قسم

المكائن و المعدات

مبادئ التبريد وتكييف الهواء

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المرحلة الأولى

The 1 st – 2 nd weeks

Property ;

A property of a system is any observable characteristic of a system . the properties we shall deal with are measurable in terms of numbers and units of measurements and include such physical quantities as location , pressure , density etc .

State ;

The state of a system is its condition or configuration , described in sufficient detail so that one state may be distinguished from all other states . the state may be identified by certain observable macroscopic properties .

Specific volume ;

It is defined as the volume per unit mass and may be expressed in m^3 / kg . it is the reciprocal of the density $v = 1/\rho$

Where (v) is the specific volume

Pressure ;

The pressure exerted by a system is the force exerted normal to a unit area of the boundary .

The standard atmospheric pressure is defined as the pressure produced by a column of mercury 760 mm high , the mercury density is 13.6 g/cm^3 .

And the acceleration due to gravity being its standard value of 980 cm/sc^3

The standard atmospheric pressure is 1.03 kgf / cm^2 and denoted by (atm)

Dry Bulb Temperature (T_d) :

it is the temperature recoded by a thermometer which is not effected by moisture or radiation .thus such a thermometer must have no moisture and must have dry surface .

Wet Bulb Temperature (T_w) :

when a mercury thermometer whose bulb is covered with muslin wick wetted with water , is moved past unsaturated air at high velocity of about 300 m/min the temperature reading obtained is the wet bulb temperature .

Specific Humidity (ω) :

it is the mass in kg of water vapour contained in the air – vapour mixture per kg of dry air

Relative Humidity (ϕ) :

It is the ratio of the mass of water vapour in air in a given volume at a given temperature to the mass of water vapour contained in the same volume at same temperature when the air is saturated .

Dew point temperature :

This corresponds to the saturation temperature as read from the steam tables for partial pressure of vapour .

Sensible heating process , sensible cooling process ;

Heating of air or cooling of air at the same humidity ratio , i . e. without any change in its moisture content per kg of dry air is called sensible heating or sensible cooling process respectively . if air is passed over a heating coil , there will be sensible heating . coils may be electric resistance heating coil or steam passed through the coils , or hot water passed through the coils. Similarly for sensible cooling , the air is passed over the cooling coils . coils may have refrigerant at low temperature evaporating in them , or cool water or cool gas flowing through them .

Air condition :

It is often a very critical decision for a design engineer to select a correct air conditioning system for a given space to be conditioned . the decision maker has to be keep in mind the satisfaction of the costumer and the occupant and also look to the fitness of the system for the space . the economic consideration also plays a very important role . thus system selection is the co-ordinated analysis of many factors .

Complete air conditioning gives an environment of a correct temperature, humidity , air movement, air cleanliness , ventilation and noise level . but more often than not , there is a compromise on some aspects keeping in view the primary functions and objectives . the economic consideration more often outweighs , depending upon the owner's desire and capacity to speed with intention to provide a minimal or maximum benefit . the return on investment is foremost in mind . thus a problem is defined first . the behavior of the proposed air conditioning system should be anticipated for a given external environment and internal load , the system must integrate with the space or the building which it will serve . the operation of the system to cope with instantaneous heat loads and also part loads is vital .

The 3d – 4 th weeks

Enthalpy of moist air ;

Normal practice is to express the enthalpy of mixture (1 kg of dry air + w kg of water vapour per kg of dry air

Thus $h = h_{air} + w\, h_{vapour}$

$= C_{pa} + w\, h_{vapour}$

Where h is the enthalpy of mixture / kg of dry air

h_{air} is the enthalpy of 1 kg of dry air

h_{vapour} is the enthalpy of 1 kg of vapour obtained from steam tables .

w is the specific humidity in kg / kg of dry air

C_{pa} is the specific heat of dry air normally

Psychrometric chart :

1 . Dry bulb temperature (t_d) lines are straight , parallel and vertical

2 . Humidity ratio (ω) lines are straight , parallel and horizontal

3 . Vapour pressure (p_v) lines are straight , parallel with non-uniform spacing and horizontal .

4 . Wet bulb temperature (t_w) lines are inclined , straight and not uniformly spaced .

5 . Relative humidity (ϕ) lines are curved .

6 . Specific volume (v) lines are straight and inclined .

7 . Enthalpy (h) lines are a long wet bulb temperature lines and

deviation is read from the deviation lines which are curved non- equally spaced with (+ve) and (-ve) correction .

The use of psychrometric chart is illustrated with the help of the following few solved problems . It may be noted that the chart is plotted at standard atmospheric pressure of 760 mm of Hg . Thus those properties which are functions of pressure can only be found for a total pressure of 760 mm Hg . At other pressure , corrections are necessary .

Problem ;

Atmospheric air at 760 mm of Hg barometric pressure has 25 ^oC dry bulb temperature and 15 ^oC wet bulb temperature . with the help of psychrometric chart , determine : (1) relative humidity (2) humidity ratio (3) dew point temperature (4) enthalpy of air per kg of dry air (5) partial pressure of vapour .

Solution ;

Constant dry bulb temperature td = 25 oc and constant wet bulb temperature line t_w = 15 ^oC

The properties recorded from the psychrometric chart at the point are the following :

1 – relative humidity (ϕ) = 33.8 % this line passes through the point

2 – draw a line from the point horizontally to the right represent--ing constant humidity ratio line , cutting the humidity ratio scale at w = 0.0066 kg / kg of dry air

3 –draw a line from the point horizontally to the left to cut 100% relative humidity line . this point gives dew point . from the chart t_{Dp} = 7.8 ^oC

4 –constant enthalpy and constant wet bulb lines are the same

Produced the constant enthalpy line through the point to cut enthalpy scale at h_1 = 10.06 kcal / kg of dry air

5 – partial pressure of vapour is got by drawing a horizontal line from the point to the left to cut the vapour pressure scale at P_v 7.9 mm of Hg in the chart

The 5th – 6th weeks

Mixing of air stream :

let the properties of one air stream be m_1 , w_1 and h_1
where ;

w_1 is specific humidity / kg of dry air
h_1 is enthalpy moist air / kg of dry air

Similarly , let the second stream of air have the properties m_2 , w_2 and h_2 .

After mixing , they form a third stream of air and let the properties of this stream be m_3 , w_3 and h_3 .

It is assumed that there is no loss of heat or moisture during the mixing flow process . thus , the following equations can be written for mass balance and energy balance .

For mass of air $m_1 + m_2 = m_3$

For moisture content or specific humidity

$$m_1w_1 + m_2w_2 = m_3w_3$$

For enthalpies $m_1h_1 + m_2h_2 = m_3h_3$

Thus from these three equations

$$m_1(w_1 – w_3) = m_2 (w_3 – w_2)$$

And $m_1 (h_1 – h_3) = m_2 (h_3 – h_2)$

Therefore $m_1/m_2 = (w_3 – w_2) / (w_1-w_3) = (h_3 – h_2) / (h_1 – h_3)$

The equation $w_3 – w_2 / w_1 – w_3 = h_3 – h_2 / h_1 – h_3$ shows that point 3 must be on the line joining point 1 to point 2 and this line 1-2 is divided at point 3 in the ratio of m_1 / m_2 .

Thus on the psychometric chart , the final condition can be found graphically dividing the line joining the two conditions inversely in the ratio of the mixing masses .

Sensible heating process , sensible cooling process :

The process can be represented on the chart by horizontal straight line . 1-2 represents heating process where dry bulb temperature gets raised from td1 to td2 while humidity ratio or specific humidity $\omega_1 = \omega_2$. it may be noted that during sensible heating process , relative humidity decreases $\phi_2 < \phi_1$. Sensible heat added is given by the intercept ($h_2 - h_1$) .

During sensible cooling process which is represented on the chart by line 21 , the dry bulb temperature gets lowered from t_{d2} to td1 at $\omega_2 = \omega_1$. sensible heat rejected is given by ($h_2 - h_1$). In this case heat is rejected as different from the former case in which heat is added .

Humidification process – dehumidification process :

If moisture is added to air but its dry bulb temperature maintained constant , the process is called humidification process . Conversely , if moisture is removed from air without changing the dry bulb temperature , the process is called dehumidification process .

During humidification process relative humidity increases i.e. $\phi_2 > \phi_1$ and converse is true for dehumidification process .

The change in enthalpy is shown by the intercept ($h_2 - h_1$) on the chart . since sensible heat has remained constant because dry bulb temperature has not changed . the change in enthalpy ($h_2 - h_1$) per kg of dry air is the latent heat of vaporization of the moisture content equal to ($\omega_2 - \omega_1$) kg per kg of dry air .

Pure humidification process is not found in practice . it may be

accompanied by cooling or heating as discussed later . if pure humidification is desired , the spray water through which air passes has to be maintained at dry bulb temperature td1 of the entering air .

Let air enter at point 1 at td1 and $\omega_1$and leave at td1 and ω_2 . the heat gained by air approximately the heat of vaporization of quantity of water $(\omega_2 - \omega_1)$, actually the enthalpy at point 1 ,

$$h_1 = h_{a1} + \omega_1 h_{v1}$$

Similarly enthalpy at point 2 ,

$$h_2 = h_{a2} + \omega_2 h_{v2}$$

but $h_{a1} = h_{a2}$

therefore $(h_2 - h_1) = \omega_2 h_{v2} - \omega_1 h_{v1}$

In order to simplify the equation the valid approximation that the enthalpy of vapour at any dry bulb temperature is equal to the enthalpy of saturated vapour at same dry bulb temperature is used .

The 7th -8th weeks

Cooling and dehumidification

When air passes over a cooling coil whose effective surface temperature te is below the dew point temperature of entering air t_{dp1} , condensation of the moisture takes place .
this separation of the moisture results in fall in the specific humidity or humidity ratio . thus both dehumidification and cooling can be obtained with a cooling coil effective surface temperature lower than the dew point temperature of the entering air . this effective surface temperature lower than the dew point temperature of the entering air , this effective surface temperature is called (The Apparatus Dew Point) , abbreviated as ADP . ideally the air at condition 1 while passing through the coil will reach the condition 2 i.e. tdp1 . At this pointcondensation starts and the air gets further cooled till point 3 , i.e. ADP of the cooling coil is reached . thus ideally the condition of leaving air is at 3 . During the process dry bulb temperature has fallen from t_{d2} to t_{d3} = ADP and the sensible heat removed is ($h_A - h_3$) .
Also , simultaneously the latent heat removed due to condensation of vapour of ($\omega_1 - \omega_2$) from the entering air is ($h_1 - h_2$)

However , in actual process all the air does not reach condition 3 it may be assumed that some air gets by – pass without getting effected by the coil and the remaining gets ideally cooled and dehumidified to condition 3 , thus net effect on the leaving air from the cooling coil may be explained as mixing process of some quantity of air at condition 1 with remaining air at condition 3. Thus the resulting condition will lie on the straight line joining point 1 to point 3 on the psychrometric chart by point 2 .

Heating and humidification
if the humidifier or the spray water through which air is washed is at a temperature higher than the dry bulb temperature of the entering air . the unsaturated air reaches the condition of saturat-

ion and the heat of vaporization of water is absorbed from the spray water itself . thus the air gets humidified and heated , and the spray water gets cooled . the leaving air can be heated to the spray water temperature and humidified up to saturation conditions corresponding to spray water temperature . the water may get cooled to a minimum of the wet bulb temperature of the entering air . during this process , the humidity ratio , the dry bulb temperature , the wet bulb temperature , the dew point temperature and the enthalpy of air increase while passing through hot spray water . the relative humidity may increase or decrease. The spray water is to be heated before being pumped to the spray nozzles . air enters at condition 1 ,i.e. td1 and ω1 and leaves as condition 2 , i.e. t_{d2} at ω_2 .
thus for mass balance of water

$$(\omega_{\omega 2} - \omega_{\omega 2}) = m\,(\omega_2 - \omega_1)$$

Where $\omega_{\omega 1}$: entering spray water in kg/min
$\omega_{\omega 2}$: leaving = = = =
m : mass of dry air entering in kg/min
ω_1 : humidity ratio of entering air
ω_2 : = = = leaving =

The 9th – 10th weeks

Sensible heat factor

The thermal properties of air can be separated into latent and sensible heat . the term sensible heat factor is the ratio of the sensible to the total heat , where total heat is the sum of the sensible and the latent heat . this ratio may be expressed as

$$SHF = SH /(SH + LH)$$

where SHF : sensible heat factor
SH : sensible heat
LH : latent heat

By – Pass Factor

Let the temperature of the heating coil be td3 . in ideal case , on entering the heating coil , air at temperature td1 should be able to leave at only td3 . this is due to inefficiency of the heating coil , and the same is expressed at its by–pass factor denoted by bf .therefore ,

$$BF = (t_{d3} – t_{d2}) / (t_{d3} – t_{d1})$$

for cooling process from td1 to td2 with cooling coil temperatu- re equal to td3 .

Problem ;

Air at dry bulb temperature of 30°C and 60% relative humidity enters a cooling coil at the rate of 250 m^3/min . (a) determine the refrigeration in tons needed to bring the temperature of air to the coil temperature of 23°C and also the relative humidity at that condition (b) if the effective surface temperature of the coo- ling coil or ADP is 12°C and the by-pass factor is 0.1 , determine the refrigeration in tons needed and the mass of water condensed out at the cooling coil per minute . determine also the sensible heat factor for the process through the coil .

Solution :

For the skeleton psychrometric chart on which all the values are labeled . the dew point temperature corresponding to entering conditions of air at t_{d1} = 30oc and ϕ_1= 60% , is given by drawi-ng the horizontal line from point 1 to cut the saturation line and from the chart , it reads 21.5 ^oC . thus the effective temperature of the cooling coil is above the dew point temperature and hence no dehumidification will occur . it is a sensible cooling process

from t_{d1}= 30^{o}C and ϕ_1= 60% , to t_{d2} = 23^{o}C .

also from the chart , the relative humidity at leaving conditions , i.e. at point 2 is ϕ_1= 90% relative humidity

from chart h_1= 17 kcal /kg of dry air

h_2= 15.32 =

hence cooling required

$= (h_1 – h_2) = (17 – 15.32) = 1.68$ kcal/kg of dry air

specific volume of air from the chart

v_1= 0.8808 m3/kg of dry air

the mass of air flowing per minute

m = volume in m^3 flowing per minute / specific volume

= 250 /0.8808 = 284 kg /min

therefore ,

total refrigeration required /min = 284 *1.68 = 477 kcal /min

= 477/ 50 = 9.55 tons

by – pass factor is defined as bf = (td2 – t_{ADP}) / (td$_1$ – t_{ADP})

and this is also equal to $(h_2 – h_{3(ADP)}) / (h_1 – h_{3(ADP)})$

$0.1 = (t_{d2} – 12) / (30 – 12)$

$t_{d2} = (30 – 12) * 0.1 + 12 = 13.8\,^oC$

Humidifying Efficiency

In the pervious discussion and ideal case of spray washing of the air has been considered . in practice the air will not come out of the spray at 100% relative humidity . and the extent to which humidification is affected on the velocity of air , the depth of the showers etc . thus humidifying efficiency = (td1– td3)/(td1– td2)

The 11th week

Simple Vapour Compression Refrigeration System

Concept development

all vapour compression refrigeration systems are designed and built around these basic thermodynamic principles :

1. Fluids absorb heat while changing from a liquid phase to vapour phase and given up heat in changing from a vapour phase to a liquid phase .

2. the temperature at which a change of phase occurs is constant during the change , but this temperature will vary with the pre ssure . at one fixed pressure vaporization takes place only at fixed corresponding temperature . however , the temperatures of vaporization at a particular pressure are different for different fluids .

3. heat will flow from body at higher temperature to a body at lower temperature .

4. in selecting metallic parts of cooling and condensing units , are selected which have a high heat conductivity .

5. heat energy and other forms of energy are mutually convertible with directional relationship imposed by the Second Law Of Thermodynamics .

Vaporization the Fluid

an insulated space can be adequately refrigerated by only allowing liquid to evaporate in a container vented out to atmosphere . since the temperature of the liquid remains same during vaporising process , the refrigeration will continue until all liquid is vaporized , where liquid is vaporized and heat is absorbed by the container from the space to be refrigerated , is called Evaporator and is one of the main components of vapour compression refrigeration system .

<u>The 12th - 14th weeks</u>

The 15th week

P – h Charts for Refrigerants

The vapour compression plant comprises basically a compressor , condenser ,a throttle valve or refrigerant control valve ,and an evaporator and there is a steady flow of the refrigerant through all these components to get a desired effect . All through the components the main properties of refrigerant that change and pressure and enthalpy . for example , for the steady flow through condenser and evaporator , the change in enthalpy of the refrigerant gives the heat transferred . the compressor can also be treated as steady flow machine . and during adiabatic compression , the work done on the compressor is directly given by the change in enthalpy of the refrigerant during its flow through compressor ,the throttling process through the refrigerant control valve is also constant enthalpy process . thus a chart giving directly the changes in enthalpy and pressure during process is preferable for thermodynamics analysis of vapour compression refrigerating plants .

The condition of the refrigerant in any thermodynamic state can be represented as a point on the p-h chart . the chart is divided into main areas and these are separated from each other by saturated liquid line and the dry saturated vapour line . the region which is shown on the left of the saturated liquid line is called sub-cooled liquid region . in this region , the temperature of the liquid is below the saturation temperature of liquid corresponding to its pressure . the region shown to the right of dry saturated vapour line is known as the super-heated region and the refrigerant vapour temperature in this region is above saturated temperature corresponding to its pressure or the refrigerant vapour are in a state of super-heat . the region bounded by saturated liquid and saturated vapour lines represents the change in phase of the refrigerant between liquid and vapour phases .
At any point between the two lines the refrigerant is in the state of liquid-vapour mixture . the horizontal intercept between the two lines along any constant pressure line as read on the abscissa which is enthalpy scale , gives latent heat of vaporization of the refrigerant at that pressure . the nature of curves for saturated liquid and saturated vapour is dependent on how the latent

heat of vaporization with pressure . both these curves will finally meet at a point and the pressure corresponding to that point is the critical pressure at which liquid changes to super-heated vapour and the latent heat or vaporization is zero . the lines of constant dryness extending from top to bottom through the central region and approximately parallel to saturated liquid and saturated vapour lines indicate the percentage of vapour in the mixture in the increment of 5% or 10% .

the pressure is plotted along the vertical axis on the logarithmic scale and enthalpy is plotted along the horizontal axis on linear scale. Therefore , horizontal lines extending across the chart are lines of constant pressure and the vertical lines extending are lines of constant enthalpy , in the sub-cooled region the constant temperature lines are nearly vertical and parallel to the constant enthalpy lines . in the central region bounded by the saturated liquid and saturated vapour line, the temperature line is horizontal and parallel to constant pressure lines ,because the refrigerant changes its phase from liquid to vapour at constant pressure and temperature . in the superheat region the temperature lines again change direction and fall sharply to the bottom of the chart .

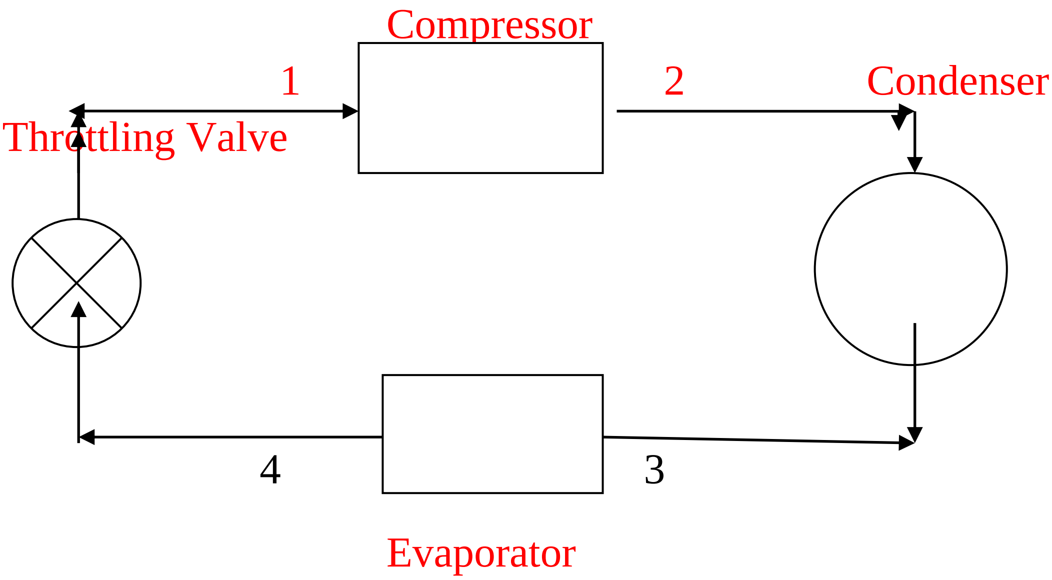
the lines , very nearly straight , extending diagonally and almost vertically across the super-heat vapour region are constant entropy lines .

the value of any of the various properties of refrigerant can be directly read off the chart. The p-h chart is plotted for 1 kg mass of refrigerant and hence all the properties are for 1 kg mass .

The 16th – 17th weeks

Simple Saturated Cycle

Vapour compression refrigeration plant is shown in figure .



The working substance is such that it readily evaporates and condenses . The cycle is thermodynamically assumed such that the refrigerant vapour leaves the evaporator and enters the compressor as dry saturated vapour . This point is denoted as point 1 both on flow diagram shown in fig. and T-S and P-h diagrams for 1kg of the working substance .

Let us take the case when the vapour is dry saturated at the suction to the compressor shown by the point 1 in the figures . It is at pressure p_1 and the temperature T_1 .

The vapour is drawn in the compressor cylinder during is suction stroke and during the compression stroke the vapour is compressed isentropically to pressure p_2 and temperature T_2 and delivered out from the compressor. this point is represented by 2 Point 2 shows the vapour in superheated state . the vapour at condition 2 passes on to condition in which cooling water is

supplied to removed heat from the vapour .

Thus vapour is first cooled the saturation temperature at pressu-

re p_2 and further removal of heat , condenses it to liquid removing its latent heat till point 3 is reached . Thus , in order to carry out this operation , the saturation , the saturation temperature corresponding to pressure p_2 should be sufficiently higher than the temperature of cooling water for efficient transfer .
It may then be possible to even sub-cool the liquid vapour to temperature below that at 3 . The high pressure liquid is now expanded through a throttle valve , and the liquid at 3 throttles to lower pressure p_1 and the condition obtained after the constat enthalpy expansion is shown at 4 . After throttling we get the liquid partly evaporated at lower temperature T_4 and lower pressure p_1 . Thus ,after the throttle valve , we get wet vapour at a low temperature . These vapours now pass through the evaporater coils immersed in brine or the chamber to be refrigerated . these vapour absorb latent heat from brine in further evaporating itself . the vapour may reach condition 1 , i.e. dry saturated at pressure p_1 . this completes the cycle . This cycle is also called *Simple Structured Cycle* .
Although the refrigerating cycle of an actual refrigerating machine will usually deviate somewhat from the simple saturated cycle , the analysis of a simple saturated cycle nonetheless is worthwhile . In such a cycle , the fundamental processes which are the basis of every actual vapour compression refrigeration cycle are easily identified and understood . Furthermore , by using simple saturated cycle as a standard against which actual cycles may be compared , the relative efficiency of the actual refrigerating conditions can be readily determined .

Analysis Of Vapour Compression Refrigeration Cycle

Simple Saturated Cycle

1.Compressor : compressor is a steady flow machine , for rever-

sible adiabatic or isentropic compression of 1kg of vapour the shaft work/kg input is given by $\omega_s = h_2 - h_1$

where ω_s is shaft work in kj/kg , h_2 and h1 are enthalpies in kj/kg at suction and delivery of the compressor respectively .

2. Condenser . steady flow process . cooling at constant pressure the heat removed is given by ${}_2q_3 = h_3 - h_2$

where ${}_2q_3$ is the heat removed in the condenser

h3 is the enthalpy of liquid/kg leaving the condenser

3. Throttling in Throttle Valve . Steady flow process throttle expansion is a constant enthalpy process . There is no work or heat transfer . on the p-h chart , it is a vertical straight line sho- wing that $h_1 = h_2$

4. Evaporator . Steady Flow Process . No work done . Heating at constant pressure . The heat absorbed/kg is given by

$${}_4q_1 = h_1 - h_4$$

where ${}_4q_1$ is the heat absorbed/kg in the evaporator .

h_4 is the enthalpy of vapour entering evaporator/kg

h_1 is the enthalpy of vapour leaving evaporator/kg

In the analysis all other losses of pressure and enthalpy for pipes etc. are neglected .

Thus C.O.P. = $(h_1 - h_4) / (h_2 - h_1)$

The 18th – 19th weeks

Actual Refrigeration Cycle :

The following are the main deviations to simple saturated cycle in actual practice .

(i) the liquid refrigerant in the condenser may be under-cooled , i.e. cooled below the condensing temperature before passing through the expansion valve . This can be obtained by large quantities of circulating cooling water which should be at a temperature much lower than the condensing temperature . the under-cooling is desirable as it increase the refrigerating effect per kg of refrigerant flow .

(ii) Normally the gas leaves the evaporator in superheated condition before it enters the compressor . this superheating may occur in the evaporator of the pipes leaving the evaporator but still in the cooled space . Thus the refrigerating effect will increase per kg of refrigerant flow . But this superheating may be occur in the pipes connecting the evaporator delivery and the compressor suction , inspite of these pipes being insulated.

Thus , this is a loss or an additional load on the compressor and condenser without any useful effect obtained . Thus these pipes should be properly insulated from heat infiltration into the system from the surroundings .

(iii) Compression has been assumed isentropic , but in actual practice it may be very complex . The cylinder walls may stabilize at a temperature between the cool suction gas and the hot discharge gas . Thus during the first part of compression heat may flow from the walls to the gas and in the later part of the stroke the reversal in the direction of flow of heat may take place . Again , there may be wire drawing taking place at the suction valve as the flow of gas has to take place through the valve with some finite pressure difference across the valve .

Same is true at the discharge valve . Thus the compressor range of pressure is slightly different from the evaporator and condenser pressures .

Since it is very difficult to evaluate these complex processes , it is a usual practice to evaluate the compression index connecting the properties at initial point of compression . This is however an approximate method .

(iv) The refrigerant has to flow through pipes in the evaporator , condenser and other connecting pipes . Thus in actual practice , due to frictional resistance to flow , there is a pressure fall in the condenser and so is the case in the evaporator . There is also a pressure fall in the connecting pipes .

It is very difficult to evaluate the deviation in the cycle in actual practice . Some of the important aspects mentioned above , have been further discussed below .

The Effect Of Sub-Cooling The Liquid

In order to appreciate the effect of sub-cooling better , numerical values have been labeled on p-h chart for both sub-cooled and simple saturated cycles .

(i) Refrigerating effect . sub-cooling increases the refrigerating effect per kg of refrigerant circulated .

(ii) Effect on flow rate /ton . Since the refrigerating effect per kg of refrigerant circulated per minute is greater in the sub-cooled cycle , the rate of flow of the refrigerant per minute per ton of refrigeration will be less as compared to that for the simple saturated cycle .

(iii) Effect on compressor volume capacity . Since the exit from the evaporator or suction to the compressor is unaltered , the specific volume of vapour sucked in by the compressor is the same in simple saturated cycle and sub-cooled cycle . But the flow rate per ton of refrigeration is reduced for sub-cooled cycle resulting in the reduction of the volume of vapour handled by the compressor per ton refrigeration .

(iv) Effect of C.O.P. . The work of compression per kg of the refrigerant flow is same for both the simple saturated as well as sub-cooled cycles, but the refrigerating effect per kg is increased in case of sub-cooled cycle resulting in the increasing of C.O.P. for the sub-cooled cycle .

The Effect Of Super-Heating The Suction Vapour

In the simple saturated cycle the exit from the evaporator or suction to the compressor is dry saturated at the evaporator vaporising pressure and temperature . In practice , this is very rare and the vaporizing refrigerant continues to absorb heat and thereby becomes superheated before it reaches compressor suction .

(i) Effect of work of compression . The work of compression in the case of superheated cycle is more than that for simple saturated cycles .

(ii) Effect on condenser performance . For the same condensing pressure and saturation temperature , the temperature of the vapours leaving the compressor and entering the condenser is very great as compared to that for simple saturated cycle .

It is very important to note that the additional heat rejected per kg in the superheat cycle is the sensible heat . The latent heat of condensation rejected per kg of superheat cycle as well as simple saturated cycle is the same . Thus greater portion of the condenser will be used for removing sensible heat in the superheat cycle . And the volume for super-heated vapours is directly proportional to the temperature following Charles law , since the condenser pressure is same throughout . Thus large volumes of gas are to be handled . But the effect of superheat cycles on C.O.P. and the system capacity can be fully appreciated if the superheating is in the useful range contributing to refrigerating effect or in the exposed pipes absorbing heat from atmosphere and not contributing to the refrigeration effect sought for .

The 20th – 22th weeks

Refrigerant Compressors :

The majority of small and medium size vapour compression refrigerating systems employ positive displacement reciprocating compressors which are designed for use with refrigerants having low specific volumes and high pressure characteristics . Ammonia , Ferion-12 (R-12) , Ferion-22 (R-22) are the only refrigerants very much used in these machines .

Performance Of Refrigerant Reciprocating Compressors

The performance of compressor and refrigerating cycle efficiency are dependent on the operating parameters . The important parameters being evaporator temperature and the condenser tem.

(i) Suction Temperature Versus Capacity

The significant variations in he compressor capacity with varing suction temperature result from the density variations of suction vapour entering the compressor . The higher the evaporator temperature , the higher is the density of suction vapour . Thus for a given piston displacement the weight of refrigerant circulated per unit time increases with rise in suction temperature .

(ii) Suction Temperature Versus Compressor Horse Power

The raising of suction temperature , with condensing temperature remaining same , leads to decrease in the horse power required per ton of refrigeration. But the total horse power of the compresser is dependent on (1) the work of compression which reduces with raising the suction temperature , (2) mass of refrigerant circulated per minute which increase with raising the suction temperature .

(iii) Condensing Temperature Versus Compressor Capacity

In general , increase of condenser temperature results in decrease of compressor refrigerating capacity .

Rotary Compressors :

Tow types of rotary compressors are popularly . One types namely the blade types .

Construction

1. Centre of rotation of the prime mover shaft .
2. Shaft . This is an eccentrically mounted shaft on the prime mover shaft and further fastened to a cylindrical steel roller .
3. Steel roller . It also rotates eccentrically with respect to prime mover shaft , being mounted on eccentric shaft . This shaft is eccentric with the cylinder and touches the cylinder having line contact at minimum clearance configuration steel roller rotes in the direction of the eccentric shaft .
4. Cylinder . The design of the cylinder and the movement of the steel roller cause the gases to be compressed .
5. Blade . It is mounted in the slot and is always in contact with the steel roller by means of a spring . Due to eccentric movment of the roller , the blade moves in and out in the slot but maintaining firm contact with the roller .
6. Spring . It is responsible for maintaining the contact of blade with roller .
7. Suction port . It is located in the cylinder near the blade slot .
8. Discharge port . It is also located in the cylinder near the blade slot , but on the side opposite to the suction port .
9. Discharge valve .
10. Oil bath .The cylinder is enclosed in a housing filled with oil and remains submerged in oil .
11. Discharge line . High pressure vapour gets collected above the oil level and goes to discharge line and then to condenser .

When the compressor is working an oil film forms the seal between the high pressure and the low pressure side . But this seal is lost when compressor stops and pressure equalizes in whole of the cylinder chamber . Check valves are provided in the suction and the discharge lines to avoid back flow of vapour .

Centrifugal Compressor

Usually the centrifugal compressors have low radial velocity component which are so small that they can be neglected and the blades are radial tipped . Also , the velocity of flow at the in-let to the impeller and the outlet from the diffuser is practically same .

Effect of variables depending upon the physical properties of working substance and compressor dimension can be predicated.

(i) As density of fluid increases , the pressure ratio will increa-se because of the increase in the value within the parenthesis .

(ii) As the adiabatic index increases , expression in parenthesis has a higher value but the exponent decreases more rapidly .This results in decrease of pressure ratio with increase of the index .

(iii) As the impeller speed , increases or the impeller diameter increases , blade tip velocity increases and hence the expression in the parenthesis increases and so does the pressure ratio .

The 23th – 24th weeks

Condensers

The compressor , condenser and liquid receiver are classified as high – pressure side equipment , since all are subject to the highest pressure in the system .

The functions of condenser in a refrigeration system are to de – superheat and condense the compressor discharge vapour and frequently to sub – cool the resultant liquid while introducing a minimum pressure drop . It is a heat rejection component in the refrigeration cycle . Condensers are of three general types .

(1) Water – cooled condensers .

(2) Air – cooled condensers .

(3) Evaporate condensers .

Water – cooled condensers : These are used for all sizes and types of refrigeration machines . When abundant clean water is cheaply available , these condensers are the first choice from an economic stand point . When water is costly , it is usually cooled in cooling towers and fed again to the condensers . Water cooled condensers require less surface than air cooled or evaporative condensers , because of high water film coefficient .

(i) Shell and Tube Condenser . In order to obtain high water film heat transfer , the cooling water always flows through the tubes and the refrigerant condensers on the outside of the tubes . The tubes are circulated to give the desired water velocity and pressure drop . Copper tubes are used for Freon group and steel tubes for Ammonia . Since , even small quantities of air or non – condensables affect the performance adversely , a purge connection is provided for air removal .

The vertical shell- and- tube condenser is essentially the same as the horizontal condenser except for the position in which it is installed resulting in a small space required .

(ii) A shell and coil condenser . It uses a helical water coil in the shell instead of straight tubes . The connection A for taking out the condensed liquid refrigerant is so made that it dips in the liquid refrigerant at the bottom position which acts as the receiver . The refrigerant vapours from the compressor enter at higher level . The cooling water is fed at C and leaves at D after passing through the helical coil .

(iii) A double pipe condenser . It consists of concentric tubes with the refrigerant condensing in the annular space and the water flowing in the inner tubes . This is popular because it can be easily made to fit the size of the unit to be cooled . The outside tube is also cooled by air in the room providing an efficient operation . Heat transfer rates are low in this type of condenser if the tubes are long because the poor drainage of the condensing refrigerant keeps the vapour from coming in contact with water tube .

Air Cooled Condensers : The condenser is usually made of copper or steel tubing with fins attached with assist in rapid radiation of heat . For domestic use the condenser is usually air–cooled by natural convection . Air surrounding the condenser will be warmer than the air in the room . This warm air will rise and cooler air will flow into take its piece . Some condensing units use motor driven fan to force air over the condenser tubing and to increase the cooling effect on the condenser . Two typical shapes of air-cooled condensers namely flat coil finned type and spiral coil finned type . The capacity of an air-cooled condenser may be calculated using one of the two basic methods .

(1) Using the total external area of the condenser to compute its heat dissipating capacity .

(2) Computations based upon what is called the frontal area of the condenser .

Evaporative Condensers : The term evaporative condenser is defined as comprising a coil in which the refrigerant is flowing and condensing inside and its outer surface is wetted with water and exposed to stream of air to which heat is rejected principally by evaporation of water .

The coils are generally made of copper or steel in multiple circuits and passes . The external surfaces are sometimes finned to increase heat transfer surface . The coil should have arrangements for cleaning under fouling water condition .

The wetting of coil is done by re-circulating system comprising water pan , a pump and water distribution system . The water distribution system mainly comprises nozzles for spray of atomised water on the coils . The pan catches the drainage of all coil. There is a float valve to admit make up water and maintain the correct level in the pan . Centrifugal pumps of moderate head are necessary . Such pumps are not affected much by extraneous matter found in such re-circulating systems .

Most evaporative condensers employ forced circulation of air with a fan to either blow or draw air through the unit . Effective elimination of moisture from the leaving air system by eliminators is essential to prevent projection of mist which can deposit moisture on the surrounding surfaces . The eliminator plates work on the simple principle of abrupt changes in flow direction . Moisture particles being heavier get deposited on these eliminator plates and get drained back to the sump or the pan .

The 25th- 26th weeks

Expansion Devices : the expansion device also forms a very important component of the vapour compression refrigerant system Compressors and condensers have already been discussed . After the condenser comes the expansion device . Common expansion device are :

(1) Capillary tube .
(2) Thermostatic expansion valve .
(3) Low and high side float valves .
(4) Constant pressure expansion valve .

Capillary Tube :

Almost all fractional horsepower , vapour compression refrigeration units employ capillary tube . The use is being extended presently to larger units up to about 5 h.p. The capillary tube when used as a liquid refrigerant expanded device usually consists of an extremely small bore tube from 0.5 mm to 2.5 mm of about 0.5 m to 5 m long . Numerous combinations of bore and length are available to get the desired restriction . Its extreme simplicity and very low cost make it very popular . In its operation , liquid refrigerant enters the capillary tube and due to flow , there is pressure drop due to friction . Some of the liquid flashes into vapour as the refrigerant flows through the tube . Once the sizing and length of tube is selected , no modifications are possible to adjust itself to variation in discharge pressure , suction pressure and load . Caremust be taken to prevent plugging of the tube by any dirt , ice or any other decomposed material . The capillary tube is substituted for the conventional liquid line from the condenser and soldered to a length of the suction line to from a simple heat exchanger .

The capillary , as the control device , does not close off and stop the refrigerant flow during the shut down period . The high side pressure and the low side pressure tend to equalize through the capillary . Thus the residual liquid the condenser passes on to evaporator till again the plant starts . This is also an important criteria for the charge in the capillary tube system to be critical and the receiver is also not used between the condenser and the capillary . The refrigerant charge should be only enough to satisfy the evaporator requirements and maintain a liquid seal in the condenser at the entrance to the capillary . If the refrigerant is in excess , it will only back up in the condenser reducing the effective surface area of condensation eventually lead to rise of condenser pressure . This can increase the flow rate through the tube. More important thing to note is that all the excess liquid in the condenser will pass to the evaporator during shut down period . This will warm up the evaporator also quickly and result in defrosting of the evaporator and short cycling of the compressor . Also , flooding back to the compressor is likely , when cycle starts again . The equalizing of pressure in case of capillary tube may be treated as an advantage is as much as the starting torque of the compressor motor may practically be one of the unloaded plant , and thus greatly reduced .

Thermostatic Expansion Valve : This is the most popular and very efficient type of expansion device in use at present . The operation of thermostatic expansion valve is based on the principle of constant degree of super-heat for the evaporator exit. This ensures the evaporator completely filled with refrigerant irrespective of the load and also no liquid can spill over to the suction line to the compressor . Because of its adaptability to load changes , it is specially suitable for variable load systems. The remote bulb charged with fluid which is open on one side of the diaphragm through a capillary tube is clamped firmly to the evaporater outlet . the temperature of the saturated liquid vapour mixture is the same as the temperature of the superheated gas leaving

The evaporator at this location . The pressure of the fluid in the bulb tends to open the valve . This pressure is balanced by pressure due to spring plus pressure in the evaporator . There is thus, inter-action of three independent forces namely force due to spring compression and the force due to saturated liquid – vapour in the bulb .

Float Valves :

Low pressure float valve . The low pressure float control maintains the liquid at constant level in the evaporator by regulating the flow into the evaporator in accordance with the supply from the evaporator to compressor or the rate of vaporization in the evaporator . If the refrigeration load increases , the evaporator temperature and pressure rises , which temporarily allows the compressor to pump a greater mass rate of flow than the valve is feeding . The valve reacts to keep the level constant by opening more. If the refrigeration load decreases, the evaporator pressure falls and the compressor now pumps less mass rate temporarily and the level in float chamber rises resulting in the tendency to close the valve .

High pressure float valve . the high pressure float valve also maintains the flow to the evaporator by actuating the level in the float chamber in the same manner as the low pressure float valve , except that the high pressure float valve is located on the high pressure side and controls the amount of liquid by maintaining level in the float chamber . The condensation rate and the evaporation rate are matched by the high pressure float valve by actuating the level and thus altering the opening and closing of the needle valve . To have further control on the expansion of liquid refrigeration, a pressure reducing valve is also used in the circuit

The 27th – 28th weeks

Three major vapour compression system components namely compressor , condenser ,and expansion device have already bee-n discussed . The fourth component to complete the cycle is the evaporator . The evaporator in the vapour compression cycle , is a heat exchanger which absorbs heat from the substance to be cooled and transfers is to a boiling refrigerant .

Types of Evaporators

The evaporators may be classified as Forced convection type or Free convection type depending upon whether the substance to be cooled is forced by pump or fan through the heat transfer surfaces of the evaporator , or it flows naturally by density difference of warmer and colder fluid . Some evaporators have refrigerant in the tubes and substance to be cooled surrounding the tube, but , in order cases , the refrigerant is in the shell with substance to be cooled passing through the tubes .

Evaporators are also classified as Flooded type and Dry type depending upon whether liquid refrigerant covers all heat transfer surface or some portion is having gas being superheated . The evaporators with thermostatic expansion valve will have some portion of heat transfer surface where superheating is taking place and can be designed as dry evaporator ; whereas evaporators with float valve will be flooded type .

Flooded Evaporator :

A typical flooded evaporator with float control . The liquid on its flow passage upwards through the tubes , boils due to absorption of heat from the warmer substance which is to be cooled . The vapour so formed on boiling bubbles up in flash chamber. The flash chamber separates vapour from liquid which flows back to the evaporator whereas vapours are sucked by the compressor .

The flash chamber collects the flash or vapour obtained in the expansion device plus the vapour formed by refrigerant liquid boiling in the evaporator .

Liquid Chiller :

The former has refrigerant in the shell and liquid to be chilled in the tubes whereas the latter has refrigerant in the tubes and liqui-d to be chilled in the shell . When the refrigerant is in the shell , the refrigerant liquid level is so kept that there is enough space on the top portion of the shell for the liquid and vapour to separ-ate . Vapours are drawn from the top portion by the compressor . Liquid level must be maintained constant as the chilled tubes are also immersed in the refrigerant liquid . Thus float control is pr-ferred . But when the liquid to be chilled is in the shell and the refrigerant , thermostatic expansion valve is preferred . The refr-igerant gets superheated in the last portion of the set of tubes an-d is collected in the end chamber from where it is sucked by the compressor . In order to facilitate proper contact of water with the refrigerant tubes , baffles are provided to ensure larger circu-it up and down for the water , resulting in increased turbulence and hence better over-all transfer co-efficient .

Direct Expansion Coil :

In the liquid chiller , the chilled liquid is fed to the coils which are used for cooling air . But , if the coils of the evaporator with refrigerant passing through them are used directly to cool air by natural or forced convection , the coil is called direct expansion coil . The refrigerant feed comes through the thermostatic expa-nsion valve more often located at the top particularly for Freon-12 and Freon – 22 to improve the lubricating oil return to the co-mpressor . Air is blown over the outside of the finned tubes . For air conditioning purposes , the direct expansion coil is preferred where the evaporator is very near to the compressor . It is direct method of cooling the substance and , therefore , quite efficient . But when the coil has to be located very far away from the compressor , it is preferred to chill the water and pump it to the air cooling coil . For long distances , there is possibility of refrigerant leakage and the cost of the refrigerant would be also high . Besides , the pressure drop in the line would impair evaporator efficiency and co-efficient of performance . The pressure effect becomes very significant if the direct expansion coil is located at a great height from the compressor and condenser unit .