thus the dimensions of the all duct runs can be obtained. The velocity of air through each duct is obtained from the Q &A.

5- Next from the dimensions of ducts in each run the total frictional pressure drop.

$\Delta P_f = (\frac{\Delta pf}{L}) * L$

6- Next the dynamic pressure losses in each duct run are obtained.

7- The total pressure drop in each duct run is obtained

$\Delta P_T = \Delta P_f + \Delta P_d$

8- The fan is selected to suit the index run with the highest pressure loss.

Dampering required = FTP - ΔP_T (run)

<u>Ex.5, Ex.6, Ex.7, Ex.8</u>

<u>3-Static regain method:</u>

This method is commonly used for high velocity systems with long duct runs.

1- velocity in the main duct leaving the fan is selected first
2- velocities in each successive runs are reduced such that gain in static pressure due to reduction in velocity pressure equals the frictional pressure drop in the next duct section. Thus the static pressure before each terminal or branch is maintained constant.

L 9 (Fans)

Fans:

A fan is a kind of pump which is used for pumping or circulating the air through the entire duct system

The air feed into a fan is called induced draft while the air exhaust from a fan is called forced draft.

Types of fans:

1- Centrifugal or radial flow fans.
2- Axial flow fan.

The centrifugal fans or widely used for ducts of A.C system. Because they can efficiently move large or small quantities of air over a greater range of operating pressure.

Total pressure developed by fan:

As mentioned earlier the total pressure created by a fan is F T P which is:

$F\,T\,P = P_{T2} - P_{T1}$

Where

P_{T2}: total pressure at fan outlet = $P_{s2} + P_{d2}$

P_{T1}: total pressure at fan inlet = $P_{s1} + P_{d1}$

Fan static pressure $FSP = FTP - P_{d2}$

$$FTP = FSP + FdP$$

Fan air power:

The power output of a fan is expressed in terms of air power and represents the work done by the fan.

P.P = Q * FTP * KP

where : P.P: pumping power (w)

Q: volumetric flow rate (m^3/s)

FTP: fan total presser (pa)

Kp : compressibility coefficient

When FTP in mm of water

P.P = 9.81* Q * FTP * KP (W)

and static fan air power

$P_{a.s}$= Q * FSP * KP

Fan efficiency:

Is the ratio of the total fan air power or (pumping power) to the driving power. It is also called mechanical efficiency.

$$\eta_{TF} = \frac{P.P}{Input\ or\ brake\ power\ (B.P)}$$

The required power input to the fan is

$$P_{in} = \frac{P.P}{\eta_{TF}}$$

Similar the static fan efficiency

$$\eta_{SF} = \frac{Static\ fan\ air\ power\ (Pas)}{Input\ or\ brake\ power\ (B.P)}$$

Ex.9, Ex.10

Fan performance curves:

Is a graph of a fan volume flow rate plotted against pressure, power, or efficiency

Specific speed of a centrifugal fan:

Is the speed of a geometrically similar fan which would deliver 1 m^3 of air per second against a head of 1m of air. It is usually denoted by Ns.

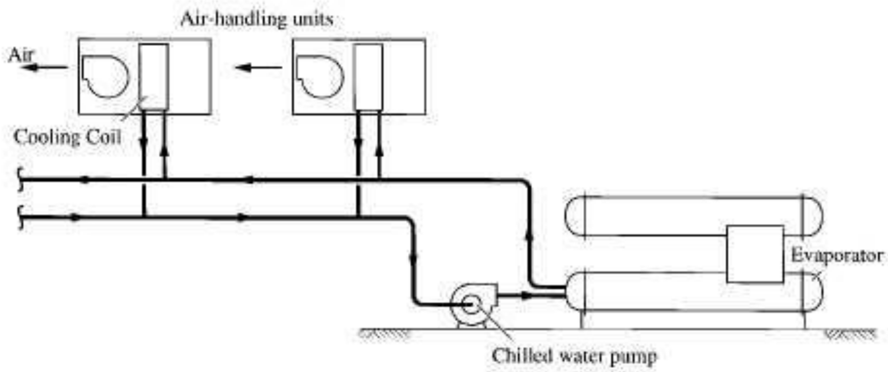
Ex9, ex10

L10 (DESIGN OF PIPING SYSTEMS)

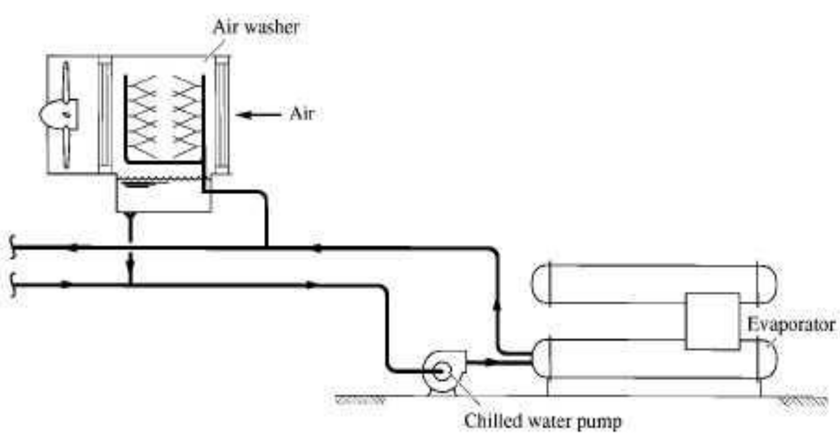
Types of piping system

The piping systems are divided into two types:

Closed system: In a closed system chilled or hot water flowing through the coils, heater, chiller, boiler or other heat exchanger and forms a closed recirculation loop as shown in the figure below. In close system water is not exposed to the atmospheric during its flowing process.



Open system: In an open system the water is expose to the atmosphere as shown in the figure below. For example, chilled water come directly into contact with the cooled and dehumidified air in the air washer and condenser water is exposed to atmosphere in the cooling tower.

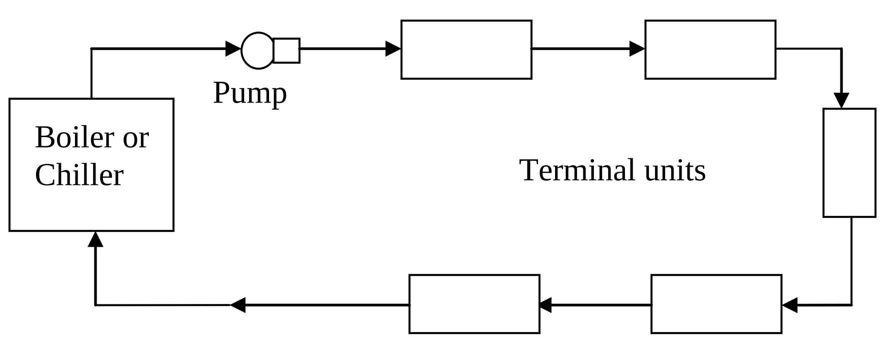


The close systems are consists of the following components:

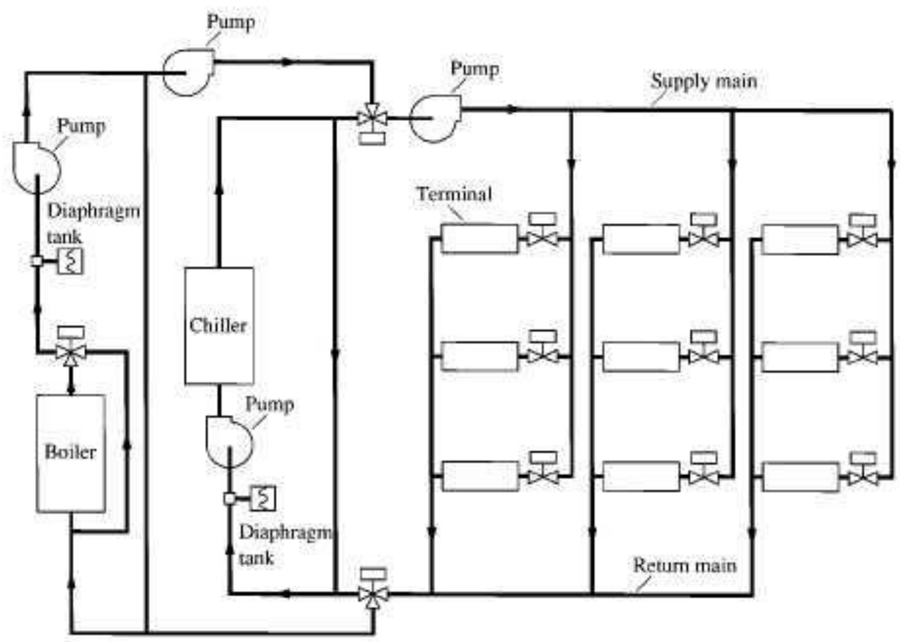
1- ***Load unit:*** which represents the terminal unite as cooling or heating coils or radiators.
2- ***Source unit:*** which represent the chiller in cooling system or the boiler and furnace in heating systems.
3- ***Distribution systems:*** which represents the piping and fitting of the piping systems.
4- ***Pump:*** that used to circulate the water in the cooling or heating systems . It is usually of a centrifugal types with constant flow rates and appropriate pressures.
5- ***Expansion tanks:*** which are of two types (open and closed tanks).

<u>Types of closed systems:</u>

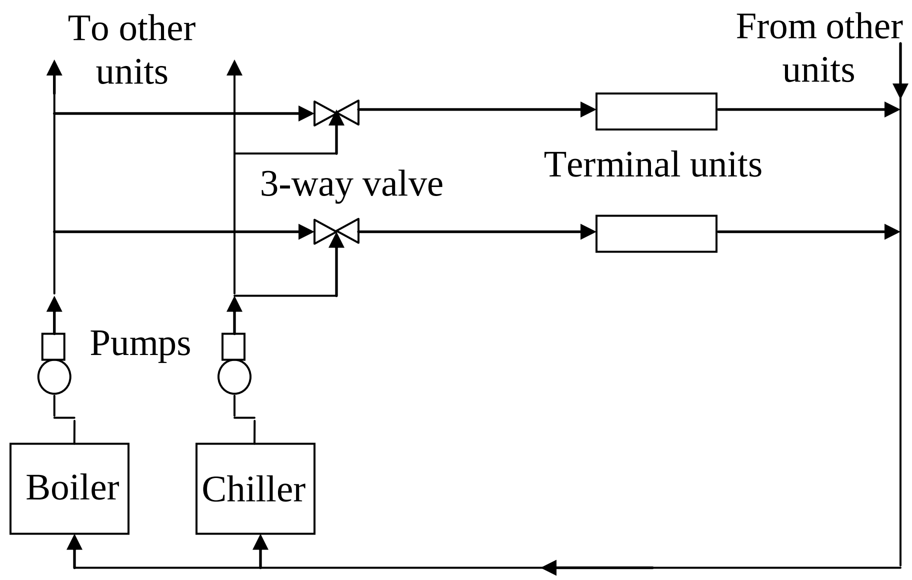
1- ***One pipe system:*** A single pipe connect all the system components i. e. the pipe started from the source unit through the pump to the load units and then return to the source. The disadvantage of this system is that the efficiency of the last units are low because the return cold or hot water of all units is added to the same pipe that supply the end units.



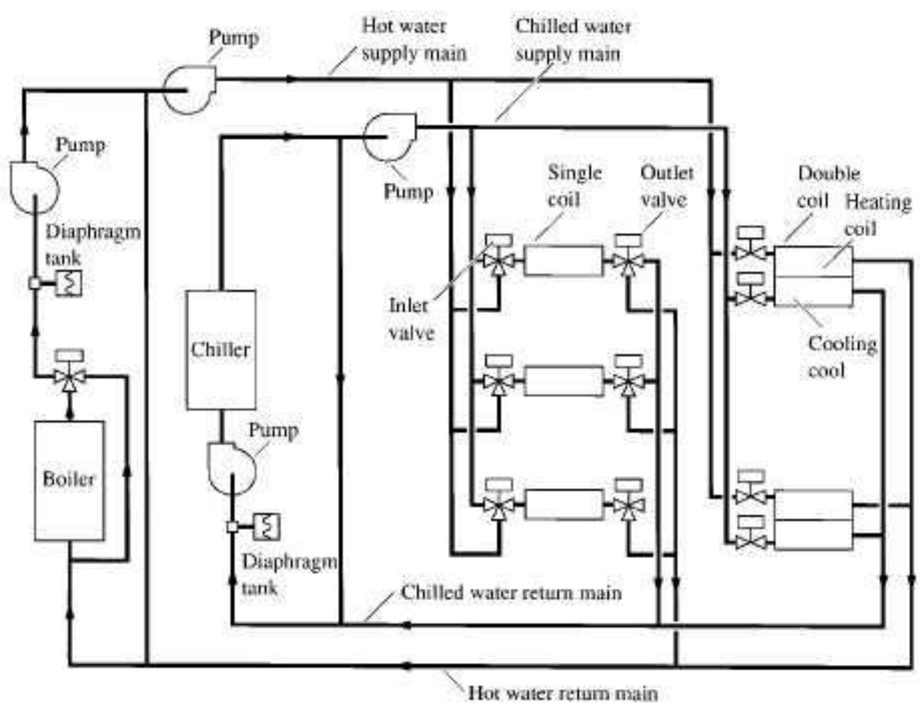
2- ***Two pipe system:*** This system has a two pipes one to the supply water and the other to the return water. In this system the disadvantage of the one pipe system is overcome. This is the most popular system in use because it is simple and cheap.



3- ***Three pipe system:*** This system can be use in central air conditioning units that used for cooling and heating in the same time. It has one pipe to supply hot water, the other to supply cold water and the third is a common return pipe i. e. the third pipe is used to return cold and hot water to the chiller and boiler. The disadvantage of the system is the waste of heat in the third common return pipe.



4- ***Four pipe system:*** The disadvantage of the three pipe system (i. e. the common third return pipe) is overcome in this system by adding a fourth pipe . The four pipe system can be used in central air conditioning plant with cold and hot circuits separated as shown in the figure below .

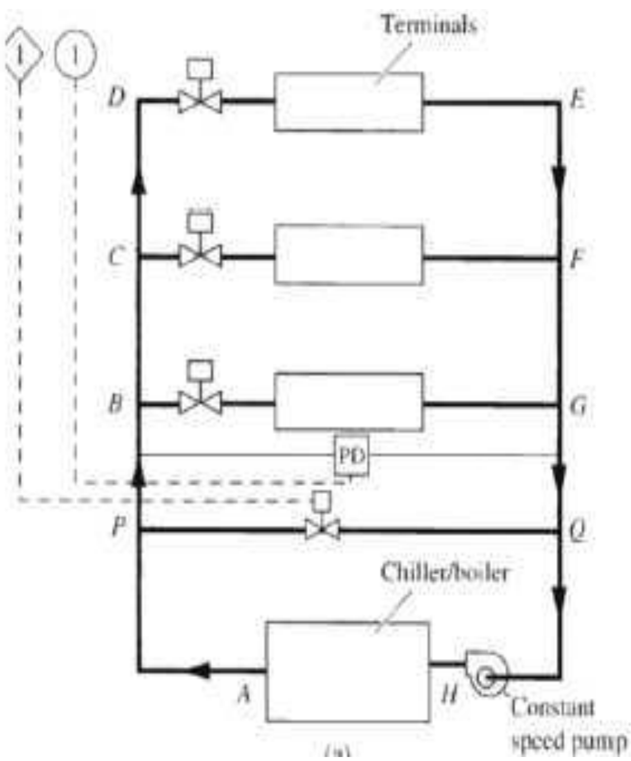


Arrangement of return pipe :

The closed piping systems can also be classified according to the arrangement of the return pipe to (Direct return systems and Reverse return systems):

In a ***direct return water system***: the various branch piping circuits such as ABGHA and ABCFGHA are not equal in length (figure below a- direct return, b- reverse return). Careful balance is often required to establish the design flow rates for a building loop when a direct return distribution loop is used .

In a ***reverse return system:*** the piping length for each branch circuits ,including the main and branch pipes are almost equal.



Procedure for sizing pipe systems:

The recommended procedure for sizing piping systems is outlined below:

1- Sketch the main lines and branches and indicate the locations of terminal units and the rate of flow of each unit. Use as short as possible runes .

2- Choose a suitable velocity in the main pipe or riser (1.0 - 2.5 m/s), and 1.25 m/s for branch pipes for D = 50 mm or less. The velocity may be greater than 2.5 m\s for large pipes.

3- Point out the locations of valves drainage and air vent openings. The drainage should be located at the lowest point while the air- vent should be at the highest point in the system.

4- Design the pipe sizes using charts and tables. Do not use an equal pressure drop as in duct system.

5- Determine the equivalent length for the main pipes branches, fittings ,coils heat exchangers plus any static head given in open circuits.

6- Calculate the pump total head or total pressure and the pump power required to deliver the required flow rate. Always use a stand by pump for emergency.

Ex.1, Ex.2

Water pumps:

The total head of a given pump may be determined by:

$$H_{pump} = [\ H_2 + 0.5 *(V_2)^2/g\ ] - [\ H_1 + 0.5 * (V_1)^2/g\ ]$$

Where H_{pump} in meter of water and subscripts (2) for discharge and (1) for suction sides.

Multiplying by ρg

$P_{pump} = [P_2 + 0.5 *\rho *(V_2)^2] - [P_1 + 0.5 * \rho *(V_1)^2]$

Where P_{pump} in Pascal and subscripts (2) for discharge and (1) for suction sides.

The power of the pump may be given by:

$P.P_{pump} = m * g * H_{pump} = Q * P_{pump}$ (W)

The pump efficiency may be given by:

$\eta = P.P_{pump} / P_{in}$

Where P_{in} is the input power of the pump (shaft power).

Ex.3, Ex.4

L11 (Refrigeration)

Refrigerator & heat pumps:

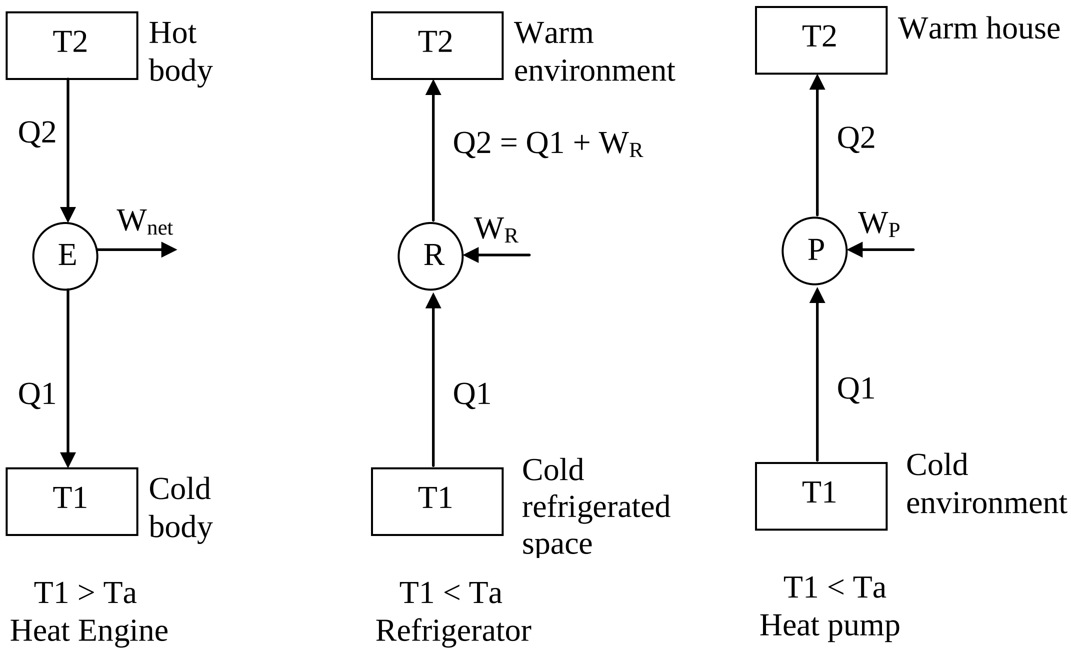
The transfer of heat from a low temperature region to a high temperature one requires special devices called refrigerator.

Refrigerator: is a cyclic device & the working fluids used in the refrigeration cycles are called refrigerants.

As shown in figure bellow the Q1 is the magnitude of the heat removed from the refrigerated space at temperature T1 and Q2 is the magnitude of the heat rejected to the warm space at temperature T2 and W_{net} is the net work input to the refrigerator.

Another device that transfer heat from a low- temperature medium to a high- temperature one is the heat pump. Refrigerators & heat pumps are the same devices they differ in their objectives only. The objective of refrigerator is to maintain the refrigerated space at low temperature by removing heat from it. The objective of a heat pump is to maintain a heated space at a high temperature. This is accomplished by absorbing heat from a low- temperature source such as cold outside air in winter & supplying this heat to a warmer medium such as a house.

The performance of refrigerators & heat pump is expressed in terms of the coefficient of performance (COP)



$$COP_R = \frac{Desired\ out\ put}{Required\ input} = \frac{cooling\ effect}{work\ \ in\ put} = \frac{QL}{wnet} = \frac{Q1}{Q2-Q1}$$

$$COP_{HP} = \frac{Desired\ out\ put}{Required\ input} = \frac{heating\ effect}{work\ \ in\ put} = \frac{QH}{wnet} = \frac{Q2}{Q2-Q1}$$

W_{net} = Q2 - Q1

$COP_{HP} = COP_R + 1$

Where

$$\eta_E = \frac{work\ done}{heat\ supplied} = \frac{Q2-Q1}{Q2}$$

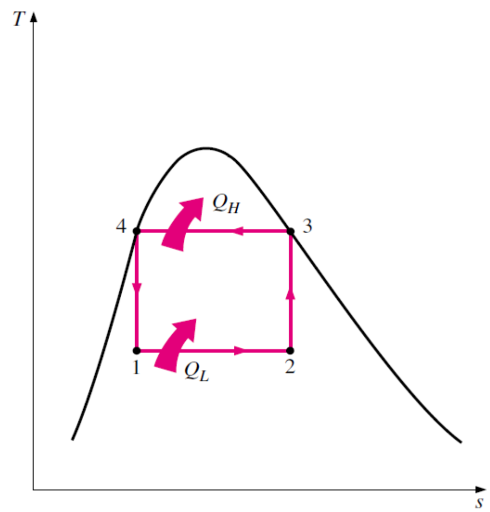
The cooling capacity of refrigeration system which is the rate of heat removal from the refrigerated space is often expressed in terms of Ton of Refrigeration (TR).

TR: is the capacity of a refrigeration system that can freeze 1 ton of liquid water at 0 C^o into ice at 0 C^o in 24 h

1TR = 210 KJ / min = 12000 Btu / hr = 200 Btu/min

Carnot engine

The heat engine that operates most efficiently between high-temperature reservoir& a low- temperature reservoir is the Carnot engine. It is an ideal engine that uses reversible processes to form its cycle of operation. Thus it is also called a reversible engine



1→2 Isothermal expansion

2→3 adiabatic reversible expansion

3→4 Isothermal compression

4→1 adiabatic reversible compression

Applying the first law to the cycle.

$$W_{net} = Q2 - Q1$$

Thermal efficiency of Carnot cycle.

$$\eta = \frac{Q2-Q1}{Q2} = 1 - \frac{Q1}{Q2}$$

$$\eta = 1-\frac{T1}{T2}$$

We can see that the thermal efficiency of a Carnot engine is dependent only on the high & low absolute temperature of the reservoirs.

This relation of efficiency is applicable for all working substances or for all reversible engines regardless of the particular design characteristics.

<u>The reversed Carnot cycle:</u>

The Carnot engine when operated in reverse become a heat pump or refrigerator. Depending on the desired heat transfer. The COP for a heat pump becomes:

$$COP_{HP} = \frac{Q2}{W_{net}} = \frac{Q2}{Q2-Q1} = \frac{1}{1-T1/T2}$$

The COP of refrigerator becomes

$$COP_{R} = \frac{Q1}{W_{net}} = \frac{Q1}{Q2-Q1} = \frac{1}{\frac{T2}{T1}-1}$$

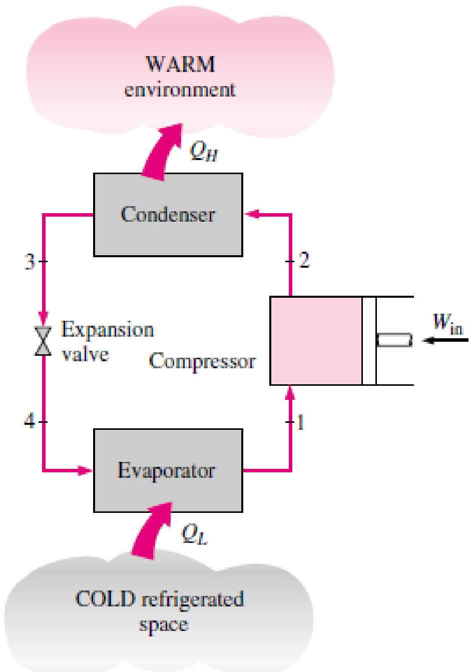
The a above measures of performance set limits that real devices can only approach. The reversible cycles assumed are unrealistic but the fact that we have limits which we know we cannot exceed is often very helpful in evaluating proposed design & determining the direction for further effort.

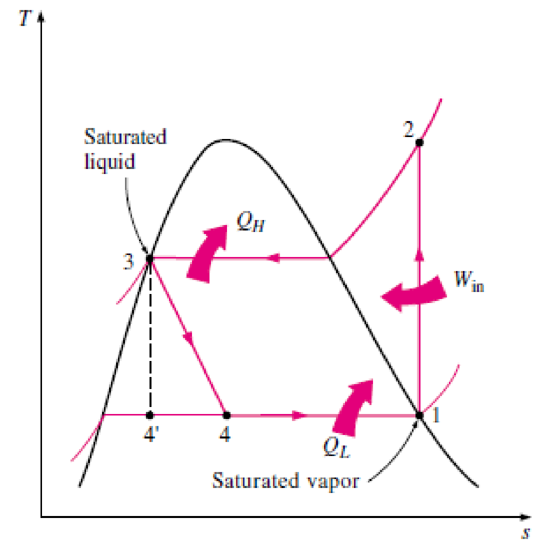
<u>Ex.1, Ex.2, Ex.3</u>

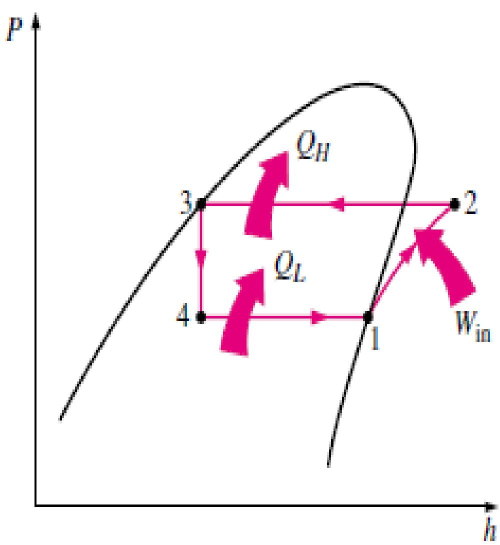
The ideal vapor compression refrigeration cycle:

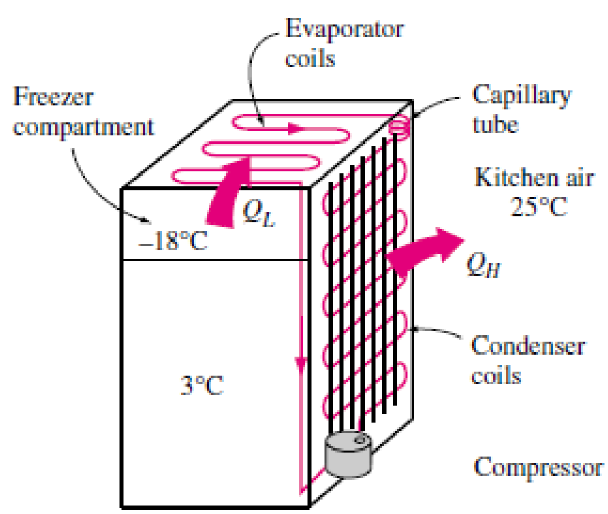
Is the most widely used cycle for refrigerators, air- conditioning systems and heat pump. It consists of four processes:

1-2 isentropic compression in a compressor.
2-3 constant pressure heat rejection in condenser
3-4 throttling in an expansion device
4-1 constant pressure heat absorption in an evaporator.









In an ideal vapor compression refrigeration cycle. The refrigerant enters the compressor at state 1 as saturated vapor & is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases during this isentropic compression process to well above the temperature of the surrounding medium. The refrigerant then enters the condenser as a superheated vapor at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings. The temperature of the refrigerant at this state is still above the temperature of the surrounding.

The saturated liquid refrigerant at state3 is throttled to the evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of refrigerant drops below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 4 as low quality saturated mixture and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapor and re enters the compressor, completing the cycle.

The steady flow energy equation per unit mass

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = he - hi$$

The condenser & evaporator do not involve any work and the compressor can be approximated as adiabatic.

$$COP_R = \frac{qL}{w_{in}} = \frac{h1-h4}{h2-h1}$$

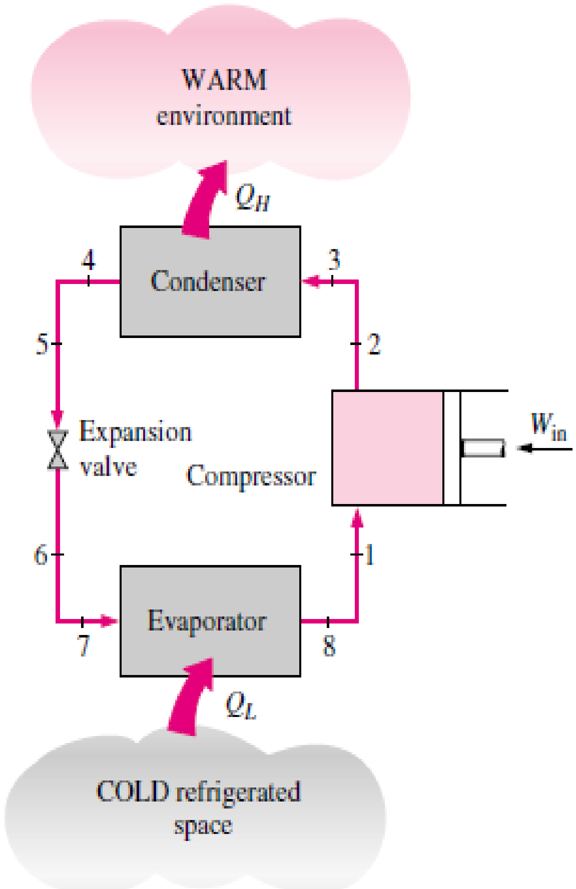
$$COP_{HP} = \frac{qH}{w_{in}} = \frac{h2-h3}{h2-h1}$$

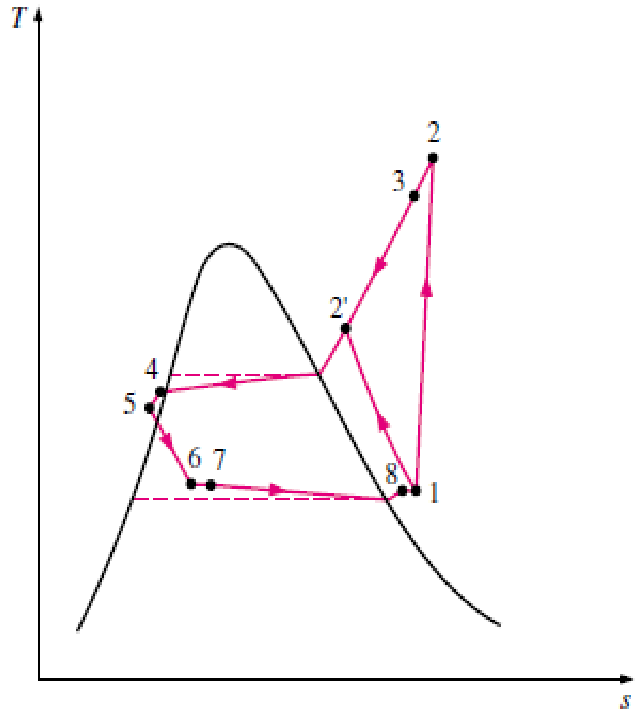
h1 = hg at P1 & h3 = hf at P3

<u>Ex.5, Ex.6, Ex.7</u>

<u>Actual vapor compression refrigeration cycle:</u>

An actual vapor compression cycle differs from the ideal one in several ways, mostly to the irreversibilities that occur in various components. Two common sources of ireversibilities are fluid friction (causes pressure drop) & heat transfer to or from the surrounding.





Compound vapor compression Refrigeration system:

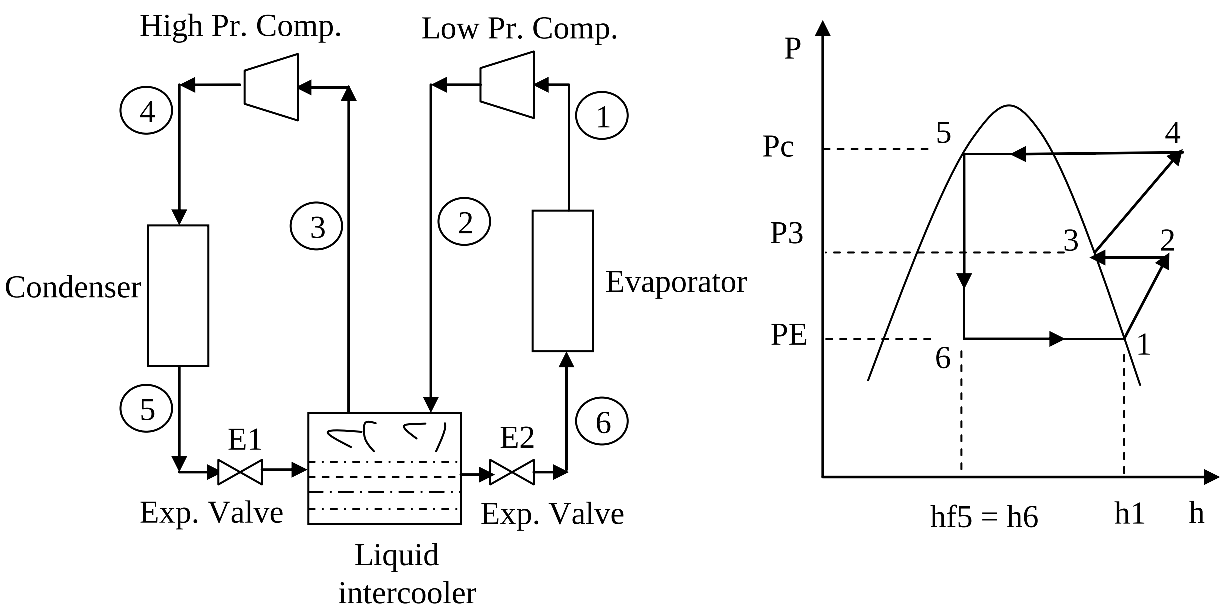
Sometimes the vapor refrigerant is required to be delivered at a very high pressure.

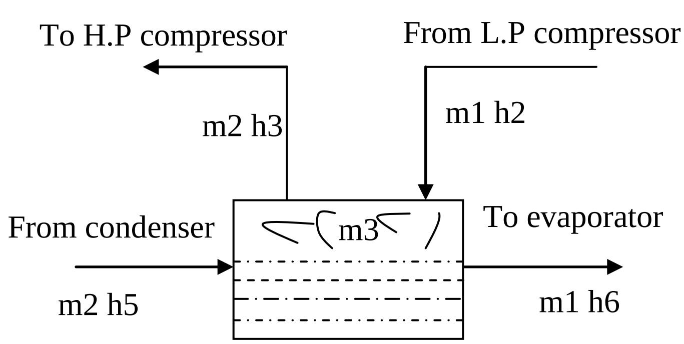
In such cases either we should compress the vapor refrigerant by employing single stage compressor with a very high pressure ratio or compress it in two or more compressors placed in series. The compression carried out in two or more compressors is called compound or multistage compression.

The COP can be increased either by increasing the refrigerating effect or by decreasing the compression work. The compression work is greatly reduced if the refrigerant is compressed very close to the saturated vapor line.

This can be achieved by compressing the refrigerant in more stages with intermediate inter-cooling.

Two stage compression with liquid inter cooler:





m1: mass of refrigerant passing through the evaporator (low pressure Compressor) (kg/min)

m2: mass of refrigerant passing through the condenser (high pressure Compressor) (kg/min)

m3: mass of liquid evaporated in the inter cooler

m3 = m2 - m1

The value of m2 may obtained by considering the thermal equilibrium for the liquid inter cooler

m2 h5 + m1 h2 = m1 h6 + m 2h3

$\therefore m2 = \frac{m1(h2-h6)}{h3-h5} = \frac{m1(h2-hf5)}{h3-hf5}$

The mass of liquid evaporated in the intercooler

$m3 = m2 - m1 = \frac{m1(h2-h3)}{h3-hf5}$

The refrigerating effect is:

RE = Ql = m1 (h1 – hf5)

Total work done in both the compressors

W = (h2 - h1) + (h4 - h3) (KJ)

Power required driving the system

P = m1 (h2 - h1) + m2 (h4 - h3) (kw)

$COP = \frac{QL}{W} = \frac{m1(h1-hf5)}{m1(h2-h1)+m2(h4-h3)}$

L12 (Refrigerants)

Refrigerant:

Is a heat carrying medium operating during a closed cycle. In the refrigeration system the refrigerant absorb heat from a low temperature system & discard it to a higher temperature system.

The suitability of a refrigerant for a certain application is determined by its physical, thermodynamic, & chemical properties also by various practical factors. A refrigerant which has greater advantages & less disadvantages is chosen.

Desirable properties of an ideal refrigerant:

1- low boiling point.
2- High latent heat of vaporization.
3- Low specific heat of liquid.
4- Low specific volume of vapor.
5- Non corrosive to metal.
6- Non flammable & non explosive.
7- Non toxic.
8- Low cost.
9- Easy to liquefy at moderate pressure & temperature.
10- Easy to locating leaks by odors or suitable indicator.

Classification of refrigerants:

The refrigerants may be classified into:

1- Primary refrigerant.

2- Secondary refrigerant.

The primary refrigerant is the fluid which is used directly as working fluid and it is classified into:

1- Halo - carbon refrigerants.

2- Azeotrope refrigerants.

3- Inorganic refrigerants.

4- Hydro- carbon refrigerants.

The secondary refrigerant: it is the refrigerant first cooled by primary refrigerant & then used for cooling purposes.

Halo- carbon refrigerant:

ASHRAE identifies 42 halo- carbon compounds as a refrigerants but only a few of them are commonly used. The halo- carbon compounds are all synthetically produced.

Azeotrope refrigerant:

The term azeotrope refer to a stable mixture of refrigerants whose vapor & liquid phases retain identical compositions over a wide range of temperature.

Inorganic refrigerant:

It is exclusively used before the introduction of halo- carbon refrigerant. These refrigerants are still in use due to their thermodynamic & physical properties.

Hydro- carbon refrigerant.

Most of the hydro- carbon refrigerants are successfully used in industrial & commercial installations. They posses satisfactory thermodynamic properties but are highly flammable & explosive.

<u>Designation system for refrigerants:</u>

The refrigerant are internationally designated as "R" followed by certain number such as R-11, R-114. A refrigerant followed by two digit No. indicates that a refrigerant is derived from methane base while three digit No. represents ethane base.

The numbers assigned to hydro- carbon & halo carbon refrigerant are:

- The first digit from right is the No. of fluorine (F) atoms.
- The second digit from right is one more than the No of hydrogen (H) atoms.
- Third digit from right is one less than the No. of carbon (C) atoms

The general chemical formula for refrigerant either for methane or ethane base is:

$$C_m\, H_n\, Cl_p\, F_q$$

where

m: No of carbon atoms

n: No of hydrogen atoms

p: No of chlorine atoms

q: No of fluorine atoms

The No. of refrigerant is:

$$R(m - 1)(n + 1)(q)$$

L13 (Refrigeration system components)

A typical refrigeration system consists of several basic components such as compressors, condensers, expansion devices and evaporators in addition to several accessories such as filters. It is essential to study the design and performance characteristics of individual components.

Compressors:

Is the most important & often the costliest component (typically 30 to 40 percent of total cost) of any vapor compression refrigeration system. The functions of a compressor is to continuously draw the refrigerant vapour from the evaporator so that a low pressure & low temperature can be maintained in the evaporator at which the refrigerant can boil extracting heat from the refrigerated space.

Classification of compressors:

The compressors may be classified in many ways as follow

1- According to the method of compression
   a- Reciprocating compressors
   b- Rotary compressors
   c- Centrifugal compressors
2- According to the No. of working strokes:
   a- Single acting compressors
   b- Double acting compressors.
3- According to the No. of stages
   a- Single stage compressors.

b- Multi stage compressors.

4- According to the method of drive employed

a- Direct drive compressors.

b- Belt drive compressors.

5- According to the location of the prime mover:

a- semi- hermetic compressors (direct drive, motor & compressor in separate housing)

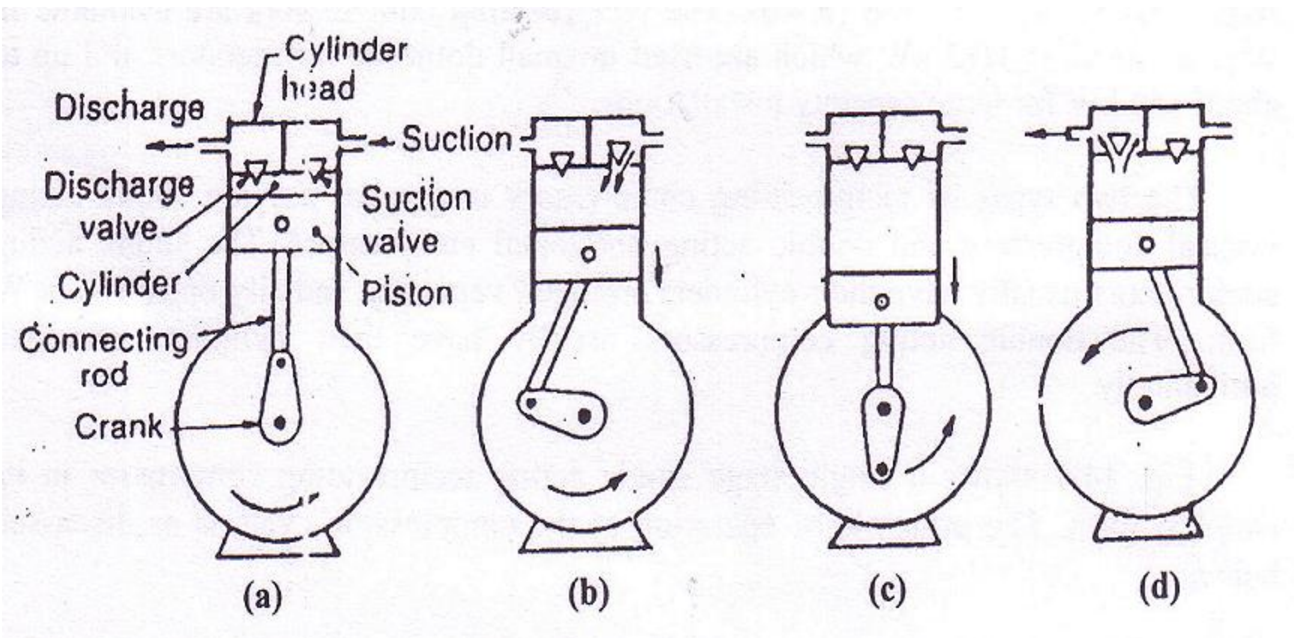
b- hermetic compressors (direct drive, motor & compressor in same housing.

Reciprocating compressor:

It is the compressor in which the vapor refrigerant is compressed by the reciprocating motion of the piston. These compressors are used for refrigerant which have comparatively low volume per kg & large differential pressure such as ammonia R-717,R-12, R-22. These compressors are available in small size which are used in small domestic refrigerators & large size for large capacity installations.

There are two types of reciprocating compressors in general use which are single acting vertical compressors & double acting horizontal compressors.

Principle of operating of single stage reciprocating compressor:



Important definitions:

Piston displacement volume or stroke volume:

It is the volume swept by the piston when it moves from its top or inner dead position to bottom or outer dead center position.

$$Vp = \frac{\pi}{4} D^2 L$$

D: diameter of cylinder.

L: length of piston stroke

Clearance factor:

It is the ratio of clearance volume (Vc) to the piston displacement volume (Vp)

$$C = \frac{vc}{vp}$$

Compressor capacity:

It is the volume of the actual amount of refrigerant passing through the compressor in a unit time it is equal to the suction volume (Vs) and expressed in (m^3/s)

Volumetric efficiency:

It is the ratio of the compressor capacity or the suction volume (Vs) to the piston displacement volume (Vp).

$$\eta_v = \frac{vs}{vp}$$

Suction pressure:

It is the a absolute pressure of refrigerant at the inlet of a compressor.

Discharge pressure:

It is the absolute pressure of refrigerant at the outlet of a compressor.

Compression ratio (pressure Ratio):

It is the ratio of absolute discharge pressure to the absolute suction pressure. Since the absolute discharge pressure is always more than the absolute suction pressure, therefore the value of compression ratio is more than unity.

Suction volume:

It is the volume of refrigerant sucked by the compressor during its suction stroke.

EX12:

L 14 (Absorption refrigeration systems)

The absorption refrigeration system is one of the oldest method of producing refrigeration effect. The principle of vapor absorption was first discovery by Michael faraday in 1824.

This system may be used in both the domestic & large industrial refrigerating plants. The refrigerants commonly used in a vapour absorption system are ammonia & water.

The vapor absorption system uses heat energy instead of mechanical energy as in vapor compression system.

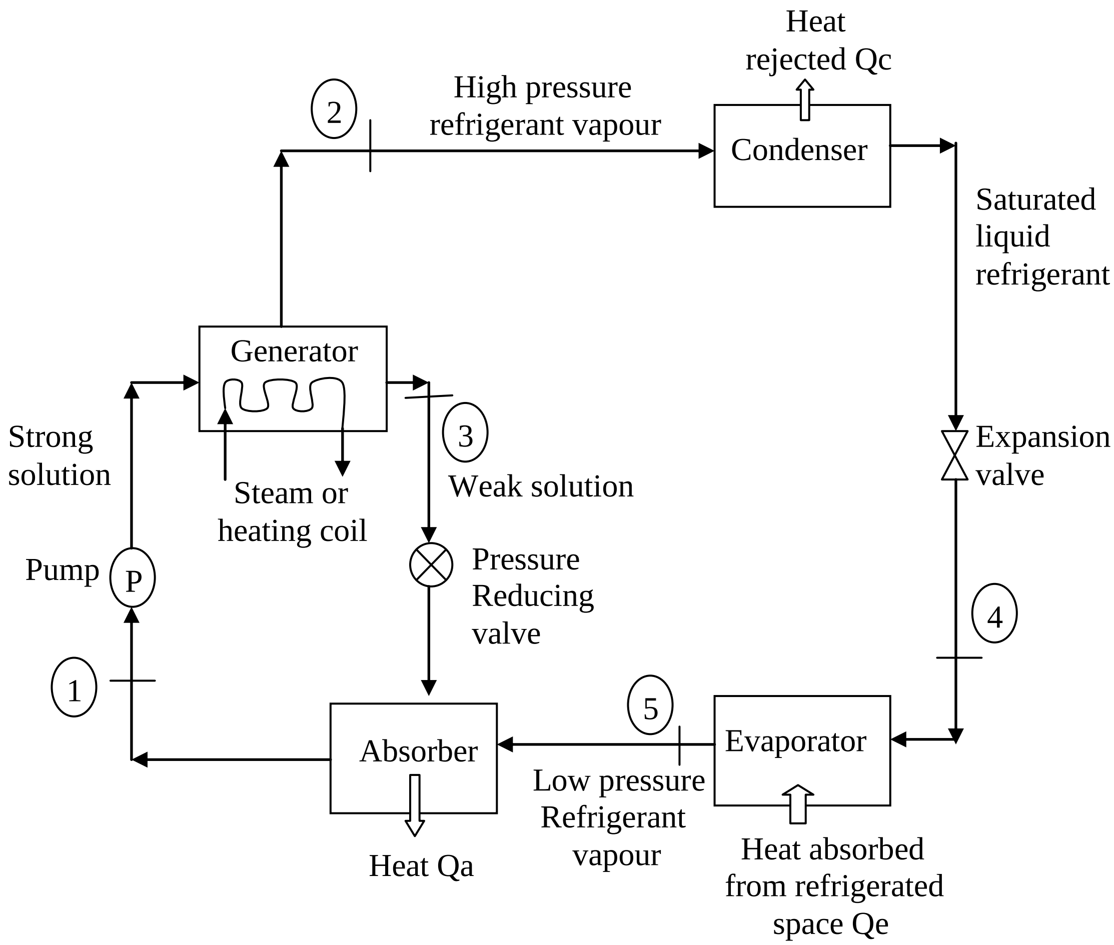
In the vapor absorption system, the compressor is replaced by an absorber, a pump, a generator & a pressure reducing valve.

Simple absorption refrigeration system:

The refrigerant vapor leaving the evaporator is absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The refrigerant in water solution is pumped to the higher pressure & is heated in the generator. Due to increasing the temperature of solution in generator as a result of adding heat, most of the refrigerant is evaporated & removed from the solution. The vapor then passes to the condenser & the weak solution is returned to the absorber.

The coefficient of performance of the absorption refrigeration system is

$$\text{COP} = \frac{heat\ absorbed\ in\ evaporator}{work\ done\ by\ pump + heat\ supplied\ in\ generator}$$



Advantages of absorption system over compression system:

1- In absorption system the only moving parts of the entire system is a pump which has a small motor. Thus the operation of this system is essentially quit & need less power.
2- The absorption system uses heat energy to change the condition of the refrigerant from evaporator while the compression system uses mechanical energy.
3- The absorption system designed to use steam, solar energy & other heat sources. Thus it can be used where the electric power is difficult to obtain or is very expensive.

4- The load variation does not effect the performance of absorption system. While the performance of compression system at partial loads is poor.

5- The absorption systems can be built in capacities well above 1000 tons of refrigeration each which is largest size for single compressor units.

Lithium bromide absorption refrigeration system:

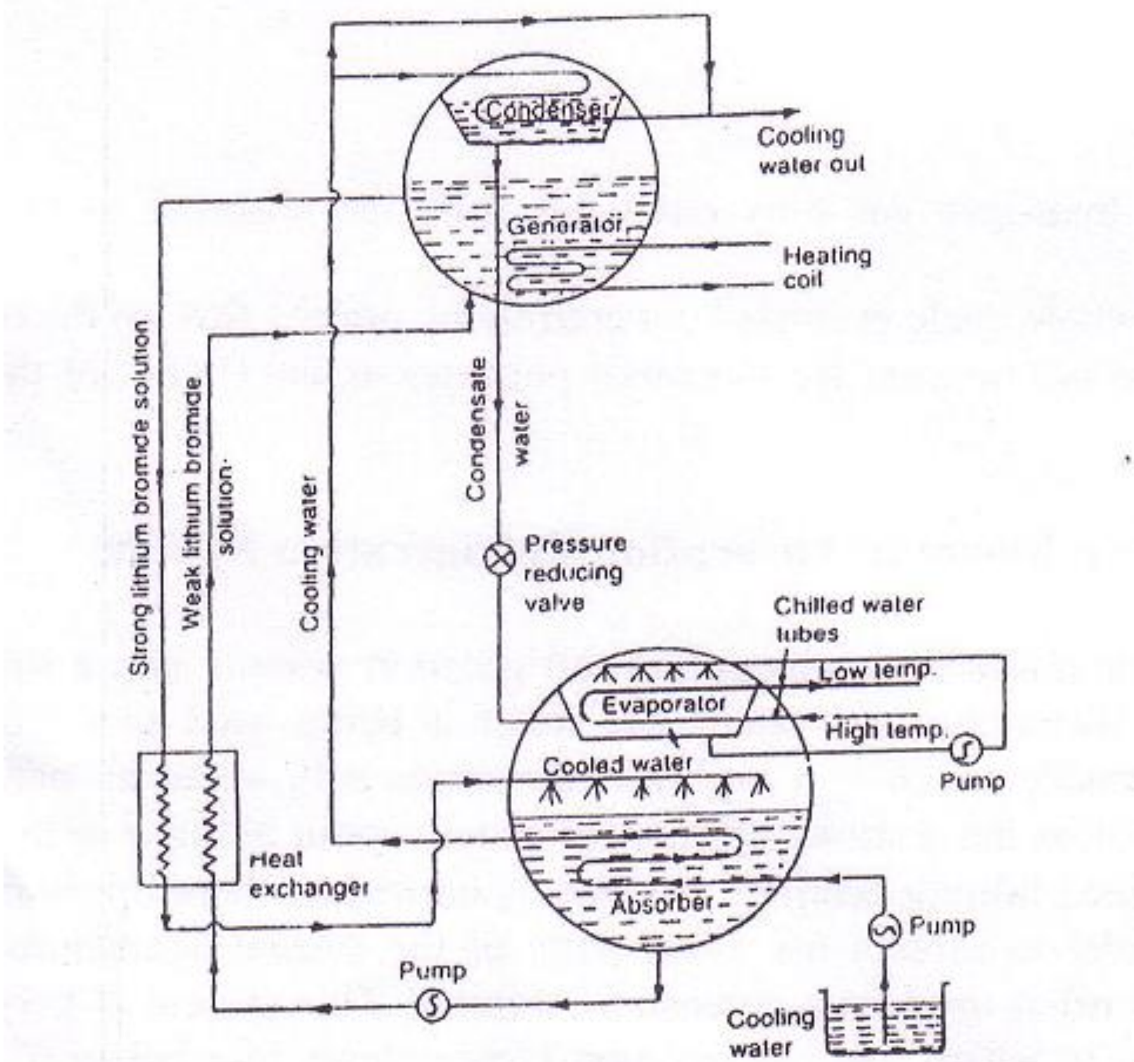
This system uses a solution of lithium bromide in water. In this system, the water is being used as a refrigerant whereas lithium bromide, which is a highly hydroscopic salt, as an absorbent. The lithium bromide solution has a strong affinity for water vapour because of its very low vapour pressure

This system is very popular for air conditioning in which low refrigeration temperature (not below 0c̊) are required.

Lithium bromide- water system designed in two forms:

The first where all the components of cycle are placed in same shell, its upper half contain the generator & condenser while its lower half contain the evaporator & absorber this type called (one shell system) as in simple absorption system.

The second form consists of two shells, the first shell (high pressure side) contain the generator & condenser, while the second shell (low pressure side) contain the evaporator & absorber. This system called (two shell system) as show in figure below.



L hium-Bromide absorption refrigeration system.

The pressure difference between the generator & the absorber & the gravity due to the height difference of the two shells is utilized to create the pressure for the spray.

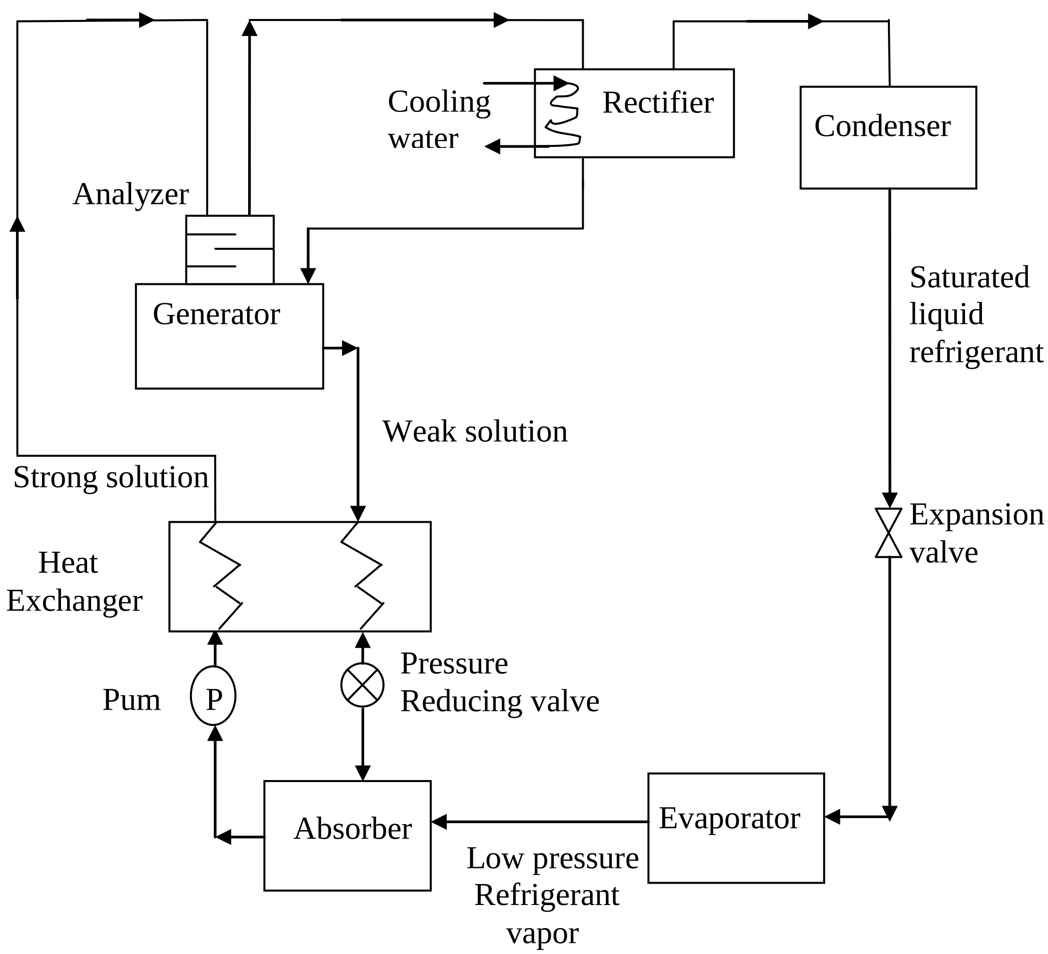
<u>Ex.1, Ex.2, Ex.3</u>

Aqua ammonia absorption system:

It is the first used absorption refrigeration system consists of the same mentioned components in the absorption system & lithium bromide- water cycle, also its principle of work is the same.

This cycle use ammonia as a refrigerant & the water as a absorbent.

Unlike lithium bromide which not evaporative there is some water evaporate with ammonia in generator. To solve this problem & to improve the performance of the cycle the following components are added:
Heat exchanger, Analyzer and Rectifier.



1-Heat exchanger:

The strong solution which is pumped from the absorber to the generator must be heated & the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger & consequently cost of heating the generator & cost of cooling the absorber are reduced.

2-Analyser:

It is consists of a series of trays mounted above the generator. Its main function is to remove partly some of the unwanted water particles associated with ammonia vapor going to condenser.

3-Rectifier:

Is a water cooled heat exchanger which condenses water vapour & some ammonia & sends back to the generator.

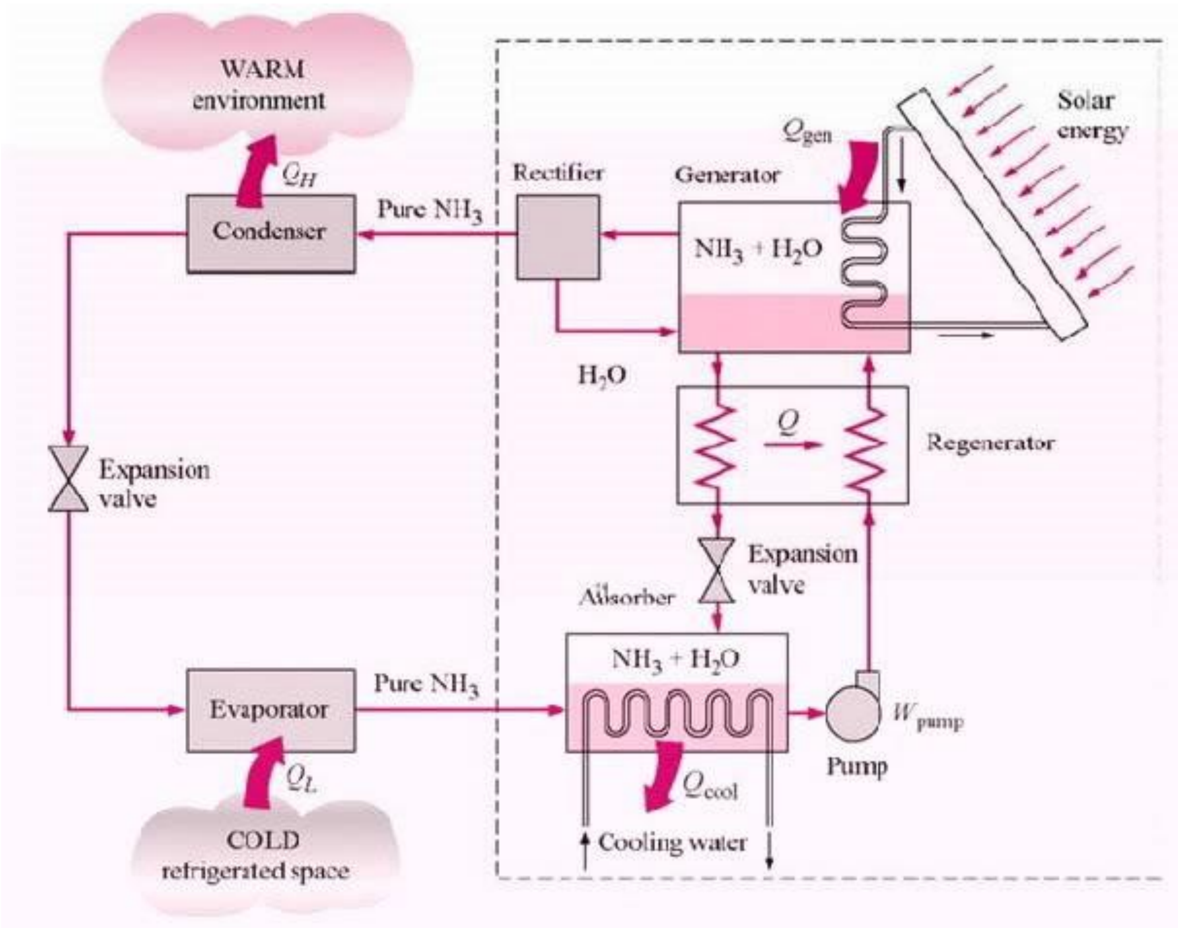
SOLAR ABSORPTION REFRIGERATION SYSTEM

The use of solar energy as an alternative source has been attracting a lot of interest in the last years due to the environmental considerations and saving in energy. So many applications have been considered among theme solar absorption refrigeration is one of the applications under review because of the advantages that present its use in sunny and warm regions, where solar power can be used as the main source for its operation. Unlike mechanical vapor compression refrigeration systems, these systems cause no ozone depletion and reduce demand on electricity supply.

Heat powered systems could be superior to electricity powered systems because of the use of inexpensive sources such as waste heat, solar, biomass or geothermal energy sources for which the supplying cost is negligible in many cases.

Despite using an economic energy sources, the system is characterized by its low COP, for that reason it is necessary to perform a study in order to find the most efficient operation range and the affecting parameters.

The solar air conditioning is highly attractive as more cooling is needed when the solar intensity is strong and higher ambient temperatures are present. There is a notion that the present cost of these equipments is still prohibitive, not only for the price of the solar cooling equipments and solar collectors themselves, but also for the cost of required backup systems.


WARM
environment
Q_H
Condenser
Pure NH_3
Rectifier
Generator
Q_{gen}
Solar
energy
$NH_3 + H_2O$
H_2O
Q
Regenerator
Expansion
valve
Expansion
valve
Absorber
$NH_3 + H_2O$
Evaporator
Pure NH_3
W_{pump}
Pump
Q_L
Q_{cool}
Cooling water
COLD
refrigerated space