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SCHOOL OF MECHANICAL ENGINEERING
DEPARTMENT OF MECHANICAL ENGINEERING

UNIT – I – Refrigeration and Air Conditioning – SME1618

I. Basics of Refrigeration

Refrigeration

Literal meaning of refrigeration is the production of cold confinement relative to its surroundings. In this, temperature of the space under consideration is maintained at a temperature lower than the surrounding atmosphere. To achieve this, the mechanical device extracts heat from the space that has to be maintained at a lower temperature and rejects it to the surrounding atmosphere that is at a relatively higher temperature. Since the volume of the space which has to be maintained at a lower temperature is always much lower than the environment, the space under consideration experiences relatively higher change in temperature than the environment where it is rejected.

The precise meaning of the refrigeration is thus the following: Refrigeration is a process of removal of heat from a space where it is unwanted and transferring the same to the surrounding environment where it makes little or no difference. To understand the above definition, let us consider two examples from the daily life.

Unit of Refrigeration and COP

The standard unit of refrigeration is ton refrigeration or simply ton denoted by TR. It is equivalent to the rate of heat transfer needed to produce 1 ton (2000 lbs) of ice at 32 ^{0}F from water at 32 ^{0}F in one day, i.e., 24 hours. The enthalpy of solidification of water from and at 32 ^{0}F in British thermal unit is 144 Btu/lb.

Refrigeration effect is an important term in refrigeration that defines the amount of cooling produced by a system. This cooling is obtained at the expense of some form of energy.

Therefore, it is customary to define a term called coefficient of performance (COP) as the ratio of the refrigeration effect to energy input.

Refrigerator

Refrigeration is the process of maintaining the temperature of a body below that of its surroundings.

The working fluid used for this purpose is called Refrigerant the equipment used is called Refrigerator.

The refrigerant is compressed to high pressure and high temperature in the compressor and it gets cooled in the condenser at constant pressure.

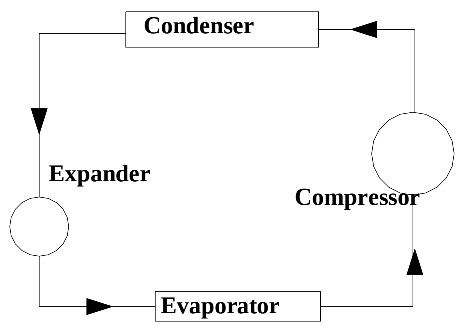


Fig 1 Basic Refrigeration Flow Diagram

The high pressure refrigerant vapour is expanded in the expander and the required quantity of refrigerant evaporates in the evaporator by absorbing heat from the space to be cooled and the cycle is repeated.

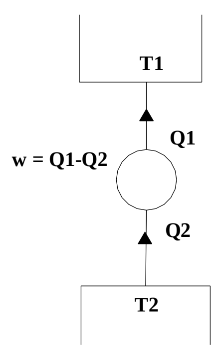


Fig 2 Refrigeration cycle

Co-efficient of Performance (C.O.P)

Co-efficient of performance

$$= \frac{\text{Refrigeration Effect}}{\textbf{Workdone}} = \frac{Q_2}{\mathbf{w}}$$

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

COP is always greater than Unity.

Heat Pump

Heat pump is a device which maintains temperature of a body greater that of its surroundings.

$$COP = \frac{Q_1}{W}$$

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

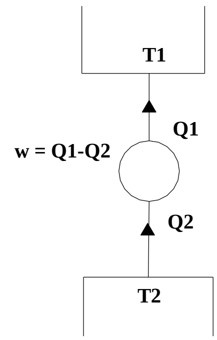


Fig 3 Heat Pump cycle

$$(COP)_{H.P} = (COP)_{ref} + 1$$

1. A refrigeration system produces 40 kg/hr of ice at 0°C from water at 25°C. Find the refrigeration effect per hour and TR. If it consumes 1 kW of energy to produce the ice, find the COP. Take latent heat of solidification of water at 0oC as 335 kJ/kg and specific heat of water 4.19 kJ/kg ^{o}C.

Solution

Heat removal rate to form 40 kg of ice at 0^{o}C from water at 25^{o}C

Qc = sensible cooling from 25^{o}C to 0^{o}C + latent heat of solidification of water

= 40 kg/hr X(25 – 0) ^{o}Cx.19 kJ/kg. ^{o}C + 40 kg/hr = 335 kJ/kg

= 4190 kJ/hr + 13400 kJ/hr

= 17590 kJ/hr

Refrigeration effect (*QC*) = 17590 kJ/hr

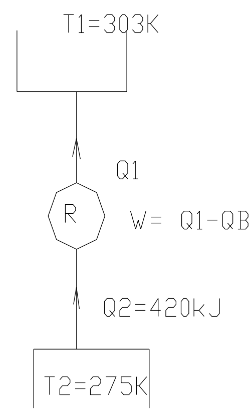
We know that 1 TR = 210 kJ/min = 12600 kJ/hr

Therefore, TR equivalent to 17590 kJ/hr = 17590kJ/hr 1.396 12600kJ/hr

Refrigeration effect 17590kJ/hr

COP = 4.886

2. A house hold refrigerator is maintained at a temperature of 2°c. Every time the door is opened, warm material is placed inside, introducing an average of 420kJ. But making only a small change in the temperature of the refrigerator. The door is opened 20 times a day, and the refrigerator operates at 15%of ideal COP. The cost of work is 32 paise per kWhr. What is the monthly bill for this refrigerator? The atmosphere is at 30°C.



Given : T_1 =303K. T_2 = 275K. Q_2 = 420kJ.

$$(COP)_{id} = \frac{T_2}{T_1 - T_2} = \frac{275}{303-275} = 9.82$$

$(COP)_{AC}$ = 0.15 x (COP)id

= 1.47.

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

$$1.47 = \frac{420}{Q_1 - Q_2} \Rightarrow Q_1 - Q_2 = 285.7kJ.$$

Cost of work is 32 paise per kWhr.

The monthly bill for the refrigerator

Air Refrigeration Cycles

Introduction

In an air refrigeration cycle, the air is used as refrigerant. In olden days, air was widely used in commercial applications because of its availability at free of cost. Since air does not change its phase i.e. remains throughout the cycle, therefore the heat carrying capacity per kg of air is very small as compared to vapour absorbing systems.

The practical unit of refrigeration is expressed in terms of ‘tone of refrigeration’ (briefly written as TR). A tone of refrigeration is defined as the amount of refrigeration effect produced by the uniform melting of one tone (1000 kg) of ice from and at 00C in 24 hours. Since the latent heat of ice is 335 kJ/kg, therefore one tone of refrigeration.

3. Air enters the compressor of air craft system at 100kpa, 277k and is compressed to 300kpa with an isentropic efficiency of 72%. After being cooled to 328k and air expands is 100kpa and an η_{Isen}=78% the load is 3 tons and find COP, power, mass flow rate.

Given data:

P_1= 100kpa, T_3=38k

T_1= 277k, P_4=100kpa

P_2= 300kpa, η_T=78%

η_c= 72%

Solution:

process 1-2 Isentropic compression

$$T2= (P2/P1)^{\gamma-1/\gamma} \times T1$$

T_2= $(300/100)^{1.4-1/1.4}$

T_2=379.14k

$\eta_c=(T_2-T_1)/(T_2'-T_1)$

$0.72= (379.14-277)/ (T_2'-277)$

T_2=418.86k

Process 3-4 isentropic compression

$T3/T4= (P3/P4)^{\gamma-1/\gamma}$

$328/T_4= (300/100)^{\gamma-1/\gamma}$

$T_4=239.64k$

$\eta_t=(T_3-T_4')/(T_3-T_4)$

$0.78= (328-T_4')/ (328-239.64)$

$T_4'= 259.08k$

$COP= (T_1-T_4')/ (T_2'-T_1)$

$COP= (277-259.08)/ (418.86-277) =0.17$

1 tonne= 3.5kw of heat

3tonne= 3×3.5=10.5kw

Energy balance.

Heat energy absorbed by I_{ce}=Heat rejected by air

$= m\times C_p\times (T_1-T_4')$

$10.5= m_a\times 1.005\times(277-259.08)$

Mass of air, $m_a= 0.583Kg/sec.$

Power, $P=m_a\times Cp_a\times (T_2'-T_1)$

$= 0.583\times 1.005\times (418.86-277)$

$= 83.12kw$

UNIT – II – Refrigeration and Air Conditioning – SME1618

II. Vapour Compression Refrigeration

SIMPLE VAPOUR COMPRESSION SYSTEM

Introduction

Out of all refrigeration systems, the vapour compression system is the most important system from the view point of commercial and domestic utility. It is the most practical form of refrigeration. In this system the working fluid is a vapour. It readily evaporates and condenses or changes alternately between the vapour and liquid phases without leaving the refrigerating plant. During evaporation, it absorbs heat from the cold body. This heat is used as its latent heat for converting it from the liquid to vapour. In condensing or cooling or liquefying, it rejects heat to external body, thus creating a cooling effect in the working fluid. This refrigeration system thus acts as a latent heat pump since it pumps its latent heat from the cold body or brine and rejects it or delivers it to the external hot body or cooling medium. The principle upon which the vapour compression system works apply to all the vapours for which tables of Thermodynamic properties are available.

Simple Vapour Compression Cycle

In a simple vapour compression system fundamental processes are completed in one cycle These are:

1. Compression
2. Condensation
3. Expansion
4. Vaporization.

The flow diagram of such a cycle is shown in Fig. 1

The vapour at low temperature and pressure (state ‗2') enters the ―compressor‖ where it is compressed isentropically and subsequently its temperature and pressure increase considerably (state ‗3'). This vapour after leaving the compressor enters the condenser where it is condensed into high pressure liquid (state ‗4') and is collected in a ―receiver tank‖. From receiver tank it passes through the ―expansion valve‖, here it is throttled down to a lower pressure and has a low temperature (state ‗1'). After finding its way through expansion ―valve‖ it finally passes on to ―evaporator‖ where it extracts heat from the surroundings or circulating fluid being refrigerated and vapourises to low pressure vapour (state ‗2').

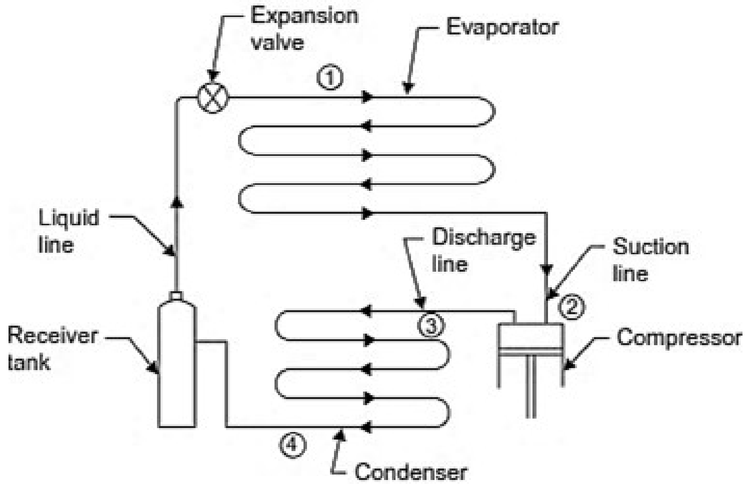


Fig. 1. Vapour compression system.

Merits and demerits of vapour compression system over Air refrigeration system:

Merits:

1. C.O.P. is quite high as the working of the cycle is very near to that of reversed Carnot cycle.
2. When used on ground level the running cost of vapour-compression refrigeration system is only 1/5th of air refrigeration system.
3. For the same refrigerating effect the size of the evaporator is smaller.
4. The required temperature of the evaporator can be achieved simply by adjusting thethrottle valve of the same unit.

Demerits:

1. Initial cost is high.
2. The major disadvantages are inflammability, leakage of vapours and toxity. These havebeen overcome to a great extent by improvement in design.

Functions of Parts of a Simple Vapour Compression System

Here follows the brief description of various parts of a simple vapour compression system shown in Fig. 1.

1. Compressor. The function of a compressor is to remove the vapour from the evaporator, and to raise its temperature and pressure to a point such that it (vapour) can be condensed with available condensing media.

2. Discharge line (or **hot gas line**). A hot gas or discharge line delivers the high-pressure, high-temperature vapour from the discharge of the compressor to the condenser.

3. Condenser. The function of a condenser is to provide a heat transfer surface through which heat passes from the hot refrigerant vapour to the condensing medium.

4. Receiver tank. A receiver tank is used to provide storage for a condensed liquid so that a constant supply of liquid is available to the evaporator as required.

5. Liquid line. A liquid line carries the liquid refrigerant from the receiver tank to the refrigerant flow control.

6. Expansion valve (refrigerant flow control). Its function is to meter the proper amount of refrigerant to the evaporator and to reduce the pressure of liquid entering the evaporator so that liquid will vapourize in the evaporator at the desired low temperature and take out sufficient amount of heat.

7. Evaporator. An evaporator provides a heat transfer surface through which heat can pass from the refrigerated space into the vapourizing refrigerant.

8. Suction line. The suction line conveys the low pressure vapour from the evaporator to the suction inlet of the compressor.

Vapour Compression Cycle on Temperature-Entropy (T-s) Diagram

We shall consider the following three cases:

1. **When the vapour is dry and saturated at the end of compression.** Fig. 2 represents the vapour compression cycle, on T-s diagram the points 1, 2, 3 and 4 correspond to the state points 1, 2, 3 and 4 in Fig. 1.

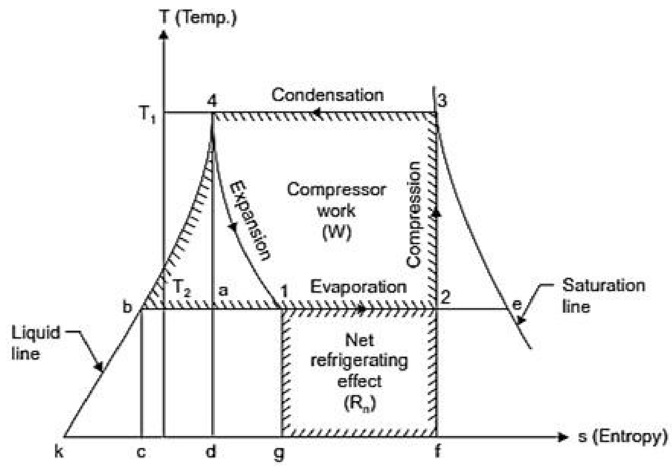


Fig. 2.T-s diagram.

At point ‗2‘ the vapour which is at low temperature (T_2) and low pressure enters the compressor‘scylinder and is compressed adiabatically to ‗3‘ when its temperature increases to thetemperature T. It is then condensed in the condenser (line 3-4) where it gives up its latent heat tothe condensing medium. It then undergoes throttling expansion while passing through the expansionvalve and it‘s again reduces to T_1, it is represented by the line 4-1. From the T-s diagram itmay be noted that due to this expansion the liquid partially evaporates, as its dryness fraction is represented by the ratio (b_1/b_2). At ‗1‘ it enters the evaporator where it is further evaporated atconstant pressure and constant temperature to the point ‗2‘ and the cycle is completed.

Work done by the compressor = W = Area ‗2-3-4-b-2

Heat absorbed = Area 2-1-g-f-2‘

$$\text{C.O.P.} = \frac{\text{Heat extracted or refrigerating effect}}{\text{Work done}} = \frac{\text{Area '2-1-}g\text{-}f\text{-2'}}{\text{Area '2-3-4-}b\text{-2'}}$$

$$\text{C.O.P} = \frac{h_2 - h_1}{h_3 - h_2}$$

$$= \frac{h_2 - h_4}{h_3 - h_2}$$

($h_1 = h_4$, since during the throttling expansion 4-1 the total heat content remains unchanged)

2. When the vapour is superheated after compression. If the compression of the vapour is continued after it has become dry, the vapour will be superheated, and its effect on T-s diagram is shown in Fig. 3. The vapour enters the compressor at condition ‗2‘ and is compressed to ‗3‘ where it is superheated to temperature T $_{sup}$ .Then it enters the condenser. Here firstly superheated vapour cools to temperature T_1(represented by line 3-3‘) and then it condenses at constant temperature along the line 3‘ -4 ; the remaining of the cycle ; however is the same as before.

Now, Work done = Area 2-3-3‘ -4-b-2

and Heat extracted/absorbed = Area 2-1-g-f-2

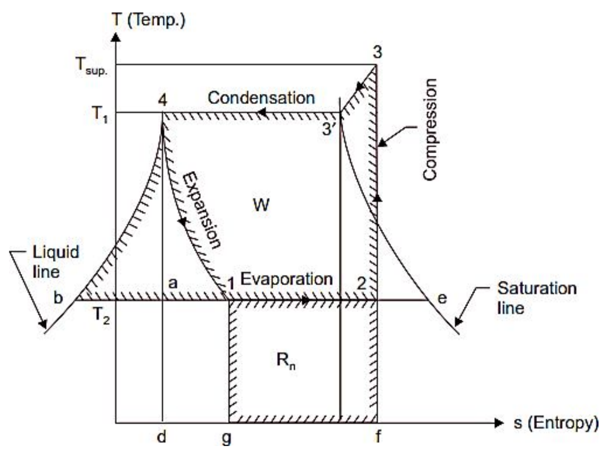


Fig. 3.T-s diagram.

$$\text{C.O.P.} = \frac{\text{Heat extracted}}{\text{Work done}} = \frac{\text{Area '2-1-}g\text{-}f\text{-2'}}{\text{Area '2-3-3'-4-}b\text{-2'}} = \frac{h_2 - h_1}{h_3 - h_2}$$

In this case h3= h3+ cp(Tsup.– Tsat.) and h3= total heat of dry and saturated vapour at thepoint ‗3 ‘.

3. When the vapour is wet after compression. Refer Fig. 4

Work done by the compressor = Area 2-3-4-b-2

Heat extracted = Area 2-1-g-f-2

$$\text{C.O.P.} = \frac{\text{Heat extracted}}{\text{Work done}} = \frac{\text{Area '2-1-}g\text{-}f\text{-2'}}{\text{Area '2-3-4-}b\text{-2'}} = \frac{h_2 - h_1}{h_3 - h_2}$$

Note. If the vapour is not superheated after compression, the operation is called ‗WET COMPRESSION' and if the vapour is superheated at the end of compression, it is known as ‗DRY COMPRESSION'. Dry compression, in actual practice is always preferred as it gives higher volumetric efficiency and mechanical efficiency and there are less chances of compressor damage.

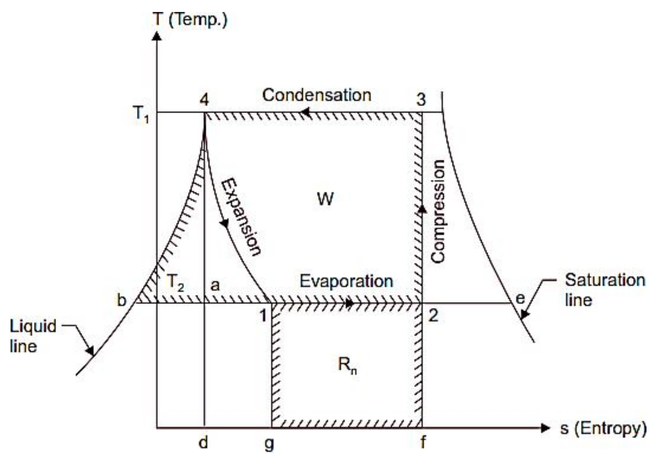


Fig. 4 T-s diagram.

Simple Vapour Compression Cycle on p-h Chart

Fig. 5 shows a simple vapour compression cycle on a p-h chart. The points 1, 2, 3 and 4 correspond to the points marked in Fig. 1

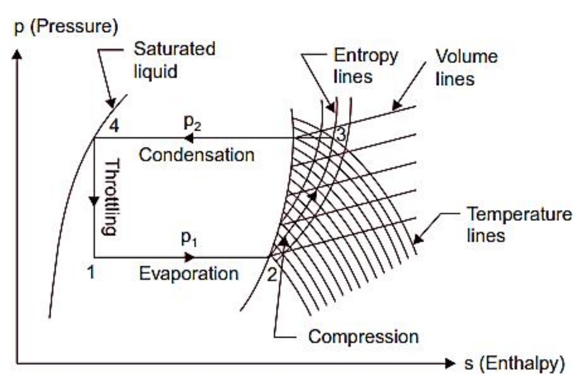


Fig. 5.Simple vapour compression cycle on p-h chart.

The dry saturated vapour (at state 2) is drawn by the compressor from evaporator at lower pressure p_1 and then it (vapour) is compressed isentropically to the upper pressure p_2. The isentropic compression is shown by the line 2-3. Since the vapour is dry and saturated at the start of compression it becomes superheated at the end of compression as given by point 3. The process of condensation which takes place at constant pressure is given by the line 3-4. The vapour now reduced to saturated liquid is throttled through the expansion valve and the process is shown by the line 4-1. At the point 1 a mixture of vapour and liquid enters the evaporator where it gets dry saturated as shown by the point 2. The cycle is thus completed. Heat extracted (or refrigerating effect produced),

$R_n = h_2 - h_1$

$W = h_3 - h_1$

$$\text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2}$$

The values of h1, h2 and h3 can be directly read from p-h chart.

Factors Affecting the Performance of a Vapour Compression System

The factors which affect the performance of a vapour compression system are given below :

1. **Effect of suction pressure.** The effect of decrease in suction pressure is shown inFig. 6. The C.O.P. of the original cycle,

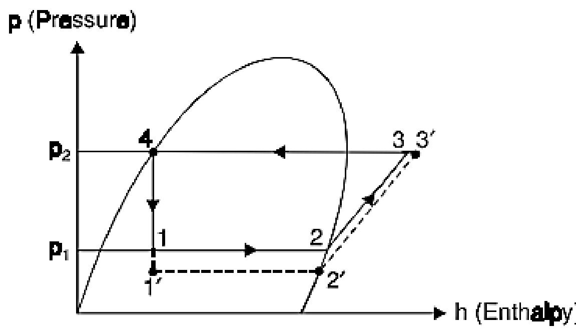


Fig. 6.Effect of decrease in suction pressure.

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

The C.O.P. of the cycle when suction pressure is decreased,

$$\text{C.O.P.} = \frac{h_2' - h_1'}{h_3' - h_2'}$$

$$= \frac{(h_2 - h_1) - (h_2 - h_2')}{(h_3 - h_2) + (h_2 - h_2') + (h_3' - h_3)} \qquad (\because \; h_1 = h_1')$$

This shows that the refrigerating effect is decreased and work required is increased. Then net effect is to reduce the refrigerating capacity of the system (with the same amount of refrigerant flow) and the C.O.P.

2. **Effect of delivery pressure.** Fig. 7 shows the effect of increase in delivery pressure.

C.O.P. of the original cycle,

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

C.O.P. of the cycle when delivery pressure is increased,

$$\text{C.O.P.} = \frac{h_2 - h_1'}{h_3' - h_2} = \frac{(h_2 - h_1) - (h_1' - h_1)}{(h_3 - h_2) + (h_3' - h_3)}$$

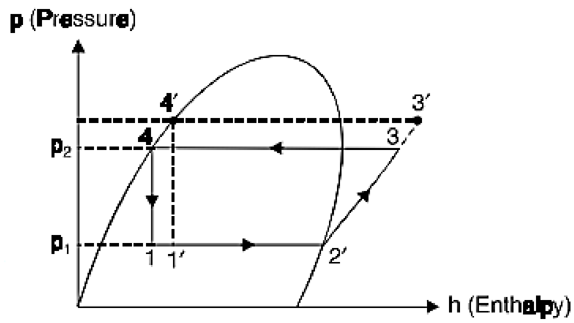


Fig. 7 Effect of increase in delivery pressure

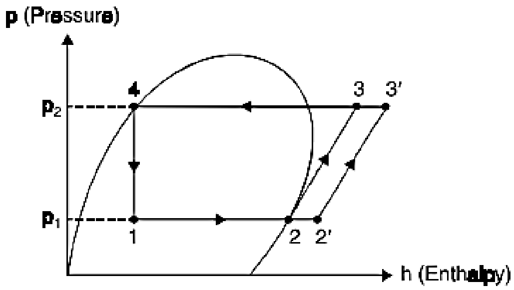
The effect of increasing the delivery/discharge pressure is just similar to the effect of decreasing the suction pressure. The only difference is that the effect of decreasing the suction pressure is more predominant than the effect of increasing the discharge pressure.

The following points may be noted:

(i) As the discharge temperature required in the summer is more as compared with winter, the same machine will give less refrigerating effect (load capacity decreased) at a higher cost.

(ii) The increase in discharge pressure is necessary for high condensing temperatures and decrease in suction pressure is necessary to maintain low temperature in the evaporator.

Effect of superheating. As may be seen from the Fig. 8 the effect of superheating is to increase the refrigerating effect but this increase in refrigerating effect is at the cost of increase in amount of work spent to attain the upper pressure limit. Since the increase in work is more as compared to increase in refrigerating effect, therefore overall effect of superheating is to give a low value of C.O.P.



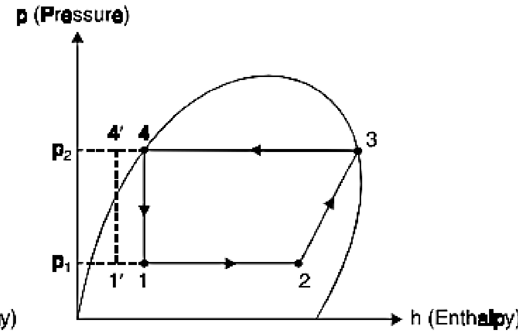


Fig. 8.Effect of superheating.Fig. 9.Effect of sub-cooling of liquid.

4. **Effect of sub-cooling of liquid.** ‗Sub-cooling' is the process of cooling the liquid refrigerant below the condensing temperature for a given pressure . In Fig. 14.18 the process of subcooling is shown by 4-4 . As is evident from the figure the effect of subcooling is to increase the refrigerating effect. Thus sub-cooling results in increase of C.O.P. provided that no further energy has to be spent to obtain the extra cold coolant required.

The sub-cooling or under cooling may be done by any of the following methods: (I) Inserting a special coil between the condenser and the expansion valve. (ii) Circulating greater quantity of cooling water through the condenser.
(iii) Using water cooler than main circulating water.

5. **Effect of suction temperature and condenser temperature.** The performance of the vapour compression refrigerating cycle varies considerably with both vapourising and condensing temperatures. Of the two, the vapourising temperature has far the greater effect. It is seen that the capacity and performance of the refrigerating system improve as the vapourising temperature increases and the condensing temperature decreases. Thus refrigerating system should always be designed to operate at the highest possible vapourising temperature and lowest possible condensing temperature, of course, keeping in view the requirements of the application.

1. An ammonia refrigerator process 20tons of ice per day from and at 0°C.The condensation and evaporation takes at 20°C and -20°C respectively the temperature of the vapour at the end of Isentropic compression is 50°C and there is no under cooling of the liquid.COP=70% of theoretical COP. Determine (i) Rate of NH_3 circulation (ii) size of compressor, N=240rpm,L=D, η_{vol}=80%. Take Laten heat of I_{cc}=335kJ/kg, C_p= 2.8 kJ/kg,

   Vs_1=0.624m^3kg. Use the following properties of ammonia.

Sat.Temp (°C)	Enthalpy(kJ/kg)		Entropy(kJ/kgk)	
	h_f	h_g	S_f	S_g
20	274.98	1461.58	1.0434	5.0919
-20	89.72	1419.05	0.3682	5.6204

Given data: 20 tons of Icc per day at °C

T_3= 20°C

T_1=-20°C

T3=50°C

COP =70% of theoretical cop

N=240rpm

L=D

h_v=80%

Latent heat of Ice=335KJ/kg

C_p=2.8KJ/kgk

V_{s1}=0.624m^3/kg

Solution:-

The refrigeration effect=20×3.5=77.55kw

h_1=1419.05KJ/kg h_{g2}=1461.58KJ/kg

at 20°C h_{f3}=274.98 KJ/kg

$h_2=h_{g2}+C_p\ (T_2-20)$

h_2=1461.58+2.8(50-20)

h_2=1545.58 KJ/kg

COP= $(h_1-h_{f3})/(h_1-h_2)$

= (1419.05-274.98)/(1545.58-1419.05)

=9.04.

2. A 5 tonne refrigerator plant uses RR as refrigerant. It enters the compressor at -5°C as saturated vapour. Condensation takes place at 32°C and there is no under cooling of refrigerant liquid. Assuming isentropic compression, determine COP of the plant, mass flow of refrigerant, power required to run the compressor in kw. The properties of R-12 are given table.

T(°C)	P(bar)	Enthalpy(kw/kg)		Entropy(KJ/kgk)
32	7.85	130.5	264.5	1.542
-5	2.61	-	249.3	1.557

Solution:

Beginning of compression in dry and end of compression is superheated. So the P-h and T-S diagrams are

From table, at point 1

T_1=-5°C=268K

hg_1=2493kJ/kg , Sg_1=1.557KJ/kgk

At point 2

T_2=32°C=305k, hf_2=130.5KJ/kg, hg_2=264.5, Sg_2=1.542KJ/kgK

From ph diagram, At point eqn(1) (dry).

At -5°C, i.e at 268k

$hg_{1=}$249.3KJ/kg=hg

h_1=249.3KJ/kg

At 32°C, i.e at 305K

$hg_{2=}$264.5KJ/kg=h_2'

h_2'=264.5KJ/kg

Entropy is constant during the compression process so,

$S_1=S_2$

From T- S diagram

At point (1) dry,

$S_1=Sg$ at -5°C

$S_1=Sg_1=1.557$KJ/kgk

$S_1=S_2=1.557$KJ/kgk

At point (2) (super heated)

$S_1=S_2' + Cp\ln (T_2/T_2')$

$1.557=S_2'+1.884 \ln(T_2/305)$-------- (1)

$S_2'=Sg$ at 32°C.

$Sg_2=1.542=S_2$

$S_2'=1.542$KJ/kg k

$1.884 \ln (T_2/305)=0.015$

$T_2=307.44$k

For super heated vapour the enthalpy is

$h_2=h_2'+C_p(T_2-T_1')$

$h_2= 264.5 + 1.884 (307.44-305)$

$h_2= 269.1$ KJ/kg

From P-h diagram, we know that,

$h_3= h_4$

$h_3=h_f$ at 32°C

$h_{f\ 2}=130.5= h_f$

We Know that,

COP=Refrigeration effect / Work done= $(h_1-h_4)/ (h_2-h_1)$

= (2493-130.5)/(269.1-249.3)=6

Refrigeration effect = m× (h1- h_4)

$m= (2×210)/ (249.3-130.5)$

$m = 8.84Kg/min$

Work done = Refrigeration effect/ cop

$= (2×210)/6 = 175$ KJ/min

Power = 2.92kw.

3. A refrigerator works between -7°C and 27°C the vapour is dry at the end of adiabatic compression. Assuming there is no under cooling determine (i) cop (ii) power of the compressor to remove a heat load of 12140KJ/hr.The properties of refrigerant are given

T(°C)	sensible Heat (h_f)	Latent heat(h_{fg}) KJ/kgk)	Entropy of liquid (KJ/kgk)	Entropy of vapourSg (KJ/kgk)
-7	-29.3	1297.9	-0.109	4.748
27	1117.23	1172.3	0.427	4.333

in table.

Solution:

The vapour is dry at end of compression i.e, beginning of compression is wet and of compression is dry saturated.

At point (1)

T_1=-7°C=266k, h_{fg1}=1297.9 KJ/kg, S_{f1}=-0.109 KJ/kgK

h_{f1}=-29.3 KJ/kg, S_{fg1}=4.478 KJ/kgK

At point (2)

T_2=27°C=300k, h_{fg2}=1172.3 KJ/kg, S_{f2}= 0.427KJ/kgK

h_{f2}=117.23 KJ/kg, S_{fg2}=4.333KJ/kgK

We point $S_1=S_2$

At point (1) (wet)

$S1=S_{wet}=S_{f1}+x_1+S_{fg}$

$1S_1= -0.109+x_1(S_{fg1}-$

$S_{f1})$ $(S_{fg}=S_g-S_f)$

S_1= -0.109+x_1(4.857)

At point (2) (dry)

S_2=S_{g2}=4.33KJ/kgK

S_2=4.33KJ/kgK

S_1=S_2 So,4.33= -0.109+x_1(4.857)

Dryness fraction

x_1=0.913

At point (1) (wet)

h1= hf1+ x1×

hfg1

h_1=-29.3+0.913×1297.3

h_1=1156.3 KJ/kg

At point (2) (dry)

h2= hf2+hg2

h_2= 117.23+1172.3

h_2= 1289.53 KJ/kg

From P-h diagram

h_3=h_4

h_3=h_{f2}

h_3=1172.3 KJ/kg

h_4=117.23

KJ/kgCOP= (h_{1-}

h_4)/ (h_2-h_1) =

(1156.3-117.23) /

(1289.53-1156.3)

= 7.7**Work done**

= Heat removed/

COP

= 12140/7.7

Power = 0.43 KJ/hr

UNIT – III – Refrigeration and Air Conditioning – SME1618

III. Vapour Absorption Refrigeration

Introduction

In vapour absorption system the refrigerant is absorbed on leaving the evaporator, the absorbing medium being a solid or liquid. In order that the sequence of events should be continuous it is necessary for the refrigerant to be separated from the absorbent and subsequently condensed before being returned to the evaporator. The separation is accomplished by the application of direct heat in a generator‘. The solubility of the refrigerant and absorbent must be suitable and the plant which uses ammonia as the refrigerant and water as absorbent will be described.

Simple Vapour Absorption System

Refer Fig. 1 for a simple absorption system. The solubility of ammonia in water at low temperatures and pressures is higher than it is at higher temperatures and pressures. The Ammonia vapour leaving the evaporator at point 2 is readily absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The ammonia in water solution is pumped to the higher pressure and is heated in the generator. Due to reduced solubility of ammonia in water at the higher pressure and temperature, the vapour is removed from the solution. The vapour then passes to the condenser and the weakened ammonia in water solution is returned to the absorber.

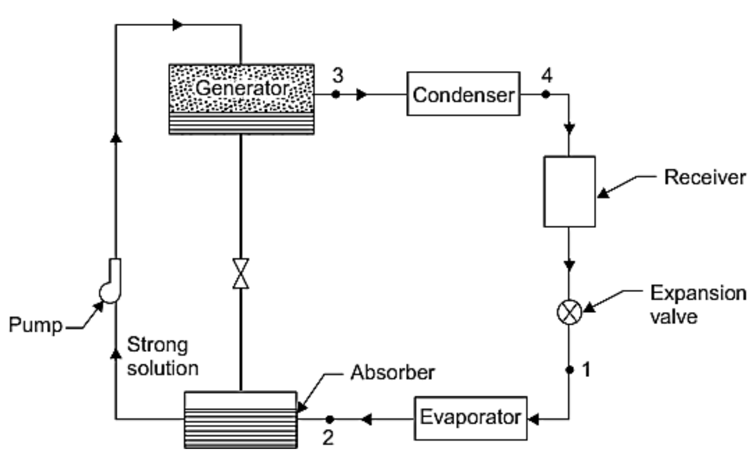


Fig. 1 Simple vapour absorption system

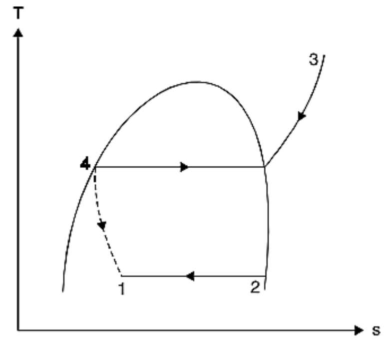


Fig. 1. (b) Simple vapour absorption system—T-s diagram

In this system the work done on compression is less than in vapour compression cycle (since pumping a liquid requires much less work than compressing a vapour between the same pressures) but a heat input to the generator is required. The heat may be supplied by any convenient form e.g. steam or gas heating.

Practical Vapour Absorption System

Refer Fig. 2. Although a simple vapour absorption system can provide refrigeration yet its operating efficiency is low. The following accessories are fitted to make the system more practical and improve the performance and working of the plant.

1. Heat exchanger. 2. Analyzer. 3. Rectifier.

1. **Heat exchanger.** A heat exchanger is located between the generator and the absorber.
The strong solution which is pumped from the absorber to the generator must be heated; and the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger and consequently cost of heating the generator and cost of cooling the absorber are reduced.

2. **Analyzer.** An analyzer 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 vapour going to condenser. If these water vapours are permitted to enter condenser they may enter the expansion valve and freeze; as a result the pipe line may get choked.

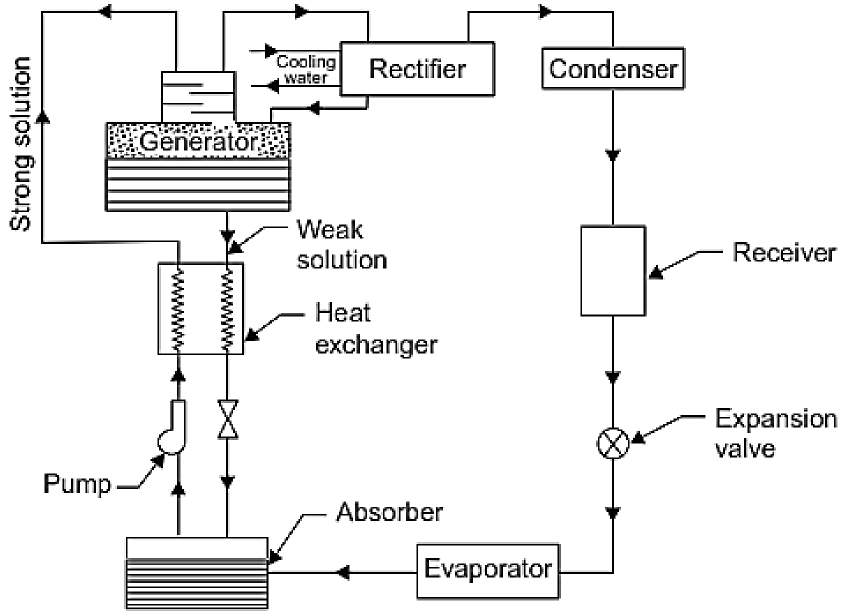


Fig. 2 Practical Vapour Absorption System

3. **Rectifier.** A rectifier is a water-cooled heat exchanger which condenses water vapour and some ammonia and sends back to the generator. Thus final reduction or elimination of the percentage of water vapour takes place in a rectifier.

The co-efficient of performance (C.O.P.) of this system is given by :

$$\text{C.O.P.} = \frac{\text{Heat extracted from the evaporator}}{\text{Heat supplied in the generator + Work done by the liquid pump}}$$

COMPARISON BETWEEN VAPOUR COMPRESSION AND VAPOUR ABSORPTION SYSTEMS

S. No.	**Particulars**	**Vapour compression System**	**Vapour absorption system**
1	Type of energy supplied	Mechanical-a high grade Energy	Mainly heat-a low grade energy
2	Energy supply	Low	High
3	Wear and tear	More	Less
4	Performance at Part loads.	Poor	System not affected by variationsloads

5	Suitability	Used where high grade mechanical energy is available	Can also be used at remote places as it can work even with a simple kerosene lamp (of course in small capacities)
6	Charging of refrigerant	Simple	Difficult
7	Leakage of refrigerant	More chances	No chance as there is no compressor or any reciprocating component to cause leakage.
8	Damage	Liquid traces in suction line may damage the compressor	Liquid traces of refrigerant present in piping at the exit ofevaporator constitute no danger.

UNIT – IV – Refrigeration and Air Conditioning – SME1618

IV. Psychrometry and Air Conditioning

Concept of Psychrometry and Psychrometrics

Air comprises of fixed gases principally, nitrogen and oxygen with an admixture of water vapour in varying amounts. In atmospheric air water is always present and its relative weight averages less than 1% of the weight of atmospheric air in temperate climates and less than 3% by weight under the most extreme natural climatic conditions, it is nevertheless one of most important factors in human comfort and has significant effects on many materials. Its effect on human activities is in fact altogether disproportionate to its relative weights. The art of measuring the moisture content of air is termed Psychrometry. The science which investigates the thermal properties of moist air, considers the measurement and control of the moisture content of air, and studies the effect of atmospheric moisture on material and human comfort may properly be termed **–psychrometrics'**'.

DEFINITIONS

Some of the more important definitions are given below:

1. **Dry air.** The international joint committee on Psychrometric Data has adopted the following exact composition of air expressed in mole fractions (Volumetric) Oxygen 0.2095, Nitrogen0.7809, Argon 0.0093, Carbon dioxide 0.0003. Traces of rare gases are neglected. Molecular weight of air for all air conditioning calculations will be taken as 28.97. Hence the gas constant,

$$R_{air} = \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K}$$

Dry air is never found in practice. Air always contains some moisture. Hence the common designation –air‖ usually means moist air. The term ‗dry air' is used to indicate the water free contents of air having any degree of moisture.

2. **Saturated air.** Moist air is said to be saturated when its condition is such that it can co-exist in natural equilibrium with an associated condensed moisture phase presenting a flat surface to it. For a given temperature, a given quantity of air can be saturated with a fixed quantity of moisture. At higher temperatures, it requires a larger quantity of moisture to saturate it. At saturation, vapour pressure of moisture in air corresponds to the saturation

pressure given in steam tables corresponding to the given temperature of air.

3. **Dry-bulb temperature (DBT).** It is the temperature of air as registered by an ordinary thermometer (t_{db}).

4. **Wet-bulb temperature (WBT).** It is the temperature registered by a thermometer when the bulb is covered by a wetted wick and is exposed to a current of rapidly moving air (t_{wb}).

5. **Adiabatic saturation temperature.** It is the temperature at which the water or ice can saturate air by evaporating adiabatically into it. It is numerically equivalent to the measured wet bulb temperature (as corrected, if necessary for radiation and conduction) ($t_{db}- t_{wb}$).

6. **Wet bulb depression.** It is the difference between dry-bulb and wet bulb temperatures.

7. **Dew point temperature (DPT).** It is the temperature to which air must be cooled a tconstant pressure in order to cause condensation of any of its water vapour. It is equal to steam table saturation temperature corresponding to the actual partial pressure of water vapour in the air (t_{dp}).

8. **Dew point depression.** It is the difference between the dry bulb and dew point temperatures ($t_{db} – t_{dp}$).

9. **Specific humidity (Humidity ratio).** It is the ratio of the mass of water vapour per unit mass of dry air in the mixture of vapour and air, it is generally expressed as grams of water per kg of dry air. For a given barometric pressure it is a function of dew point temperature alone.

10. **Relative humidity(RH), (□).** It is the ratio of the partial pressure of water vapour in the mixture to the saturated partial pressure at the dry bulb temperature, expressed as percentage.

11. **Sensible heat.** It is the heat that changes the temperature of a substance when added to or abstracted from it.

12. **Latent heat.** It is the heat that does not affect the temperature but changes the state of substance when added to or abstracted from it.

13. **Enthalpy.** It is the combination energy which represents the sum of internal and flow energy in a steady flow process. It is determined from an arbitrary datum point for the air mixture and is expressed as kJ per kg of dry air (h).

Note. When air is saturated DBT, WBT, DPT are equal.

Psychrometric Relations

Pressure

Dalton's law of partial pressure is employed to determine the pressure of a mixture of gases. This law states that the total pressure of a mixture of gases is equal to the sum of partial pressures which the component gases would exert if each existed alone in the mixture volume at the mixture temperature. Precise measurements made during the last few years indicate that this law as well as Boyle's and Charle's laws are only approximately correct. Modern tables of atmospheric air properties are based on the correct versions. For calculating partial pressure of water vapour in the air many equations have been proposed, probably Dr. Carrier's equation is most widely used.

$$p_v = (p_{vs})_{wb} - \frac{[pt - (p_{vs})_{wb}](t_{db} - t_{wb})}{1527.4 - 1.3\, t_{wb}}$$

where

p_v = Partial pressure of water vapour,

p_{vs}= Partial pressure of water vapour when air is fully saturated,

p_t = Total pressure of moist air,

t_{db}= Dry bulb temperature (ºC), and

t_{wb}= Wet bulb temperature (ºC).

Specific humidity W:

$$\text{Specific humidity} = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}}$$

$$W = \frac{m_v}{m_a}$$

Also,

$$m_a = \frac{p_a V}{R_a T}$$

$$m_v = \frac{p_v \times V}{R_v \times T}$$

Where,

p_a= Partial pressure of dry air,

p_v= Partial pressure of water vapour, V=

Volume of mixture,

$$W = \frac{p_v \times V}{R_v \times T} \times \frac{R_a\,T}{p_a\,V} = \frac{R_a}{R_v} \times \frac{p_v}{p_a}$$

$$R_a = \frac{R_0}{M_a} \qquad \left(= \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K in SI units} \right)$$

$$R_v = \frac{R_0}{M_v} \qquad \left(= \frac{8.3143}{18} = 0.462 \text{ kJ/kg K in SI units} \right)$$

Where

R_0 = Universal gas constant,

M_a= Molecular weight of air, and

M_v = Molecular weight of water vapour.

$$W = \frac{0.287}{0.462} \cdot \frac{p_v}{p_a} = 0.622 \frac{p_v}{p_t - p_v}$$

$$W = 0.622 \frac{p_v}{p_t - p_v}$$

The masses of air and water vapour in terms of specific volumes are given by expression as

$$m_a = \frac{V}{v_a} \quad \text{and} \quad m_v = \frac{V}{v_v}$$

Where

v_a= Specific volume of dry air, and

v_v= Specific volume of water vapour.

$$W = \frac{v_a}{v_v}$$

Degree of saturation (μ):

$$\text{Degree of saturation} = \frac{\text{Mass of water vapour associated with unit mass of dry air}}{\text{Mass of water vapour associated with saturated unit mass of dry saturated air}}$$

$$\mu = \frac{W}{W_s}$$

W_s = Specific humidity of air when air is fully saturated

$$\mu = \frac{0.622\left(\dfrac{p_v}{p_t - p_v}\right)}{0.622\left(\dfrac{p_{vs}}{p_t - p_{vs}}\right)} = \frac{p_v(p_t - p_{vs})}{p_{vs}(p_t - p_v)}$$

$$= \frac{p_v}{p_s}\left[\frac{\left(1 - \dfrac{p_{vs}}{p_t}\right)}{\left(1 - \dfrac{p_v}{p_t}\right)}\right]$$

Where

p_{vs} = Partial pressure of water vapour when air is fully saturated (p_{vs}can be calculatedfrom steam tables corresponding to the dry bulb temperature of the air).

Relative humidity (RH) :

$$\text{Relative humidity, } \phi = \frac{\text{Mass of water vapour in a given volume}}{\text{Mass of water vapour in the same volume if saturated at the same temp.}}$$

$$= \frac{m}{m_{vs}} = \frac{\dfrac{p_v T}{R_v T}}{\dfrac{p_{vs} T}{R_v T}} = \frac{p_v}{p_{vs}}$$

$$\phi = \frac{p_a W}{0.622} \times \frac{1}{p_{vs}} = 1.6\ W\ \frac{p_a}{p_{vs}}$$

Note 1. Relative humidity as compared to specific humidity plays a vital role in comfort air-conditioning and industrial air-conditioning. Relative humidity signifies the absorption capacity of air. If initial relative humidity of air is less it will absorb more moisture.

.

2. W, μ and ϕ cannot be conveniently measured as they require measurement of p_v and p_{vs}. The value of p_v can be obtained from the measurement of the wet bulb temperature and the value of p_{vs} can be calculated from steam tables corresponding to given air temperature.

Enthalpy of moist air

It is the sum of enthalpy of dry air and enthalpy of water vapour associated with dry air. It is expressed in kJ/kg of dry air.

$$h = h_{air} + W \,.\, h_{vapour}$$
$$= c_p t_{db} + W \,.\, h_{vapour}$$

where h = Enthalpy of mixture/kg of dry air,

h_{air} = Enthalpy of 1 kg of dry air,

h_{vapour} = Enthalpy of 1 kg of vapour obtained from steam tables,

W = Specific humidity in kg/kg of dry air, and

c_p = Specific heat of dry air normally assumed as 1.005 kJ/kg K.

Also $$h_{vapour} = h_g + c_{ps}\,(t_{db} - t_{dp})$$

where h_g = Enthalpy of saturated steam at dew point temperature,

and c_{ps} = 1.88 kJ/kg K.

$$\therefore \quad h = c_p t_{db} + W[h_g + c_{ps}(t_{db} - t_{dp})]$$
$$= (c_p + c_{ps}\,W)\,t_{db} + W(h_g - c_{ps} t_{dp})$$
$$= c_{pm} t_{db} + W(h_g - c_{ps} t_{dp})$$

Where $C_{pm} = (C_p + C_{ps}W)$ is the specific heat of humid air or humid specific heat.

The value of C_{pm} is taken as 1.021 kJ/kg dry air per K. It is the heat capacity of (1 + *W*) kg of moisture per kg of dry air.

$H_{vapour} = h_g$ at dry bulb temperature. So,

$$h = c_p t_{db} + W\, h_g.$$

However, a better approximation is given by the following relationship:

$$h_{\text{vapour}} = 2500 + 1.88 t_{db} \text{ kJ/kg of water vapour}$$

Where t_{db} is dry bulb temperature in ºC, and the datum state is liquid water at 0ºC.

$$h = 1.005\ t_{db} + W(2500 + 1.88\ t_{db}) \text{ kJ/kg dry air.}$$

PSYCHROMETRIC CHARTS

The psychrometric charts are prepared to represent graphically all the necessary moist air properties used for air conditioning calculations. The values are based on actual measurements verified for thermodynamic consistency.

For psychrometric charts the most convenient co-ordinates are dry bulb temperature of air vapour mixture as the abcissa and moisture content (kg/kg of dry air) or water vapour pressure as the ordinate. Depending upon whether the humidity contents are abcissa or ordinate with temperature co-ordinate, the charts are generally classified as Mollier chart and Carrier chart. Carrier chart having t_{db} as the abcissa and W as the ordinate finds a wide application. The chart is constructed as under:

1. The dry bulb temperature (ºC) of unit mass of dry air for different humidity contents or humidity ratios are indicated by vertical lines drawn parallel to the ordinate.

2. The mass of water vapour in kg (or grams) per kg of dry air is drawn parallel to the abcissa for different values of dry bulb temperature. It is the major vertical scale of the chart.

3. Pressure of water vapour in mm of mercury is shown in the scale at left and is the absolute pressure of steam.

4. Dew point temperatures are temperatures corresponding to the boiling points of water at low pressures of water vapour and are shown in the scale on the upper curved line. The dew points for

different low pressures are read on diagonal co-ordinates.

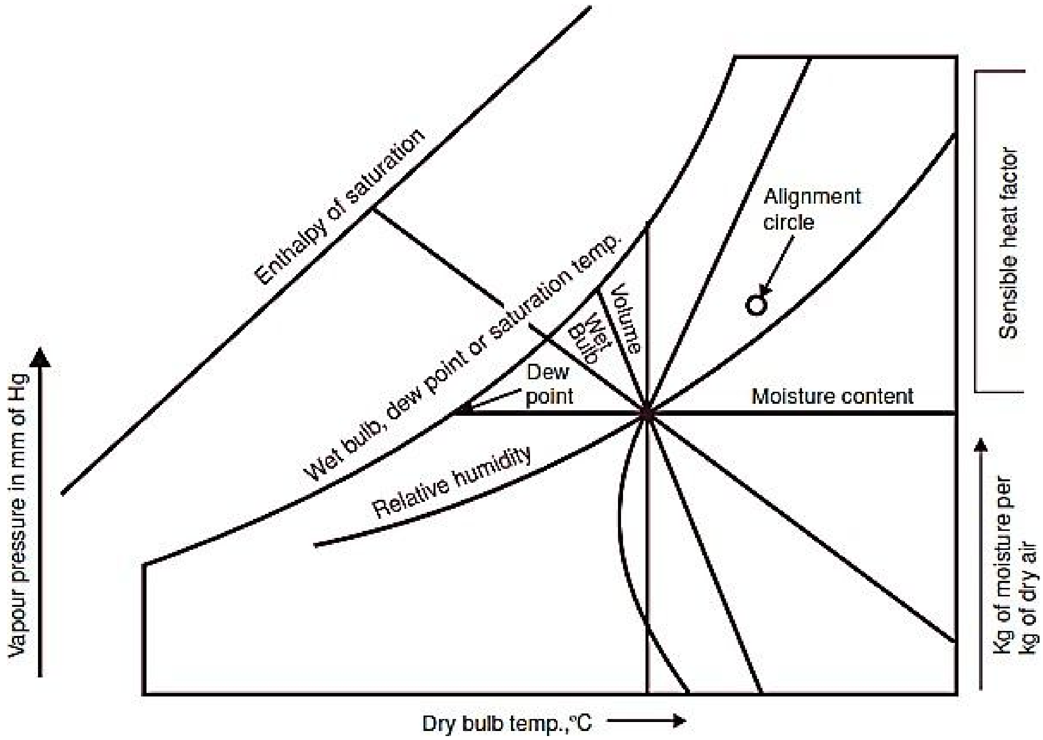


Fig.a. Skeleton psychrometric chart.

5. Constant relative humidity lines in per cent are indicated by marking off vertical distances between the saturation line or the upper curved line and the base of the chart. The relative humidity curve depicts quantity (kg) of moisture actually present in the air as a percentage of the total amount possible at various dry bulb temperatures and masses of vapour.

6. Enthalpy or total heat at saturation temperature in kJ/kg of dry air is shown by a diagonal system of co-ordinates. The scale on the diagonal line is separate from the body of the chart and is indicated above the saturation line.

7. Wet bulb temperatures are shown on the diagonal co-ordinates coinciding with heat coordinates. The scale of wet bulb temperatures is shown on the saturation curve. The diagonals run downwards to the right at an angle of 30° to the horizontal.

8. The volume of air vapour mixture per kg of dry air (specific volume) is also indicated bya set of

diagonal co-ordinates but at an angle of 60º with the horizontal.

The other properties of air vapour mixtures can be determined by using formulae (already discussed).In relation to the psychrometric chart, these terms can quickly indicate many things about the condition of air, for example:

1. If dry bulb and wet bulb temperatures are known, the relative humidity can be read from the chart.
2. If the dry bulb and relative humidity are known, the wet bulb temperature can be deter- mined.
3. If wet bulb temperature and relative humidity are known, the dry bulb temperature can be found.
4. If wet bulb and dry bulb temperatures are known, the dew point can be found.
5. If wet bulb and relative humidity are known, dew point can be read from the chart.
6. If dry-bulb and relative humidity are known, dew point can be found.
7. The quantity (kg) of moisture in air can be determined from any of the following combinations:

(i) Dry bulb temperature and relative humidity;

(ii) Dry bulb temperature and dew point;

(iii) Wet bulb temperature and relative humidity;

(iv) Wet bulb temperature and dew point temperature;

(v) Dry bulb temperature and wet bulb temperature; and

(vi) Dew point temperature alone.

Figs. a an b show the skeleton psychrometric chart and lines on carrier chart respectively.

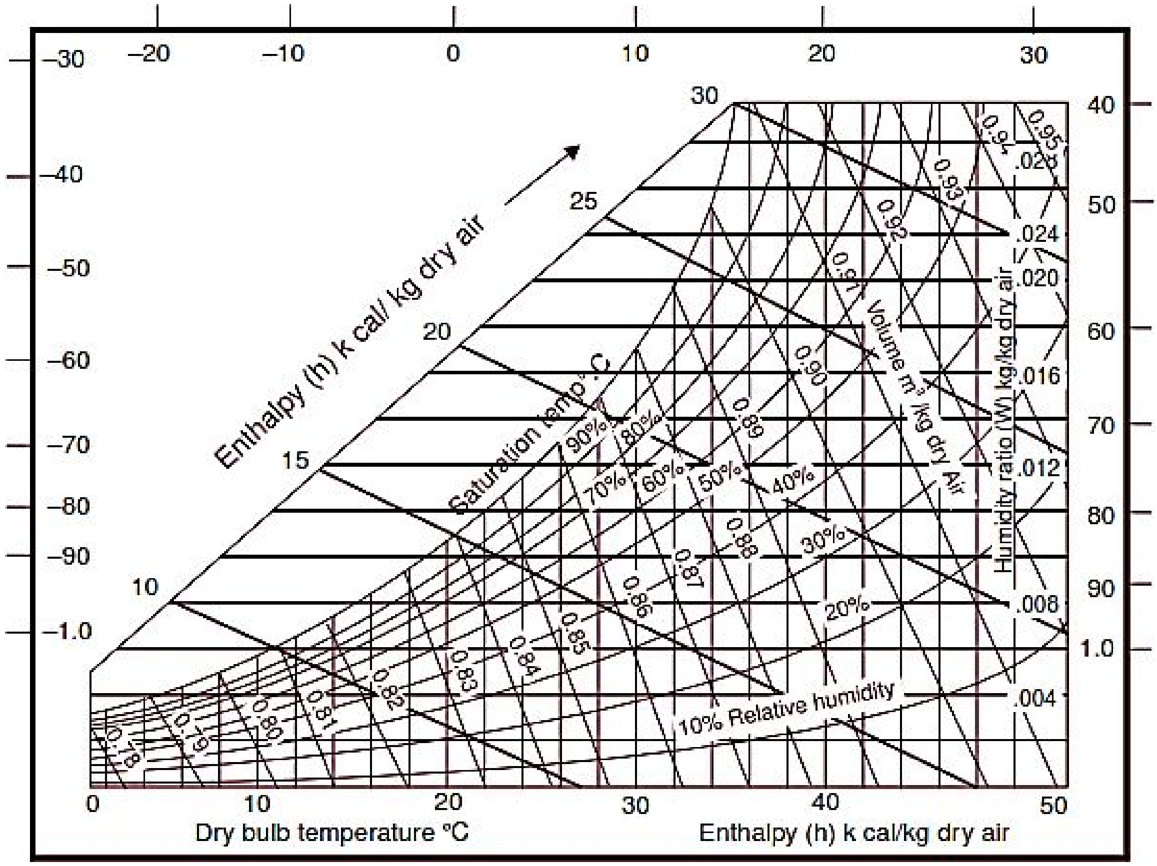


Fig b..Carrier chart.

PSYCHROMETRIC PROCESSES

In order to condition air to the conditions of human comfort or of the optimum control of an industrial process required, certain processes are to be carried out on the outside air available. The processes affecting the psychrometric properties of air are called **psychrometric processes.** These processes involve mixing of air streams, heating, cooling, humidifying, dehumidifying, adiabatic saturation and mostly the combinations of these. The important psychrometric processes are enumerated and explained in the following text:

4. Mixing of air streams
5. Sensible heating
6. Sensible cooling

7. Cooling and dehumidification
8. Cooling and humidification
9. Heating and dehumidification
10. Heating and humidification.

Mixing of Air Streams

Refer Figs. C and D. Mixing of several air streams is the process which is very frequentlyused in air conditioning. This mixing normally takes place without the addition or rejection of either heat or moisture, i.e., adiabatically and at constant total moisture content. Thus we canwrite the following equations :

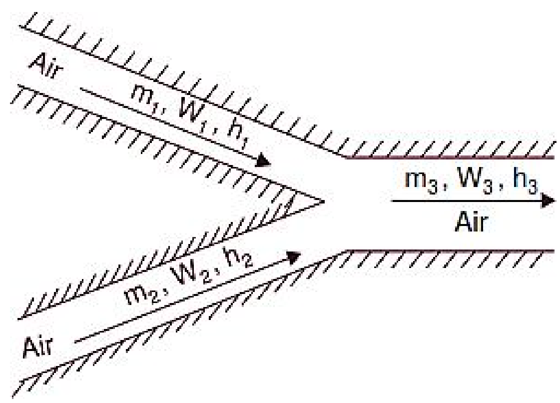


Fig. C. Mixing of air streams.

$$m_1 + m_2 = m_3$$
$$m_1W_1 + m_2W_2 = m_3W_3$$
$$m_1h_1 + m_2h_2 = m_3h_3$$

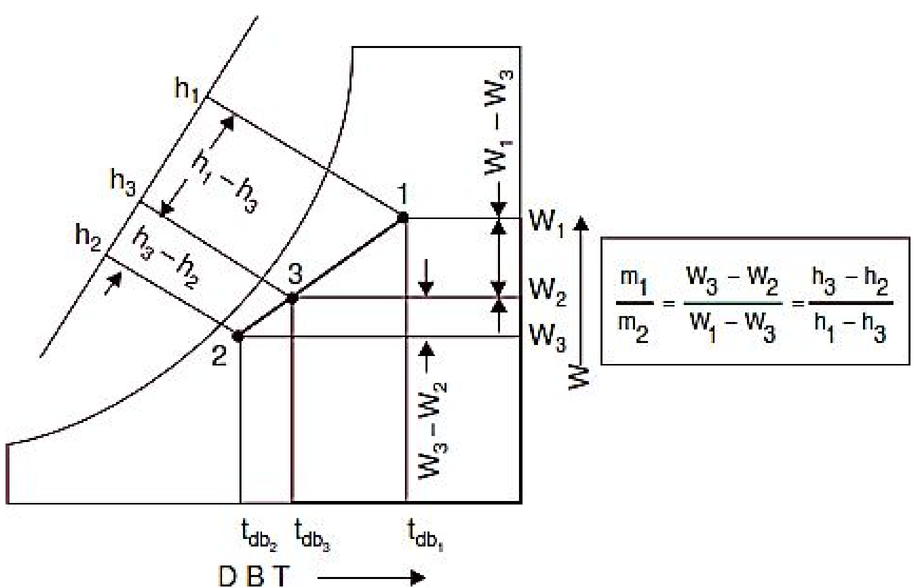


Fig D

Rearranging of last two equations gives the following:

$$m_1(W_1 - W_3) = m_2(W_3 - W_2)$$
$$m_1(h_1 - h_3) = m_2(h_3 - h_2)$$
$$\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3} = \frac{h_3 - h_2}{h_1 - h_3}$$

Where,

m = Mass of dry air at particular state points W=

Specific humidity at particular state points h =

Enthalpy at particular state points

On the psychrometric chart, the specific humidity and enthalpy scales are linear, ignoring enthalpy deviations. Therefore, the final state 3 lies on a straight line connecting the initial states of the two streams before mixing, and the final state 3 divides this line into two parts that are in the same ratio as were the two masses of air before mixing. If the air quantities are known in volume instead of mass units, it is generally sufficiently accurate to units of m^3 or m^3/min. in the mixing equations. The inaccuracy introduced is due to the difference in specific volume at two initial states. This difference in densities is small for most of the comfort air conditioning problems.

Sensible Heating

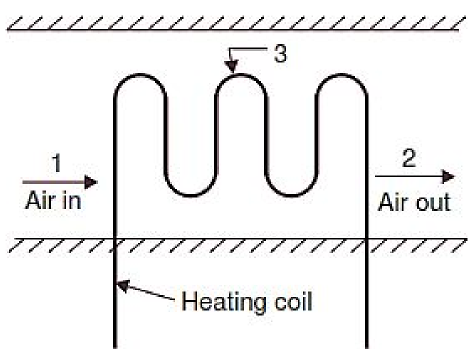
When air passes over a dry surface which is at a temperature greater than its (air) dry bulb temperature, it undergoes sensible heating. Thus the heating can be achieved by passing the air over heating coil like electric resistance heating coils or steam coils. During such a process, the specific humidity remains constant but the dry bulb temperature rises and approaches that of the surface. The extent to which it approaches the mean effective surface temperature of the coil is conveniently expressed in terms of the equivalent **by-pass factor.**

The by-pass factor (BF) for the process is defined as the ratio of the difference between the mean surface temperature of the coil and leaving air temperature to the difference between the mean surface temperature and the entering air temperature.

Thus on Fig. E, air at temperature t_{db1}, passes over a heating coil with an average surface temperature t_{db3} and leaves at temperature t_{db2}

The by-pass factor is expressed as follows :

$$BF = \frac{t_{db_3} - t_{db_2}}{t_{db_3} - t_{db_1}}$$



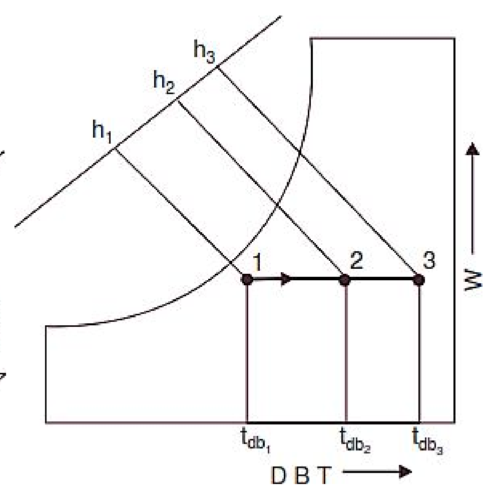


Fig.E. Sensible heating. Fig.F

Or in terms of lengths on the chart (Fig. F) it is $\frac{\text{length } 2-3}{\text{length } 1-3}$

The value of the by-pass factor is a function of coil design and velocity. The heat added to the air can be obtained directly from the entering and leaving enthalpies (h_2-h_1) or it can be obtained from the humid specific heat multiplied by the temperature difference (t_{db2}-t_{db1}).

In a complete air conditioning system the preheating and reheating of air are among the familiar examples of sensible heating.

Note. By-pass factor can be considered to represent the fraction of air which does not come into contact with coil surface.

Sensible Cooling

Refer Fig. G. Air undergoes sensible cooling whenever it passes over a surface that is ata temperature less than the dry bulb temperature of the air but greater than the dew point temperature. Thus sensible cooling can be achieved by passing the air over cooling coil like evaporating coil of the refrigeration cycle or secondary brine coil.

During the process, the specific humidity remains constant and dry bulb temperature decreases, approaching the mean effective surface temperature. On a psychrometric chart the process will appear as a horizontal line 1–2(Fig. H), where point 3 represents the effective surface temperature. For this process:

$$\text{By-pass factor BF} = \frac{t_{db_2} - t_{db_3}}{t_{db_1} - t_{db_3}}$$

The heat removed from air can be obtained from the enthalpy difference (h_1-h_2) or from humid specific heat multiplied by the temperature difference(t_{db1}-t_{db2}).

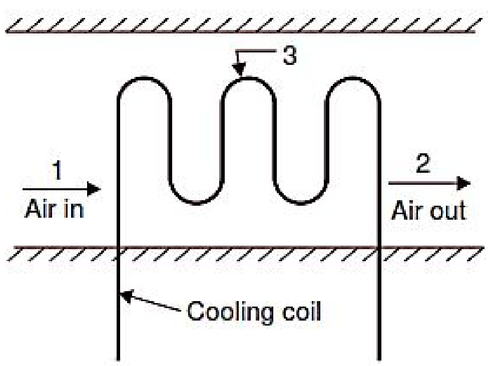


Fig.G. Sensible cooling.

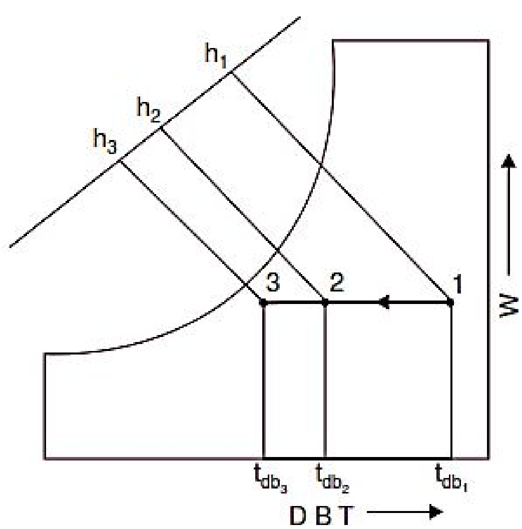


Fig.H

Cooling and Dehumidification

Refer Fig. I. Whenever air is made to pass over a surface or through a spray of water that is at a temperature less than the dew point temperature of the air, condensation of some of the water vapour in air will occur simultaneously with the sensible cooling process. Any air that comes into sufficient contact with the cooling surface will be reduced in temperature to the mean surface temperature along a path such as 1-2-3 in Fig. I, with condensation and therefore dehumidification occurring between points 2 and 3. The air that does not contact the surface will be finally cooled by mixing with the portion that did, and the final state point will somewhere on the straight line connecting points 1 and 3. The actual path of air during the path will not be straight line shown but will be something similarly to the curved dashed line 1–4.

It will result from a continuous mixing of air which is connecting a particular part of the coil and air which is by passing it. It is convenient, however to analyse the problem with the straight line shown, and to assume that the final air state results from the mixing of air that has completely by passed the coil with air that has been cooled to the mean effective surface temperature. If there is enough contact between air and surface for all the air to come to the mean surface temperature, the process is one of zero by pass. In any practical system, complete saturation is not obtained and final state will be a point such as 4 in Fig. I with an equivalent by pass factor equal to $\frac{\text{length } 3-4}{\text{length } 3-1}$ or processes involving condensation, the effective surface temperature, e.g t_{db3}i

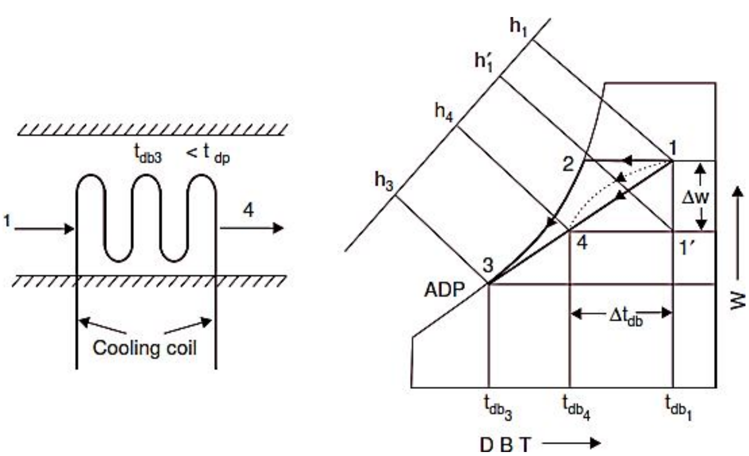


Fig. I Cooling and dehumidification

Shown in Fig. I is called **apparatus dew point**' (ADP). The final state point of air passing through a cooling and dehumidifying apparatus is in effect a mixture condition that results from mixing the fraction of the air, which is equal to the equivalent by- pass factor (BF) and is at initial state point and the remaining fraction which is equal to one minus by pass factor (1–BF) and is saturated at the apparatus dew point (ADP).

Total heat removed from the air is given by

$$Q_t = h_1 - h_4 = (h_1 - h_1') + (h_1' - h_4)$$
$$= Q_L + Q_S$$

where, W_L = Latent heat removed $(h_1 - h_1')$, and

Q_S = Sensible heat removed $(h_1' - h_4)$

The ratio $\frac{Q_S}{Q_L}$ *is called sensible heat factor* (SHF) Or *sensible heat ratio* (SHR)

:

$$\text{SHF} = \frac{Q_S}{Q_L + Q_S}$$

The ratio fixes the slope of the line 1—4 on the psychrometric chart. Sensible heat factor slope lines are given on the psychrometric chart. If the initial condition and SHF are known for the given process, then the process line can be drawn through the given initial condition at a slope given by SHF on the psychrometric chart.

The capacity of the cooling coil in *tonnes* of refrigeration is given by,

$$\text{Capacity in TR} = \frac{m_a(h_1 - h_4) \times 60}{14000},$$

Where m_a= mass of air, kg/min and h = enthalpy in kJ/kg of air.

Cooling and Humidification

If unsaturated air is passed through a spray of continuously recirculated water, the specific humidity will increase while the dry bulb temperature decreases. This is the process of **adiabatic saturation or evaporative cooling.** This process is one of constant adiabatic- saturation temperature and for all practical purposes, one of constant wet bulb temperature. The process is illustrated as path 1-2 on Fig. J, with wet bulb temperature of air being that of point 3, which is also equilibrium temperature of the recirculated water if there is sufficient contact between air and spray, the air will leave at a condition very close to that of point 3. The concept of equivalent by pass can be applied to this process but another term is more used to describe the performance of a humidifying apparatus. It is the **'saturating'** or **'humidifying efficiency'** which is defined as the ratio of dry-bulb temperature decrease to the entering wet bulb depression usually expressed as percentage. Thus, from Fig. J, the saturating efficiency is :

$$\% \eta_{sat} = \left(\frac{t_{db_1} - t_{db_2}}{t_{db_1} - t_{db_3}} \right) \times 100$$

As a fraction, it is equal to one minus the by pass factor for the process. This adiabatic process, for all practical purposes, is line of constant enthalpy. The moisture added can be obtained from the increase in specific humidity.

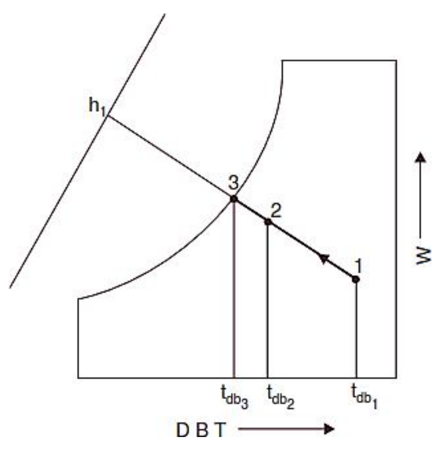


Fig J Cooling and humidification.

Heating and Dehumidification

If air is passed over a solid absorbent surface or through a liquid absorbent spray simultaneous heating and dehumidification is accompanied. In either case the dehumidification results from adsorbent or absorbent having a lower water vapour pressure than air. Moisture is condensed out of the air, and consequently the latent heat of condensation is liberated, causing sensible heating of air. If these were the only energies involved, the process would be the inverse of the adiabatic saturation process. There is, however, an additional energy absorbed or liberated by the active material, termed the heat of adsorption or absorption. For the solid adsorbents used commercially, such as silica gel or activated alumina, and for the more common liquid absorbents, such as solutions of organic salts or inorganic compounds like ethylene, glycol, heat is involved and results in additional sensible heating. Thus the path lies above a constant wet bulb line on the psychrometric chart such as path 1-2 in Fig. K

Heating and Humidification

If air is passed through a humidifier which has heated water sprays instead of simply recirculated spray, the air is humidified and may be heated, cooled or unchanged in temperature. In such a process the air increases in specific humidity and the enthalpy, and the dry bulb temperature will increase or decrease according to the initial temperature of the air and that of the spray. If sufficient water is supplied relative to the mass flow of air, the air will approach saturation at water temperature. Examples of such processes are shown on Fig. L

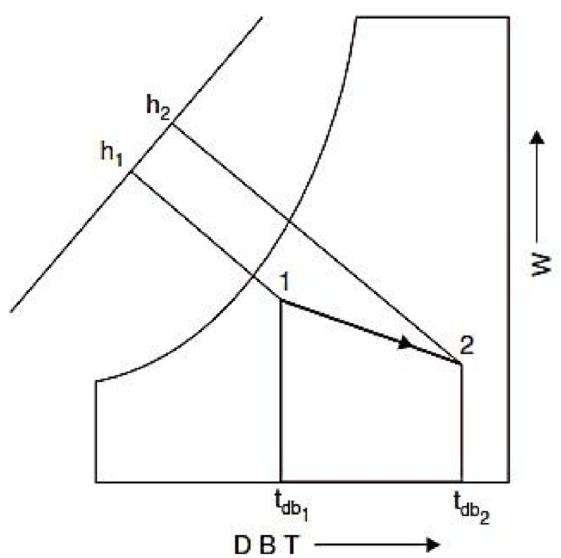


Fig. K. Heating and dehumidification.

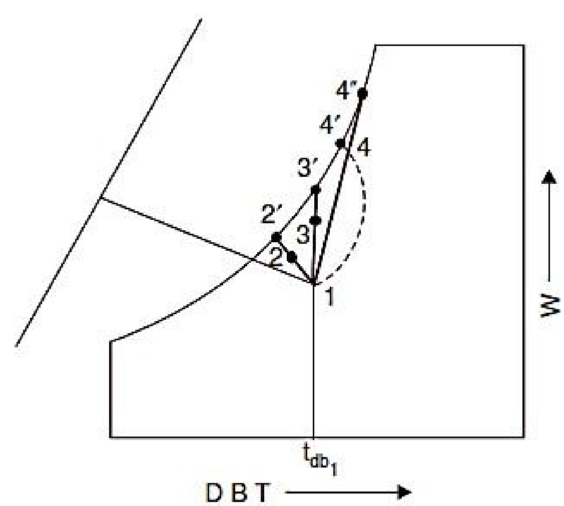


Fig. L. Heating and humidification.

Process 1-2 : It denotes the cases in which the temperature of the heated spray water is less than the air DBT

Process 1-3 : It denotes the cases in which the temperature is equal to the air DBT. Process 1-4 : It denotes the cases in which a spray temperature is greater than air DBT.

As in the case of adiabatic saturation, the degree to which the process approaches saturation can be expressed in terms of the by-pass factor or a saturating efficiency.

If the water rate relative to the air quantity is smaller, the water temperature will drop significantly during the process. The resultant process will be a curved line such as the dashed 1-4 where 4 represents the leaving water temperature.

Note. It is possible to accomplish heating and humidification by evaporation from an open pan of heated water, or by direct injection of heated water or steam. The latter is more common. The process line for it is of little value because the process is essentially an instantaneous mixing of steam and the air. The final state point of the air can be found, however by making a humidity and enthalpy balance for the process. The solution of such a problem usually involves cut-and-try procedure.

UNIT –V – Refrigeration and Air Conditioning – SME1618

V. Accessories of Refrigeration and Air Conditioning Systems

Cooling towers overcome most of these problems and therefore are commonly used to dissipate heat from water-cooled refrigeration, air-conditioning, and industrial process systems. The water consumption rate of a cooling tower system is only about 5% of that

of a once-through system, making it the least expensive system to operate with purchased water supplies. Additionally, the amount of heated water discharged (blowdown) is very small, so the ecological effect is greatly reduced. Lastly, cooling towers can cool water to

within 4 to 5°F of the ambient wet-bulb temperature, or about 35°F lower than can air-cooled systems of reasonable size. This lower temperature improves the efficiency of the overall system, thereby reducing energy use significantly and increasing process

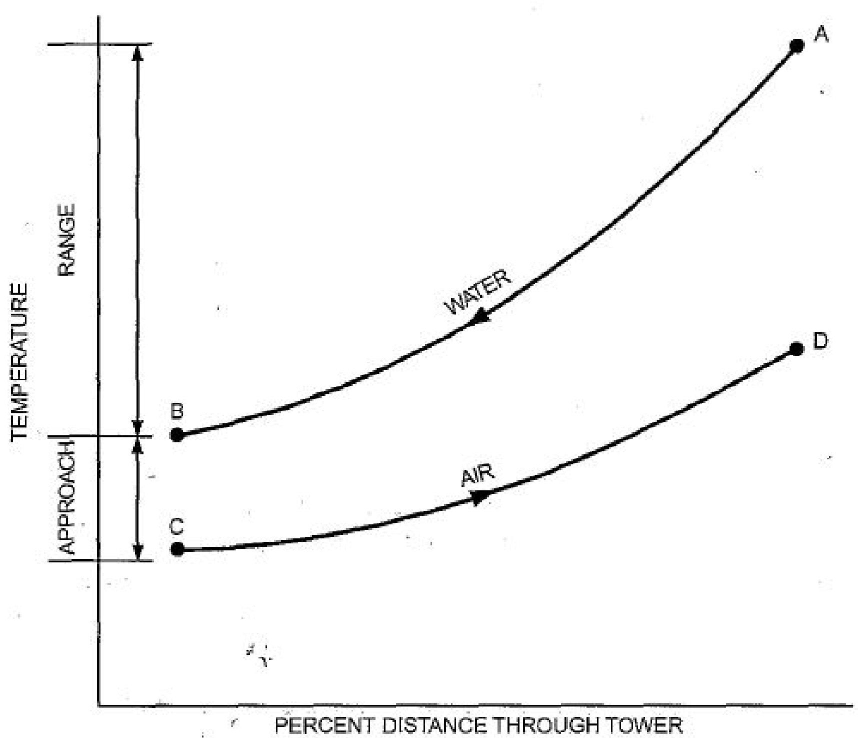


Fig. 1 Temperature Relationship Between Water and Air in Counterflow Cooling Tower

output.

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 (1) fans, (2) convective currents, (3) natural wind currents, or (4) 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 airstream. Figure 1 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 (A to B) and the rise in the air wet-bulb temperature (C to 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. For a Counter flow Cooling Tower steady-state system, the range is the same as the water temperature rise through the load heat exchanger, provided the flow rate through the cooling tower and heat exchanger are the same. Accordingly, the range is determined by the heat load and water flow rate, not by the size or thermal capability of the cooling tower. The difference between the leaving water temperature and entering air wet-bulb temperature (B minus C) in Figure 1 is the approach to the wet bulb or simply the approach of the cooling tower. The approach is a method 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. Thus, the amount of heat transferred to the atmosphere by the cooling tower is always equal to the heat load imposed on the tower, whereas the temperature level at which the heat is transferred is determined by the thermal capability of the cooling tower and the

entering air wet-bulb temperature. 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 do affect the rate of water evaporation in the cooling tower. A psychrometric analysis of the air passing through a cooling tower illustrates this effect (Figure 2). Air enters at the ambient condition Point A, absorbs heat and mass (moisture) from the water. and exits at Point B in a saturated condition (at verv light d " loads, the discharge air may not be fully saturated). The amount of The preparation of this chapter is assigned to TC 8.6, Coollng Towers and heat transferred from the water to the air is proportional to the Evaporative Condensers. Reference in enthalpy of the air between the entering and leaving conditions (hB - hA). Because lines of constant enthalpy correspond

almost exactly to lines of constant wet-bulb temperature, the change in enthalpy of the air may be determined by the change in wet-bulb temperature of the air.

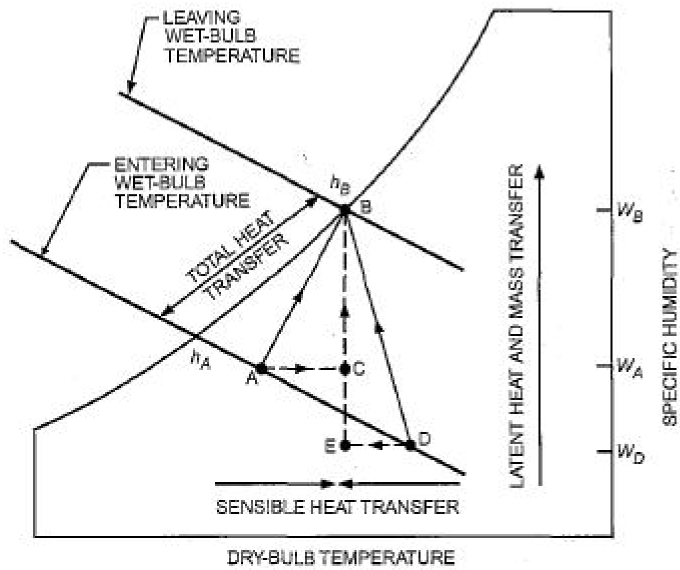


Fig. 2 Psychrometric Analysis of Air Passing Through Cooling Tower

Two basic types of evaporative cooling devices are used. The first of these, the direct-contact or open cooling tower (Figure 3), exposes water directly to the cooling atmosphere, thereby transferring the source heat load directly to the air. The second type, often called a closed-circuit cooling tower, involves indirect contact between heated fluid and atmosphere (Figure 4), essentially combining a heat exchanger and cooling tower into one relatively compact device.

Of the direct-contact devices, the most rudimentary is a sprayfilled 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 spray efficiency, and the time of contact depends on the elevation aid pressure ofthe water distribution system.

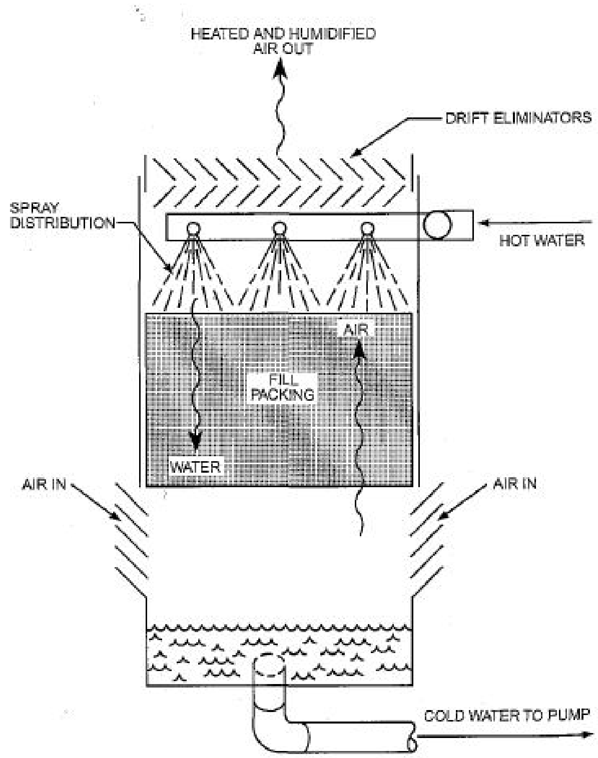


Fig. 3 Direct-Contact or Open Evaporative Cooling Tower

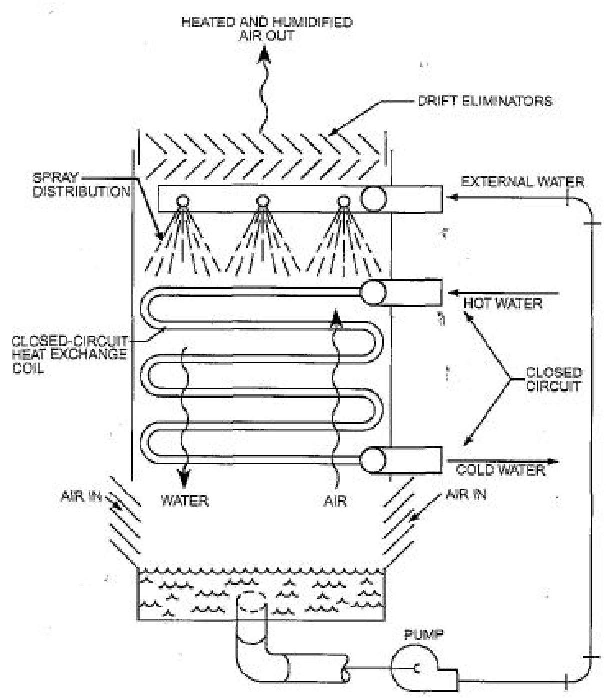


Fig. 4 Indirect-Contact or Closed-Circuit Evaporative Cooling Tower

Ducts for Air Conditioning Systems

Fig 5 Duct design Photograph

Air flow problems have plagued the HVAC industry for years. No matter how much money you spend on a high-quality HVAC system, the equipment won't work at its best without properly designed and installed ductwork. Ducts that are not well designed result in discomfort, high energy costs, bad air quality, and increased noise levels.

A well-designed ductwork system should deliver maximum interior comfort at the lowest operating cost while also preserving indoor air quality. The chief requirements of an air conditioning duct system are:

1. It should convey specified rates of air flow to prescribed locations.
2. It should be economical in combined initial cost, fan operating cost and cost of building space.
3. It should not transmit or generate objectionable noise.

A primary issue is the tradeoff between the initial cost of the duct system and the energy cost of the air distribution system. Larger ducts require a larger initial investment, but result in lower fan energy costs over the life of the system. Other issues include space restrictions, noise level, capacity for expansion, appearance, etc.

This course will discuss the basic fundamentals and principles of air conditioning duct design and layout.

DUCTWORK DESIGN PRINCIPLES

Starting with the basics, let's start at the most elementary level of air flow fundamentals.

Basic Definitions

The following basic terminology is extensively used in this course.

- **cfm:** volume of air flow; cubic feet/minute
- **fpm:** velocity or speed of air flow; feet/minute
- **sq.ft:** duct size or cross-sectional area; square feet

Air volume in cfm can be calculated by multiplying the air velocity by the cross-sectional area of the duct in square feet.

- cfm = fpm x Area

Given any two of these three quantities, the third can be readily determined:

- fpm = cfm/area
- Area = cfm/fpm

Gauge and Absolute Pressures:

Gauge pressure is indicated on the gauge; absolute pressure is the total of the indicated gauge pressure plus atmospheric pressure. The general equation for absolute pressure is:

Gauge pressure + atmospheric pressure = absolute pressure

For example, if the gauge reads 10 psig then, using the above equation, the absolute pressure would be 24. 7 psia:

10 psig + 14.7 psi = 24.7psia

Ordinary heating, ventilating, and air conditioning duct systems read air pressures at 0.4 psi or less, often much less. 1 psi equals 27.7 inches of water gauge; a common duct pressure of 0.25 inches water column is equal to (0.25 divided by 27.7 in-wc/psi) = 0.009 psi.

Duct Pressure:

Duct system is pressurized by three pressures:

- **Static pressure:** It is the air pressure in the duct, which is used for fan selection.
- **Velocity pressure:** It is the pressure generated by the velocity and weight of the air, which is used for measuring the flow (cfm) in a system.
- **Total pressure:** It is used to find velocity pressure. Static pressure plus velocity pressure equals total pressure.

Pressure in the ductwork is measured in inches of water column (in-wc).

Standard Air Density:

Air has mass. Standard air has a density of 0.075 lbs/ ft^3.

System capacity is directly affected by changes in air flow. As air is heated or humidified, its specific volume increases and its density decreases. If the air density is low, more

cfm is required to keep the mass flow rate the same. If air density is not considered, many systems will have very low air flow.

Correction for the density is however not needed in air conditioning or cooling applications, if the temperature is between 40°F to 100°F and up to 1000 ft. in elevation.

Fan Capacity:

The volume of air will not be affected in a given system because a fan will move the same amount of air regardless of the air density. In other words, if a fan will move 3,000 cfm at 70°F, it will also move 3,000 cfm at 250°F

Volumetric Air Flow Rate:

The volumetric flow rate of air that will be conveyed through the duct in an air conditioning system is determined by the cooling/heat load and the desired supply air temperature. Since we are not conditioning cfms of air but rather pounds of it, we need a mass-balance equation:

$$Q\,[\frac{Btu}{h}] = \dot{m}\left[\frac{lb}{hr}\right] c_p[\frac{Btu}{°F * lb}]\Delta T[°F]$$

$$Q\left[\frac{Btu}{h}\right] = CFM * \left(60\,\frac{min}{hr}\right) * (\frac{049lbm}{ft^3})0.24[\frac{Btu}{°F * lb}]\Delta T[°F]$$

$$air\ conditions\ at\ 70°F\ and\ 1\ atm.$$

$$Q\left[\frac{Btu}{h}\right] = 1.08 * CFM * \Delta T[°F]$$

It is important that the air conditioning ductwork system delivers and return the right amount of air from each room and provide comfort year round. This implies room by room heat loss and heat gain calculations.

Air Flow Principles

Flow of air is caused as a result of pressure differential between two points. Flow will originate from an area of high energy (or pressure) and proceed to area(s) of lower energy.

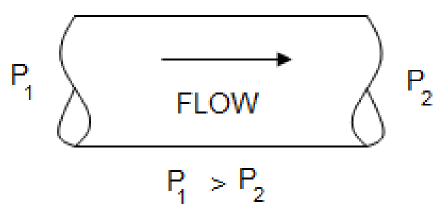


Fig 6 Air flow movement

Air moves according to three fundamental laws of physics: conservation of mass, conservation of energy, and conservation of momentum.

1. **Conservation of mass:** It simply states that an air mass is neither created nor destroyed. From this principle it follows that the amount of air mass coming into a junction in a ductwork system is equal to the amount of air mass leaving the junction, or the sum of air masses at each junction is equal to zero. In most cases the air in a duct is assumed to be incompressible, an assumption that overlooks the change of air density that occurs as a result of pressure loss and flow in the ductwork. In ductwork, the law of conservation of mass means a duct size can be recalculated for a new air velocity using the simple equation:

   $V_2 = (V_1 * A_1)/A_2$

   where V is velocity and A is Area

2. **The law of energy conservation:** It states that energy cannot disappear; it is only converted from one form to another. This is the basis of one of the main expression of aerodynamics, the Bernoulli equation. Bernoulli's equation in its simple form shows that, for an elemental flow stream, the difference in total pressures between any two points in a duct is equal to the pressure loss between these points, or:

   $(\text{Pressure loss})_{1\text{-}2} = (\text{Total pressure})_1 - (\text{Total pressure})_2$

3. **Conservation of momentum:** It is based on Newton's law that a body will maintain its state of rest or uniform motion unless compelled by another force to change that state. This law is useful to explain flow behavior in a duct system's fitting.

Total Pressure, Velocity Pressure, and Static Pressure

Air flow through a duct system creates three types of pressures: static, dynamic (velocity), and total.

1. **Static pressure:** Static Pressure is the pressure that causes air in the duct to flow. Static pressure is the outward push of air against duct surfaces and is a measure of resistance when air moves through an object like duct work. Measured in inches of water column (in-wc), it acts equally in all directions and is independent of velocity.

2. **Velocity pressure:** Velocity pressure is the pressure caused by air in motion. It is equal to the product of air density and the square of the velocity divided by 2.

   $$VP = 0.5 \times \rho \times v^2$$

   Using standard air, the relationship between V and VP is given by:

   $$VP = \left(\frac{V}{4005}\right)^2$$

VP will only be exerted in the direction of air flow and is always positive.

3. **Total Pressure:** Total pressure is the algebraic sum of velocity pressure and static pressure.

   $$TP = VP + SP$$

   - **TP** = Total Pressure
   - **VP** = Velocity Pressure
   - **SP** = Static Pressure

Air Flow Characteristics in Duct

1. At any point, the total pressure is equal to the sum of the static and velocity pressures.
2. The static pressure is exerted equally in all directions and the velocity pressure is exerted only in the direction of air flow. This makes it difficult to directly measure velocity pressure in a duct. Simply put, because static pressure is also pushing in the direction of air flow, you can never measure just velocity pressure. Practically, velocity pressure is calculated by measuring pressure perpendicular to the air flow (Static Pressure) and also measuring pressure parallel to the air flow (Total Pressure).

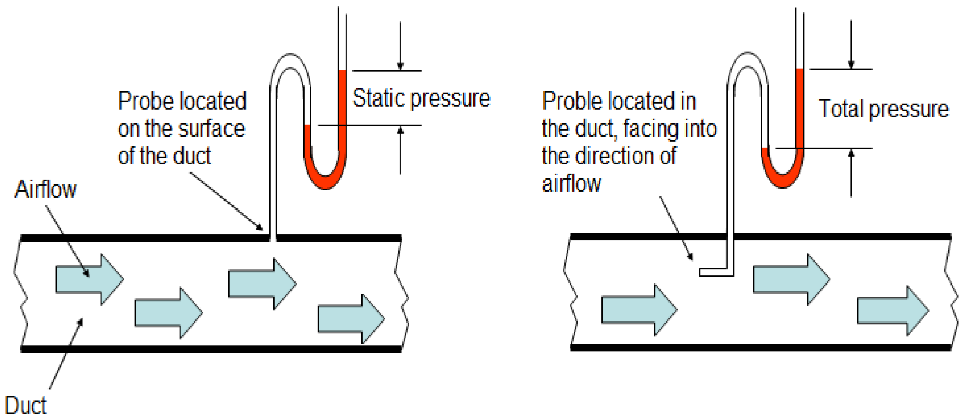


Fig 7 Total pressure variation

Once you have these two values you can just subtract static pressure from the total pressure and derive the velocity pressure. VP = TP - SP

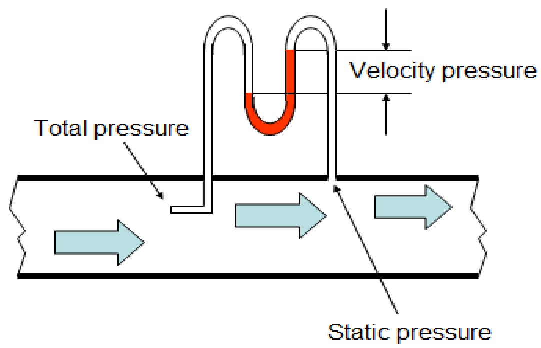


Fig 8 Static pressure variation

3. Static and velocity pressure are mutually convertible. The magnitude of each is dependent on the local duct cross-section which determines the flow velocity. The following pressure changes are affected in the ducts:

   – **Constant cross-sectional areas:** Total and static losses are equal.

   – **Diverging sections (increase in duct size):** Velocity pressure decreases, total pressure decreases, and static pressure may increase (static regain).

   – **Converging sections (decrease in duct size):** Velocity pressure increases in the direction of flow, total and static pressure decrease.

4. The total pressure generally drops along the air flow because of frictional and turbulence losses.

Confusion in the use of the terms “Static Pressure”, “Velocity Pressure” and “Total Pressure” is widely prevalent among HVAC engineers and contractors. The term “Static Pressure” is typically used for fan selection; “Velocity Pressure” is used for measuring cfm in a system, and “Total Pressure” is used to find the velocity pressure. Total Pressure determines the actual mechanical energy that must be supplied to the system.

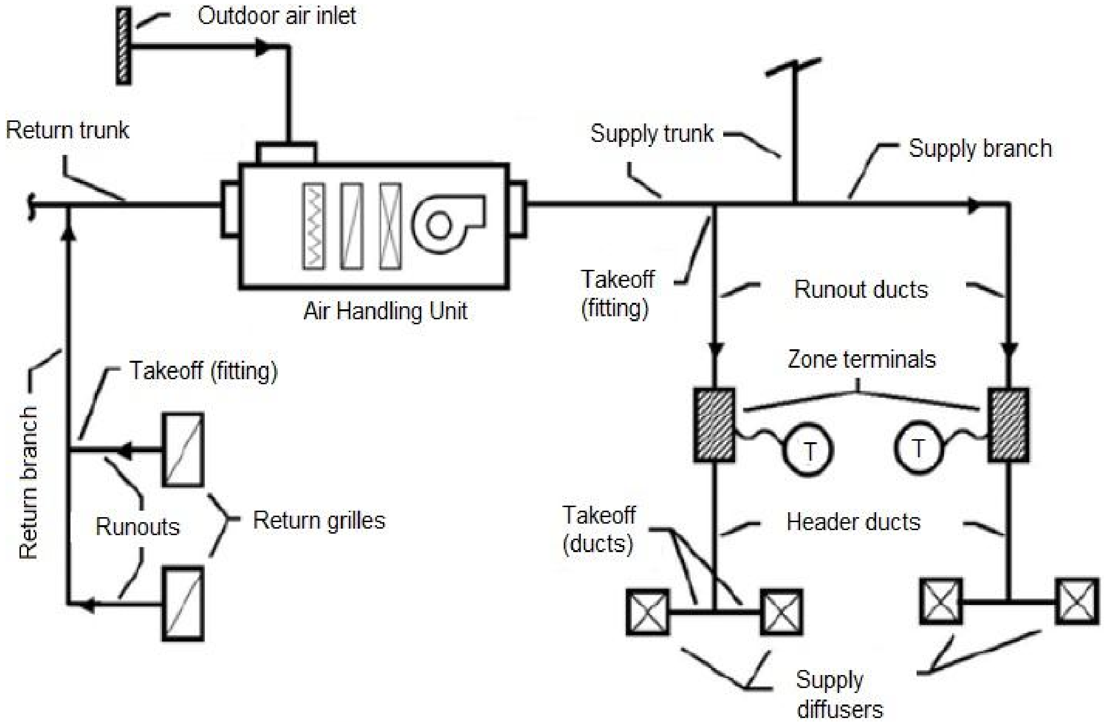
DUCT COMPONENTS & MATERIALS

The air distribution system will have a designation depending on the function of the duct. Broadly, there are five designations of ducts:

1. Supply air ductwork supplies conditioned air from the air handling unit to the conditioned area.
2. Return air ductwork removes air from the conditioned building spaces and returns the air to the air handling unit, which reconditions the air. In some cases, part of the return air in this ductwork is exhausted to the building exterior.
3. Fresh air ductwork supplies outdoor air to the air handling unit. Outdoor air is used for ventilating the occupied building space.
4. Exhaust (relief) air ductwork carries and discharges air to the outdoors. Exhaust air is taken from toilets, kitchen, laboratories and other areas requiring ventilation.
5. Mixed air ductwork mixes air from the outdoor air and the return air then supplies this mixed air to the air handling unit.

Duct Components

The figure below shows a schematic and a 3-D representation of supply and return air ductwork. The central air handling unit (AHU) is connected to the air plenum at the starting point. AHU fans draw in air through grilles called returns and force air through the plenum and into the conditioned space through supply registers.



Air conditioning Schematic Drawing

Fig 9 Air Conditioing Schematic Diagram

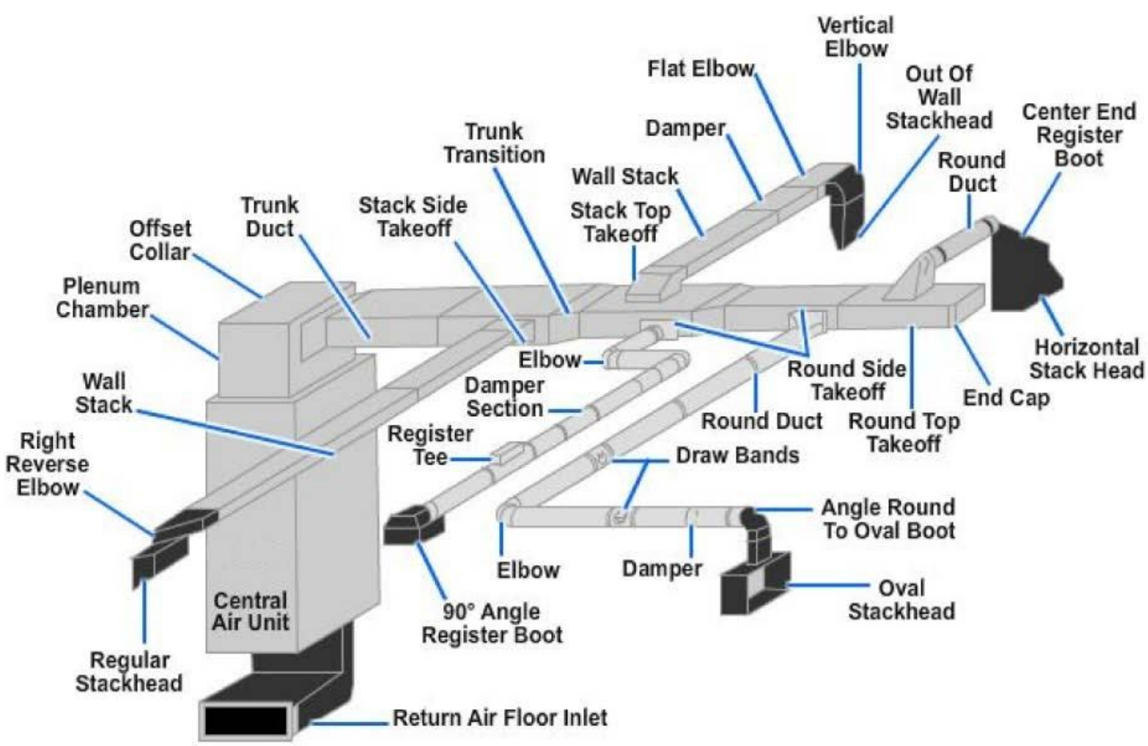


Fig 10. Components of Ducts

The duct components are as follows:

Plenum or Main Trunk: The plenum is the main part of the supply and return duct system that goes directly from the air handler to the “Trunk Duct”.

Trunk Duct: A duct that is split into more than one duct is called a “trunk”, just like a tree. Ducts that are on the end of a trunk and terminate in a register are called branches.

Take Off: Branch ducts are fastened to the main trunk by a takeoff-fitting. The takeoff encourages the air moving the duct to enter the takeoff to the branch duct.

Air Terminals Devices: Air terminals are the supply air outlets and return or exhaust air inlets. For supply, diffusers are most common, but grilles and registers are also used widely. A diffuser is an outlet device discharging supply air in a direction radially to the axis of entry. A register is a grille equipped with a volume control damper. A grille is without a damper.

Duct Materials

Ducting is generally formed by folding sheet metal into the desired shape. Traditionally, air conditioning ductwork is made of galvanized steel, next in popularity is aluminum. Other metals used under special circumstances are copper and stainless steel. Metals that are used extensively depend on the application of the duct and are listed below:

1. **Galvanized Steel:** It is a standard, most common material used in fabricating ductwork for most comfort air conditioning systems. The specifications for galvanized steel sheet are ASTM A653, coating G90.

2. **Aluminium:** It is widely used in clean room applications. These are also preferred systems for moisture laden air, special exhaust systems and ornamental duct systems. The specifications for Aluminium sheet are ASTM B209, alloy 1100, 3003 or 5052.

3. **Stainless Steel:** It is used in duct systems for kitchen exhaust, moisture laden air, and fume exhaust. The specifications for stainless steel sheet are ASTM A167, Class 302 or 304, Condition A (annealed) Finish No. 4 for exposed ducts and Finish No.

2B for concealed duct.

4. **Carbon Steel (Black Iron):** It is widely used in applications involving flues, stacks, hoods, other high temperature and special coating requirements for industrial use.
5. **Copper:** It is mainly used for certain chemical exhaust and ornamental ductwork.

Pressure in the air conditioning ducts is small, so materials with a great deal of strength are not needed. The thickness of the material depends on the dimensions of the duct, the length of the individual sections, and the cross-sectional area of the duct.

Non-Metallic ducts

This category includes ducts made from plastic or foam boards, shaped by cutting and folded to produce the required cross-sectional geometry. Boards are faced usually with an aluminum coating, both internal and external.

The main drawback of this type of ducting is its fire classification. Even if it complies with local standards, when exposed to fire, it often exhibits poor performance in terms of the production of both smoke and flaming droplets.

1. **Fibreglass Reinforced Plastic (FRP):** It is used mainly for chemical exhaust, scrubbers, and underground duct systems. Advantages are resistance to corrosion, self-insulation, excellent sound attenuation and high quality sealing. Limiting characteristics include cost, weight, range of chemical and physical properties, and code acceptance.
2. **Polyvinyl Chloride (PVC):** It is used for exhaust systems for chemical fumes and underground duct systems. Advantages include resistance to corrosion, light weight, and ease of modification. Limiting characteristics include cost, fabrication, code acceptance, thermal shock, and weight.
3. **Fabric:** Fabric ducting, also known as textile ducts, is usually made of special permeable polyester material and is normally used where even air distribution is essential. Due to the nature or the air distribution, textile ducts are not usually concealed within false ceilings. Condensation is not a concern with fabric ducts and therefore these can be used where air is to be supplied below the dew point without insulation.
4. **Flex Duct:** Flex ducts consist of a duct inner liner supported on the inside by a

helix wire coil and covered by blanket insulation with a flexible vapor barrier jacket on the outside. Flex ducts are often used for runouts, as well as with metal collars used to connect the flexible ducts to supply plenums, trunks and branches constructed from sheet metal or duct board. Flex ducts provide convenience of installation as these can be easily adapted to avoid clashes but has certain disadvantages. These have more friction loss inside them than metal ducting. Flex duct runs should be as short as possible (5 to 6 ft. max.) and should be stretched as tight as possible.

DUCT CLASSIFICATION

Ducts are classified in terms of velocity and pressure.

Velocity Classification

Ducts are classified into 3 basic categories:

1. **Low Velocity Systems:** They are characterized by air velocities up to 2000 fpm.
2. **Medium Velocity Systems:** They are characterized by air velocities in the range of 2,000 to 2,500 fpm.
3. **High Velocity Systems:** They are characterized by air velocities greater than 2,500 fpm.

The low velocity system is used in most air conditioning installations because it is quieter, has lower friction losses, lower fan power, and lower air leakage.

High duct velocities result in lower initial costs but require increased fan static pressures; therefore, resulting in increased operating costs. Often these need additional noise attenuation (use of noise silencers) and are not suitable for comfort applications.

Generally, high-velocity systems are applicable to large multi-story buildings, primarily because the advantage of savings in duct shafts and floor-to-floor heights is more substantial. Small two- and three-story buildings are normally low velocity. A velocity of 1,000 to 1,500 fpm for main ducts and a velocity of 700 to 1,000 fpm for the branch take offs are recommended.

Pressure Classification

Duct systems are also divided into three pressure classifications, matching the way supply fans are classified.

1. **Low Pressure:** The term low-pressure applies to systems with fan static pressures less than 3 inches WC. Generally, duct velocities are less than 1,500 fpm.
2. **Medium Pressure:** The term medium pressure applies to systems with fan static pressures between 3 to 6 inches WC. Generally, duct velocities are less than or equal to 2,500 fpm.
3. **High Pressure:** The term high pressure applies to systems with fan static pressures between 6 to 10 inches WC. Usually the static pressure is limited to a maximum of 7 inches WC, and duct velocities are limited to 4,000 fpm. Systems requiring pressures more than 7 inches WC are normally unwarranted and could result in very high operating costs.

General good engineering practices are:

1. Use of medium pressure classification for primary air ductwork (fan connections, risers, and main distribution ducts).
2. Use of low pressure classification for secondary air ductwork (run- outs/branches from main to terminal boxes and distribution devices).

Velocity Classification vs. Pressure Classification

1. Duct pressure classification influences the duct strength, deflection and air leakage.
2. Duct velocity classification influences noise, vibration, friction losses and fan power.

4.0. DUCT SHAPES

Ducts commonly used for carrying air are of round, square, or rectangular shape. All have advantages and disadvantages, and find applications where one is definitely superior to the other.

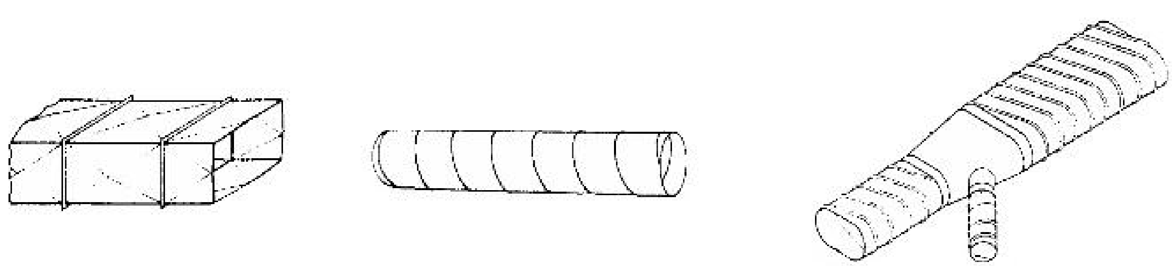


Fig 11. Duct Shapes

Round Ducts

The duct shape that is the most efficient (offers the least resistance) in conveying moving air is a round duct, because it has the greatest cross-sectional area and a minimum contact surface. In other words, it uses less material compared to square or rectangular ducts for the same volume of air handled.

An 18 inch diameter duct, for example, has the same air-carrying capacity as a 26” x 11” rectangular duct. The round duct has a cross-sectional area of 254.5 sq.-in and a perimeter of 4.7 ft., while the rectangular duct has a 286 sq.-in area and a perimeter of 6.2 ft. The rectangular duct thus has 32% more metal in it and would cost proportionately more. Also the insulation, supports and labor are higher for rectangular ducts of similar capacity.

Some of the advantages of round ductwork include:

- Round shape results in lower pressure drops, thereby requiring less fan horsepower to move the air and, consequently, smaller equipment.
- Round shape also has less surface area and requires less insulation when externally wrapped.
- Round ducts are available in longer lengths than rectangular ducts, thereby eliminating costly field joints. Spiral lock-seams add rigidity; therefore, spiral ducts can be fabricated using lighter gauges than longitudinal seam ducts. Spiral ducts leak less and can be more easily sealed compared to rectangular ducts.
- The acoustic performance of round and oval ducts is superior because their curved surfaces allow less breakout noise. The low-frequency sound is well contained in round ducts.
- Round ducts can help promote healthier indoor environments. Less surface area, no corners and better air flow reduce the chance of dirt and grime accumulating inside the duct and, therefore, becoming a breeding ground for bacterial growth.

While round air ducts have great advantages, there are some disadvantages to them. One of the most notable drawbacks of round air ducts is that they need more clear height for installation. If the net clear height of a furred space above a suspended ceiling is 14 inches, an 18-in diameter duct cannot be installed therein; however, its equivalent 26” x 11” rectangular duct will fit the space easily. A combination of a rectangular plenum and round branches sometimes is a good compromise.

Rectangular Ducts

Square or rectangular ducts fit better to building construction. They fit above ceilings and into walls, and they are much easier to install between joists and studs.

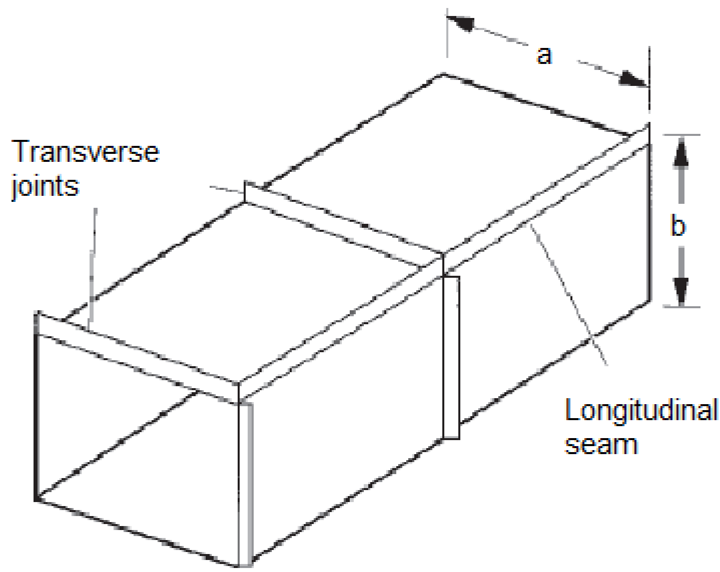


Fig 12. Rectangular Ducts

When rectangular ducts must be used due to space limitations, keep the width-to-height ratio (aspect ratio) low. A rectangular duct section with an aspect ratio close to 1 yields the most efficient rectangular duct shape in terms of conveying air. A duct with an aspect ratio above 4 is much less efficient in use of material and experiences great pressure losses. Aspect ratios of 2 to 3 are ideal in trading off added duct cost of material and fan energy for headroom savings.

Disadvantages of rectangular ducts are as follows:

1. They create higher pressure drop;
2. They use more pounds of metal for the same air-flow rate as round ducts;
3.

4. Their joint length is limited to the sheet widths stocked by the contractor;
5. Their joints are more difficult to seal;
6. Those with high aspect ratio can transmit excessive noise if not properly supported.

Oval Ducts

Flat oval ducts have smaller height requirements than round ducts and retain most of the advantages of the round ducts. However, fittings for flat oval ducts are difficult to fabricate or modify in the field. Other disadvantages include:

1. Difficulty of handling and shipping larger sizes;
2. Tendency of these ducts to become more round under pressure; and,
3. In large aspect ratios, difficulties of assembling oval slip joints.

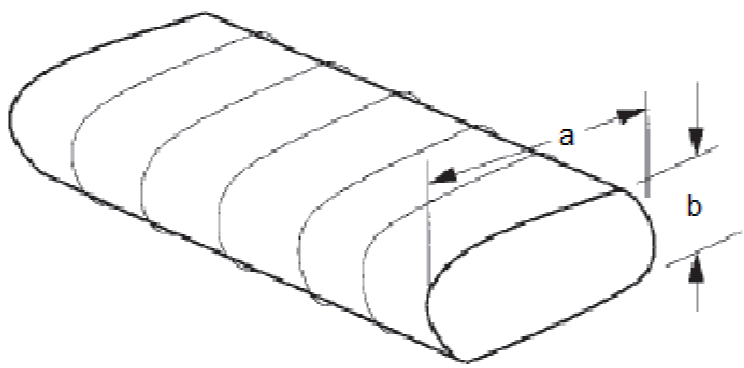


Fig 13. Oval Ducts

Equivalent Diameter

Since both round and rectangular ducts are extensively used in air conditioning systems, it is quite possible that a contractor may wish to substitute one for the other while working on new construction or modifying an existing system. With this likelihood, there is the general tendency to substitute cross-sectional areas of round and rectangular ducts. This is improper and will affect air distribution system performance. Therefore, it is necessary for the HVAC designer to fully understand the conditions under which round and rectangular ducts can be interchanged. The important thing is the duct pressure drop and that's where the concept of "equivalent diameter" comes into picture.

By definition, equivalent diameter (D_{eq}) is the diameter of a circular duct that will give the same pressure drop at the same air flow as the rectangular duct.

From ASHRAE Fundamentals Handbook, the following equations may be used to convert rectangular and flat oval ducts to and from round.

$$\text{Rectangular ducts: } D_{eq} = \frac{1.30\,(ab)^{0.625}}{(a+b)^{0.250}}$$

$$\text{Flat oval ducts : } D_{eq} = \frac{1.55\,A^{0.625}}{p^{\,0.250}}$$

$$A = \frac{\pi\, b^2}{4} + b\,(a - b)$$

$$p = \pi\, b + 2\,(a - b)$$

where,

- p = perimeter of oval duct (in.)
- A = cross-sectional area (sq.-in)
- a = length of major axis (in.)
- b = length of minor axis (in.)

Equivalent Diameter vs. Equivalent Cross-sectional Area Approach

Consider an air flow rate of 7,500 cfm and compare a 30” diameter round duct to equivalent rectangular and oval duct options.

Equivalent Diameter Approach

For a given round duct diameter (30 inches), the dimensions for rectangular and flat oval ducts must be solved for by trial and error. Fix one dimension and substitute in the equations above. Let’s use 16 inches for the minor axis, then the equivalent rectangular duct dimension will be 16”X 51”, and flat oval ducts with a 16-in. minor axis will be 16” X 53”.

What this means is that all three ducts, 30” round, 16” x 51” rectangular, and 16” x 53” flat oval will have the same friction loss for a given cfm. The table below summarizes the equivalent diameter approach.

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