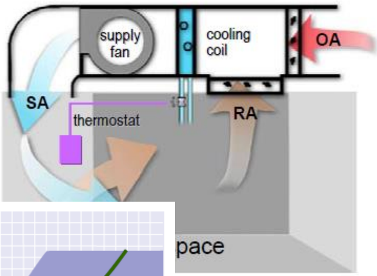
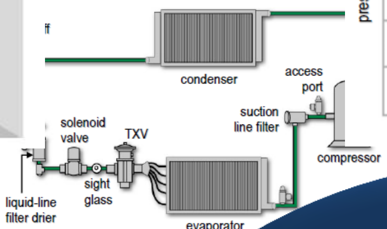
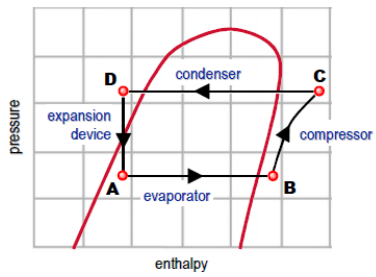
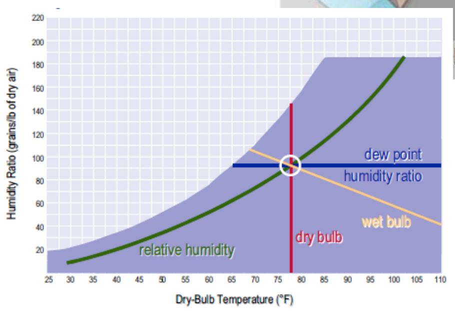
Northern Technical University
Engineering Technical College of Mosul
Power Mechanics Techniques
Engineering Department

Refrigeration and air conditioning Lectures









Second Level

By

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AIR CONDITIONING

Air conditioning:- is a combined process that performs many functions simultaneously. It conditions the air, transports it, and introduces it to the conditioned space. It provides heating and cooling from its central plant or rooftop units. It also controls and maintains the temperature, humidity, air movement, air cleanliness, sound level, and pressure differential in a space within predetermined limits for the comfort and health of the occupants of the conditioned space or for the purpose of product processing.

An air conditioning was first systematically developed by ***Dr. Willis H. Carrier***, who is recognized as the father of air conditioning. In 1902, Carrier discovered the relationship between temperature and humidity and how to control them.

The term ***HVAC&R*** is an abbreviation of **Heating, Ventilating, Air Conditioning, and Refrigerating**. The combination of processes in this commonly adopted term is equivalent to the current definition of air conditioning. Because all these individual component processes were developed prior to the more complete concept of air conditioning, the term ***HVAC&R*** is often used by the industry.

Air Conditioning Systems:-

Air conditioning systems can be classified according to their application as **(1)** Comfort air conditioning systems and **(2)** Process air conditioning systems.

(1) Comfort Air Conditioning Systems:-

Comfort air conditioning systems provide occupants with a comfortable and healthy indoor environment in which to carry out their activities. The various sectors of the economy using comfort air conditioning systems are as follows:-

1- The commercial sector includes office buildings, supermarkets, department stores, shopping centers, restaurants, and others.

2- The institutional sector includes such applications as schools, colleges, universities, libraries, museums, indoor stadiums, cinemas, theaters, and concert halls.

3- The residential and lodging sector consists of hotels, motels, apartment houses, and private homes.
4- The health care sector encompasses hospitals, nursing homes, and convalescent care facilities.
5- The transportation sector includes aircraft, automobiles, railroad cars, buses, and cruising ships.

<u>(2) Process Air Conditioning Systems:-</u>

Process air conditioning systems provide needed indoor environmental control for manufacturing, product storage, or other research and development processes. The following areas are examples of process air conditioning systems:-

1- In textile mills, natural fibers and manufactured fibers are hygroscopic.
2- Many electronic products require clean rooms for manufacturing such things as integrated circuits, since their quality is adversely affected by airborne particles.
3- Precision manufacturers always need precise temperature control during production of precision instruments, tools, and equipment.
4- Pharmaceutical products require temperature, humidity, and air cleanliness control.

SI UNITS

Any physical quantity can be characterized by **dimensions.** The magnitudes assigned to the dimensions are called **units.** Some basic dimensions such as mass *m*, length *L*, time *t*, and temperature *T* are selected as **primary** or **fundamental dimensions,** while others such as force *F*, energy *E*, pressure *P* and volume *V* are expressed in terms of the primary dimensions and are called **secondary dimensions,** or **derived dimensions**, (see Table (1)).

A number of unit systems have been developed over the years. Despite strong efforts in the scientific and engineering community to unify the world with a single unit system, two sets of units are still in common use today: the ***English system,*** which is also known as the *United States Customary System* (**USCS**), and the metric **SI** (from *Le Système International d' Unités*), which is also known as the ***International System***. The SI is a simple and logical system based on a decimal relationship between the various units, and it is being used for scientific and engineering work in most of the industrialized nations, including England.

TABLE 1:- Primary Dimensions and Units for the SI and USCS

Dimension (Quantity)	Unit and Symbol	
	SI System	**English System**
Mass (*m*)	kilogram (kg)	pound-mass (Ibm)
Length (*L*)	meter (m)	foot (ft)
Time (*t*)	second (s or sec)	second (s or sec)
Temperature (*T*)	celsius (°C) kelvin (°K)	rankine (°R)

The SI system of units is an extension of the metric system and has been adopted in many countries as the only system accepted legally. So, these lectures will be presented in SI units.

In the SI system, force is a secondary dimension. The unit of force is therefore a secondary or derived unit and is the *newton* (N), which can be obtained from Newton's second law, $F = ma$, as

$$1\ \text{N} = 1\ \text{kg}(\text{m/s}^2) = 1\ \text{kg} \cdot \text{m/s}^2$$

Because pressure is force per unit area, $P = F/A$, the unit of pressure, the *pascal* (Pa), can be expressed as

$$1\ \text{Pa} = 1\ \text{N/m}^2 = (1\ \text{kg} \cdot \text{m/s}^2)(1/\text{m}^2) = 1\ \text{kg/m} \cdot \text{s}^2$$

A work interaction, or work, is represented by $\delta W = Fx$ so the unit of work or *energy* is the *joule* (J), defined as

$$W = 1\ \text{J} = 1\ \text{N} \cdot \text{m}$$

and because *power* P is the rate of doing work, the unit of power is the *watt*:

$$P = dW/dt = W/t = 1\ \text{J/s} = 1\ \text{N} \cdot \text{m/s}$$

Observe that the *weight* of a body is equal to the force of gravity on the body. Hence, weight always refers to a force, and in the SI system this force is always in newtons. The mass of the body can always be related to its weight via ($W = mg$)

where g, the local gravitational acceleration, has a mean value at sea level of

$$g = 9.807\ \text{m/s}^2$$

and is a function of location. This shows that the weight of a body may vary, whereas the mass of the body is always the same.

Conversion factors from English engineering units to SI units are given by ***American Society of Heating, Refrigerating, and Air-conditioning Engineers*** (**ASHRAE**).

FUNDAMENTAL PROPERTIES OF AIR AND WATER VAPOUR MIXTURE

The surface of the earth is surrounded by a layer of air called the atmosphere, or atmospheric air. Air in the atmosphere normally contains some water vapour (or moisture). Before we begin the discussion of thermodynamics of moist air mixture, we shall define their most important properties and indicate where each is used.

1- Dry Bulb Temperature (***DBT***):- It is the temperature of moist-air mixture measured by a perfectly dry sensor, such as thermocouple or glass thermometer.

2- Wet Bulb Temperature (***WBT***):- It is the temperature of moist-air mixture measured by thermometer's bulb covered by a wick that has been thoroughly wetted with water.

> Partial pressure (***Ps***):- is the pressure exerted by one gas component on a mixture of several gases. The partial pressure of water vapour is used to define relative humidity.

3- Relative humidity (***ϕ***):- It is the ratio of the partial pressure of water vapour to the saturation pressure of water vapour at the existing dry bulb temperature.

4- Moisture content (***w***):- It is the ratio of the mass of water vapour to the mass of dry air in a moist-air mixture (it's also called ***humidity ratio***).

5- Dew Point Temperature (***DPT***):- It is the temperature at which a moist-air mixture at a given humidity ratio become saturated. In other words, ***DPT*** is the temperature at which moisture will start to condense from the air.

6- Specific Volume (***v***):- It is the volume in cubic meters of one kg of dry air together with the mass of water vapour associated with it.

7- Enthalpy (***h***):- It is the measurement of the heat content of the air in *kJ* per *kg* of dry air (it's also called ***heat content***).

Standard Adopted:-

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

The General Gas Law:-

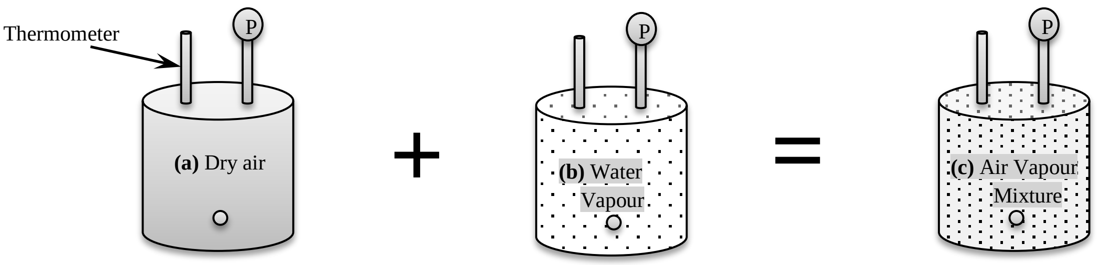
It is possible, as we saw from the first year of thermodynamics subject, to combine Boyle's and Charle's laws as one equation:-

$$PV = m.R.T$$

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

Dalton's law of partial pressure:-

If a mixture of gases occupies a given volume at a given temperature, then, the total pressure by the mixture equals the sum of the pressure of the constituents, each being considered at the same volume and temperature. Figure (1) shows the variation of mass and pressure of dry air and water vapour, at an atmospheric pressure of 101325 Pa and a temperature of 20°C.



T	20°C
m_a	1kg
m_w	0kg
P_a	100143Pa
P_s	0Pa
P_{at}	100143Pa

T	20°C
m_a	0kg
m_w	0.0073kg
P_a	0Pa
P_s	1182Pa
P_{at}	1182Pa

T	20°C
m_a	1kg
m_w	0.0073kg
P_a	100143Pa
P_s	1182Pa
P_{at}	101325Pa

FIGURE (1) Mass and pressure of dry air, water vapor, and moist air.

The principle of conservation of mass for nonnuclear processes gives the following relationship:-

$$m_m = m_a + m_w$$

where m_m = mass of moist air (kg)
m_a = mass of dry air (kg)
m_w = mass of water vapor (kg)

Applying Dalton's law for moist air, we have

$$Pat = Pa + Ps$$

where Pat = atmospheric pressure or pressure of the outdoor moist air (Pa)
Pa = partial pressure of dry air (Pa)
Ps = partial pressure of water vapor (Pa)

The vapour pressure of steam in moist-air (*Ps*):-

The question arises; how do we determine the vapour pressure for relative humidity less than 100%? An empirical equation exists which answer this question:-

$$Ps = Pssw - Pat \cdot A \cdot (DBT - WBT)$$

where:- Ps Vapour pressure
$Pssw$ Saturation vapour pressure at wet bulb temperature (*WBT*)
Pat Barometric pressure
A constant $A = 6.66 \times 10^{-4}$ °C^{-1} when WBT ≥ 0
$A = 5.94 \times 10^{-4}$ °C^{-1} when WBT < 0

Example 1:-

Calculate the vapour pressure of moist-air at 20°C *DBT*, 15°C *WBT* and 95 kPa.

Solution:-

From **saturated water and steam table** at 15°C *WBT*, $Pssw$ = 1.704kPa

$Ps = 1.704 - 95 \times 6.66 \times 10^{-4} \times (20 - 15) = 1.3876$kPa

Relative humidity (ϕ):-

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

$$\phi = \frac{Ps}{Pss}$$

Pss = saturated water vapour pressure at dry bulb temperature kPa.

Saturated Water and Steam

T [°C]	p_s [bar]	v_g [m^3/kg]	h_f [kJ/kg]	h_{fg} [kJ/kg]	h_g [kJ/kg]	s_f [kJ/kg K]	s_{fg} [kJ/kg K]	s_g [kJ/kg K]
0.01	0.006112	206.1	0*	2500.8	2500.8	0†	9.155	9.155
1	0.006566	192.6	4.2	2498.3	2502.5	0.015	9.113	9.128
2	0.007054	179.9	8.4	2495.9	2504.3	0.031	9.071	9.102
3	0.007575	168.2	12.6	2493.6	2506.2	0.046	9.030	9.076
4	0.008129	157.3	16.8	2491.3	2508.1	0.061	8.989	9.050
5	0.008719	147.1	21.0	2488.9	2509.9	0.076	8.948	9.024
6	0.009346	137.8	25.2	2486.6	2511.8	0.091	8.908	8.999
7	0.01001	129.1	29.4	2484.3	2513.7	0.106	8.868	8.974
8	0.01072	121.0	33.6	2481.9	2515.5	0.121	8.828	8.949
9	0.01147	113.4	37.8	2479.6	2517.4	0.136	8.788	8.924
10	0.01227	106.4	42.0	2477.2	2519.2	0.151	8.749	8.900
11	0.01312	99.90	46.2	2474.9	2521.1	0.166	8.710	8.876
12	0.01401	93.83	50.4	2472.5	2522.9	0.180	8.671	8.851
13	0.01497	88.17	54.6	2470.2	2524.8	0.195	8.633	8.828
14	0.01597	82.89	58.8	2467.8	2526.6	0.210	8.594	8.804
15	0.01704	77.97	62.9	2465.5	2528.4	0.224	8.556	8.780
16	0.01817	73.38	67.1	2463.1	2530.2	0.239	8.518	8.757
17	0.01936	69.09	71.3	2460.8	2532.1	0.253	8.481	8.734
18	0.02063	65.08	75.5	2458.4	2533.9	0.268	8.444	8.712
19	0.02196	61.34	79.7	2456.0	2535.7	0.282	8.407	8.689
20	0.02337	57.84	83.9	2453.7	2537.6	0.296	8.370	8.666
21	0.02486	54.56	88.0	2451.4	2539.4	0.310	8.334	8.644
22	0.02642	51.49	92.2	2449.0	2541.2	0.325	8.297	8.622
23	0.02808	48.62	96.4	2446.6	2543.0	0.339	8.261	8.600
24	0.02982	45.92	100.6	2444.2	2544.8	0.353	8.226	8.579
25	0.03166	43.40	104.8	2441.8	2546.6	0.367	8.190	8.557
26	0.03360	41.03	108.9	2439.5	2548.4	0.381	8.155	8.536
27	0.03564	38.81	113.1	2437.2	2550.3	0.395	8.120	8.515
28	0.03778	36.73	117.3	2434.8	2552.1	0.409	8.085	8.494
29	0.04004	34.77	121.5	2432.4	2553.9	0.423	8.050	8.473
30	0.04242	32.93	125.7	2430.0	2555.7	0.436	8.016	8.452
32	0.04754	29.57	134.0	2425.3	2559.3	0.464	7.948	8.412
34	0.05318	26.60	142.4	2420.5	2562.9	0.491	7.881	8.372
36	0.05940	23.97	150.7	2415.8	2566.5	0.518	7.814	8.332
38	0.06624	21.63	159.1	2411.0	2570.1	0.545	7.749	8.294
40	0.07375	19.55	167.5	2406.2	2573.7	0.572	7.684	8.256
42	0.08198	17.69	175.8	2401.4	2577.2	0.599	7.620	8.219
44	0.09100	16.03	184.2	2396.6	2580.8	0.625	7.557	8.182
46	0.1009	14.56	192.5	2391.8	2584.3	0.651	7.494	8.145
48	0.1116	13.23	200.9	2387.0	2587.9	0.678	7.433	8.111
50	0.1233	12.04	209.3	2382.1	2591.4	0.704	7.371	8.075
55	0.1574	9.578	230.2	2370.1	2600.3	0.768	7.223	7.991
60	0.1992	7.678	251.1	2357.9	2609.0	0.831	7.078	7.909
65	0.2501	6.201	272.0	2345.7	2617.7	0.893	6.937	7.830
70	0.3116	5.045	293.0	2333.3	2626.3	0.955	6.800	7.755
75	0.3855	4.133	313.9	2320.8	2634.7	1.015	6.666	7.681
80	0.4736	3.408	334.9	2308.3	2643.2	1.075	6.536	7.611
85	0.5780	2.828	355.9	2295.6	2651.5	1.134	6.410	7.544
90	0.7011	2.361	376.9	2282.8	2659.7	1.192	6.286	7.478
95	0.8453	1.982	398.0	2269.8	2667.8	1.250	6.166	7.416
100	1.01325	1.673	419.1	2256.7	2675.8	1.307	6.048	7.355

† u and s are chosen to be zero for saturated liquid at the triple point.

Note: values of v_f can be found on p. 10.

Example 2:-

Calculate the relative humidity of moist-air at 30°C *DBT*, 20°C *WBT* and 100 kPa barometric pressure?

Solution:-

$Ps = Pssw - Pat \, . \, A \, . \, (DBT - WBT)$

($Pssw$ = 2.337 kPa at 20°C *WBT*) From saturated water and steam table.

$Ps = 2.337 - 100 \times 6.66 \times 10^{-4} \, (30 - 20) = 1.671 \text{kPa}$

Pss = 4.242 kPa at 30°C *DBT*

$$\phi = \frac{Ps}{Pss} \times 100\% = \frac{1.671}{4.242} \times 100\% = 39.39\%$$

Moisture-content (*w*):-

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

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

where m_s mass of water vapour (kg)

m_a mass of dry air (kg)

Since dry air and water vapor can occupy the same volume at the same temperature, we can apply the ideal gas equation and Dalton's law for dry air and water vapor. ($PV = mRT$).

For air $P_aV = m_aR_aT \;\rightarrow\; m_a = \frac{P_aV}{R_aT}$

For water $P_sV = m_sR_sT \;\rightarrow\; m_s = \frac{P_sV}{R_sT}$

$$w = \frac{m_s}{m_a} = \frac{P_sV}{R_sT} \cdot \frac{R_aT}{P_aV} = \frac{P_s}{P_a} \cdot \frac{R_a}{R_s}$$

$$R_a = \frac{R_o}{M_a} \rightarrow R_s = \frac{R_o}{M_s} \qquad M_a = 28.97 \qquad \& \qquad M_s = 18$$

$$\therefore w = \frac{P_s}{P_a} \cdot \frac{R_a}{R_s} = \frac{P_s}{P_a} \cdot \frac{M_s}{M_a} = \frac{18}{28.97} \frac{P_s}{P_a}$$

$$w = 0.622 \frac{P_s}{P_a} \qquad OR \qquad w = 0.622 \frac{P_s}{P_{at} - P_s}$$

Another form of calculation by using Relative Humidity is:-

$$w = 0.622 \frac{P_s \cdot \frac{P_{ss}}{P_{ss}}}{P_{at} - P_s \cdot \frac{P_{ss}}{P_{ss}}} = 0.622 \frac{P_{ss} \cdot \frac{P_s}{P_{ss}}}{P_{at} - P_{ss} \cdot \frac{P_s}{P_{ss}}}$$

$$w = 0.622 \frac{\phi . P_{ss}}{P_{at} - \phi . P_{ss}}$$

Example 3:-

Calculate the moisture content of moist air at 20°C *DBT*, 15°C *WBT* and 95 kPa.

Solution:-

$Ps = 1.704 - 95 \times 6.66 \times 10^{-4} (20 - 15) = 1.387$kPa

$$w = 0.622 \frac{1.387}{95 - 1.387} = 9.215 \times 10^{-3}\ kg_w/kg_a$$

Example 4:-

Calculate the moisture content and relative humidity of saturated air at 20°C and 95kPa.

Solution:-

Since the air is saturated, therefore *WBT* = *DBT* = 20°C

$Ps = Pssw = 2.337$kPa, $\phi = 100\%$

$$w = 0.622 \frac{2.337}{95 - 2.337} = 0{,}0157\ kg_w/kg_a$$

Dew Point Temperature (*DPT*):-

If unsaturated moist air is cooled at constant pressure, then the temperature at which the moisture in the air begins to condense is known as Dew Point Temperature (*DPT*) of air. An approximate equation for dew-point temperature is given by:-

$$DPT = \frac{4030(DBT + 235)}{4030 - (DBT + 235) ln\, \phi} - 235$$

Example 5:-

Calculate the dew point temperature of moist air at 20°C *DBT*, 15°C *WBT* and 95 kPa.

Solution:-

$Ps = 1.704 - 95 \times 6.66 \times 10^{-4} (20 - 15) = 1.387$kPa

$Pss = 2.337$ kPa at 20°C *DBT* $\rightarrow \quad \phi = \frac{Ps}{Pss} = \frac{1.387}{2.337} = 0.593$

$$DPT = \frac{4030(DBT+235)}{4030-(DBT+235)ln\,\phi} - 235 = \frac{4030(20+235)}{4030-(20+235)ln\,0.593} - 235 = 11.838°C$$

Another solution:-

At Ps = 1.387 the air is at dew point temperature, therefore the air is saturated and the saturation pressure is 1.387 kPa, then the dew point temperature is:-

By interpolation:-

T_1 =11	P_1 = 1.312
T = DPT	Ps = 1.387
T_2 = 12	P_2 = 1.401

$$\frac{T_2 - T_1}{T_2 - DPT} = \frac{P_2 - P_1}{P_2 - P_s} \quad \rightarrow \quad DPT = 12 - \frac{1.401 - 1.387}{1.401 - 1.312} = 11.84°C$$

Specific Volume (v):-

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

Example 6:-

Calculated the specific volume of moist air at 45°C *DBT*, 25 °C *WBT*, and Barometric pressure is 101.25kPa.

Solution:-

Ps = 3.166 – 101.25 × 6.66 × 10^{-4} (45 – 25) = 1. 81735kPa

$Pa = Pat – Ps$ = 101.25 – 1. 81735 = 99.4365kPa

For one kg of moist air:- $w = 0.622\,\frac{1.81735}{99.4365} = 0.01137\,kgs/kga$

m_a = 1 – 0.01137 = 0.9886 kga

$$V_a = \frac{m_a R_a T}{P_a} = \frac{0.9886 \times 0.287 \times (45+273)}{99.4365} = 0.9074m^3$$

$$V_s = \frac{m_s R_s T}{P_s} = \frac{0.01137 \times 0.461 \times (45+273)}{1.81735} = 0.917m^3 \approx V_a$$

$$v = \frac{V}{m_a} = \frac{0.9074}{0.9886} = 0.918m^3/kg$$

Moist Air Enthalpy (*h*):-

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

$$\boldsymbol{h = h_a + w.h_w}$$

where:- h_a enthalpy of dry air kJ/kg

h_w enthalpy of water vapour kJ/kg

w moisture content kgs/kga

Enthalpy values are always based on some reference value. For moist air, the enthalpy of dry air is given a zero value at 0°C, and for water vapour the enthalpy of saturated water is taken as zero at 0°C.

The enthalpy of moist air is given by:-

$$\boldsymbol{h = c_{pa}\,.DBT + w(\,h_{fg} + c_{pv}\,.DBT\,)}$$

where:- h_{fg} enthalpy of water vapour at reference temperature =2501.3 kJ/kg.K

c_{pa} specific heat of air at constant pressure = 1.005 kJ/kg.K

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

$$\boldsymbol{h = 1.005DBT + w(\,2501.3 + 1.86DBT\,)}$$

Example 7:-

Calculate the approximate enthalpy of moist air at 35°C *DBT*, 25°C *WBT* and 101.325 kPa.

Solution:-

P_{ssw}= 3.166 kPa at 25°C WBT

$Ps = 3.166 - 101.325 \times 6.66 \times 10^{-4}\,(35 - 25) = 2.491$ kPa

$$w = 0.622\,\frac{2.491}{101.325-2.492} = 0.0156\,kgs/kga$$

$h = 1.005 \times 35 + 0.0156(\,2501.3 + 1.86 \times 35\,) = 75.21\ kJ/kg$

PSYCHROMETRIC CHART

Psychrometric chart provides a graphical representation of the thermodynamic properties of moist air, various air conditioning processes, and air conditioning cycles. The chart is very helpful during the calculation, analysis, and solution of the complicated problems encountered in air conditioning processes and cycles.

Construction of psychrometric chart:-

Standard psychrometric chart is bounded by the dry-bulb temperature line (abscissa) and the vapour pressure or moisture content (ordinate). The Left Hand Side of the psychrometric chart is bounded by the saturation line. The wet bulb temperatures appear as diagonal lines, coinciding with the dry bulb at the saturation line. If measurements are taken with the two thermometers of the sling psychrometer, the condition can be plotted on the psychrometric chart by taking the intersection of the dry bulb temperature, as read on the vertical line, with the wet bulb temperature, read down the diagonal wet bulb line. Figure (2) shows the schematic of a psychrometric chart.

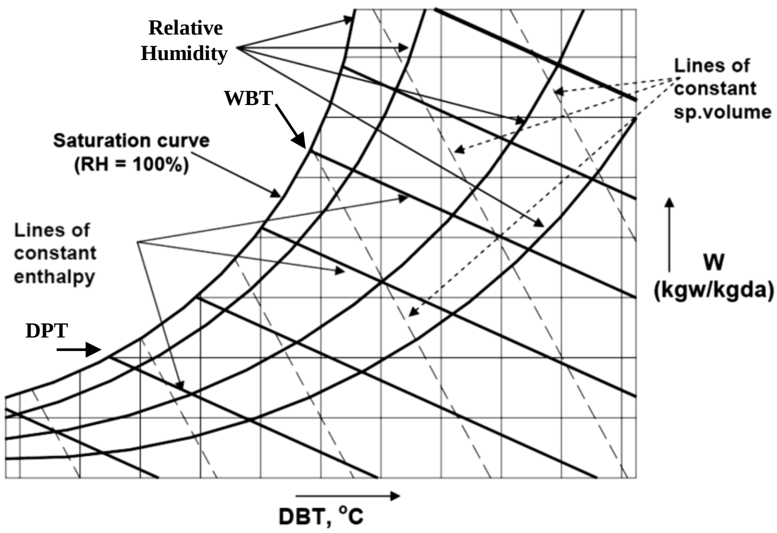


FIGURE (2) Schematic of a psychrometric chart.

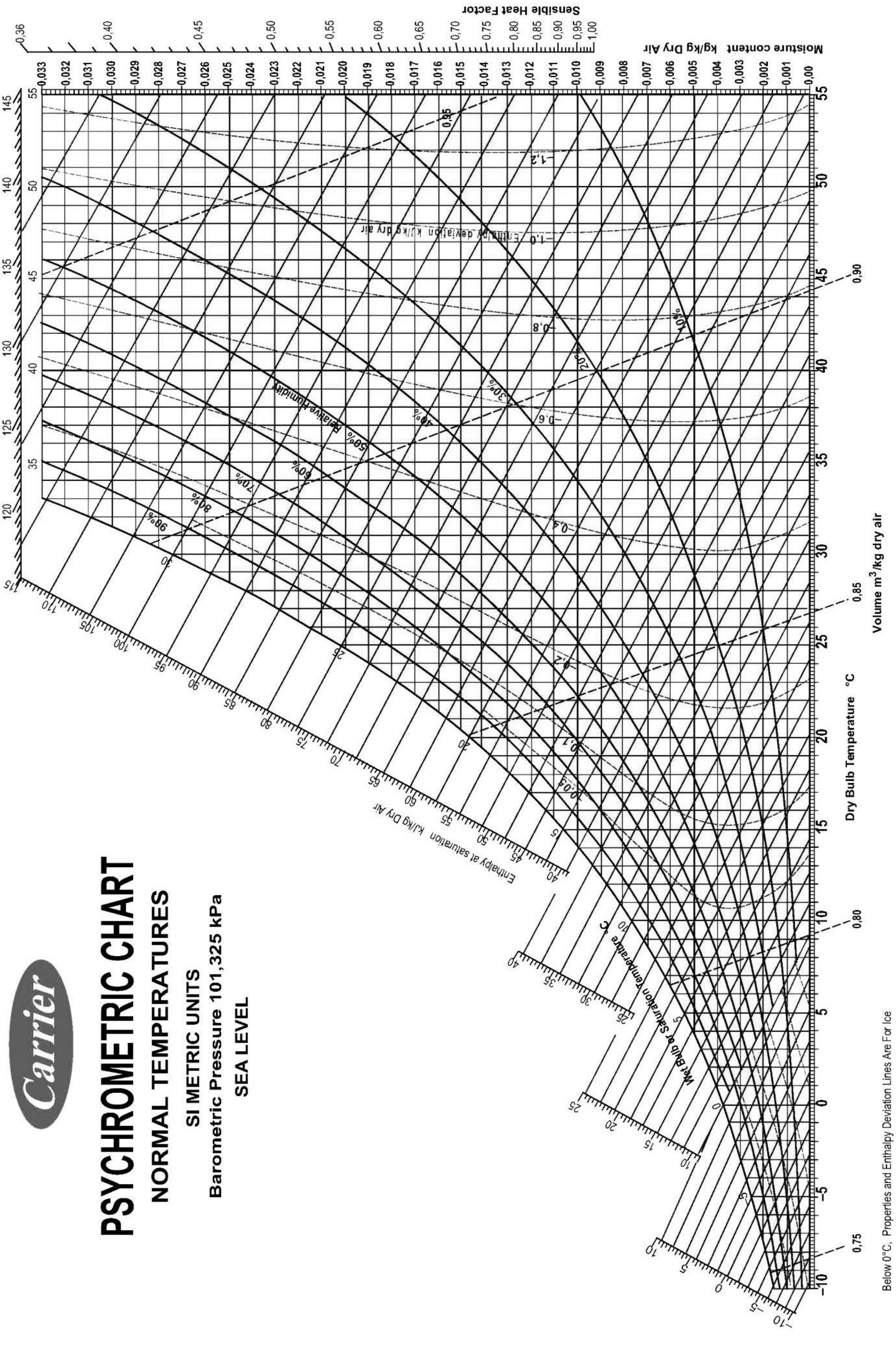
Example 8:-

Use the Psychrometric chart to find approximate enthalpy, moisture content, relative humidity, specific volume and dew point temperature of moist air at a state of 20°C *DBT*, 15°C *WBT* and 101.325 kPa.

Solution:-

$\boldsymbol{h} = 41.9\ kJ/kg$, $\mathbf{w} = 0.0086\ kg_w/kg_a$, $\mathbf{v} = 0.842\ m^3/kg$, $\boldsymbol{\phi} = 59\%$,

and $\boldsymbol{DPT} = 11.4^{\circ}\mathrm{C}$.


Carrier
PSYCHROMETRIC CHART
NORMAL TEMPERATURES
SI METRIC UNITS
Barometric Pressure 101,325 kPa
SEA LEVEL
Sensible Heat Factor
Moisture content kg/kg Dry Air
Dry Bulb Temperature °C
Volume m³/kg dry air
Enthalpy at saturation kJ/kg Dry Air
Wet Bulb or Saturation Temperature °C
Relative Humidity
Enthalpy deviation kJ/kg dry air
Below 0°C, Properties and Enthalpy Deviation Lines Are For Ice

The Psychrometry of Air-Conditioning Process

Sensible Heat and Latent Heat:-

Sensible heat:- is that heat energy associated with the change of air temperature between two state points.(**Sensible heat** causes a change in the air's dry-bulb temperature with no change in moisture content).

Latent heat:- is the heat energy associated with the change of the state of water vapor. (**Latent heat** causes a change in the air's moisture content with no change in dry-bulb temperature).

Adiabatic saturation process:-

Adiabatic saturation temperature is defined as that temperature at which water, by evaporating into air, can bring the air to saturation at the same temperature adiabatically. An adiabatic saturator is a device using which one can measure theoretically the adiabatic saturation temperature of air.

As shown in Figure (3), an adiabatic saturator is a device in which air flows through an infinitely long duct containing water. As the air comes in contact with water in the duct, there will be heat and mass transfer between water and air. If the duct is infinitely long, then at the exit, there would exist perfect equilibrium between air and water at steady state. Air at the exit would be fully saturated and its temperature is equal to that of water temperature. The device is adiabatic as the walls of the chamber are thermally insulated. In order to continue the process, make-up water has to be provided to compensate for the amount of water evaporated into the air. The temperature of the make-up water is controlled so that it is the same as that in the duct.

As there is no heat transfer between this insulated chamber and the surroundings, the latent heat required for the evaporation of water will come from the sensible heat released by the moist air. This process results in a drop in temperature of the moist air. At the end of this evaporation process, the moist air is always saturated. Such a process is called an *ideal adiabatic saturation process*, where an adiabatic process is defined as a process without heat transfer to or from the process.

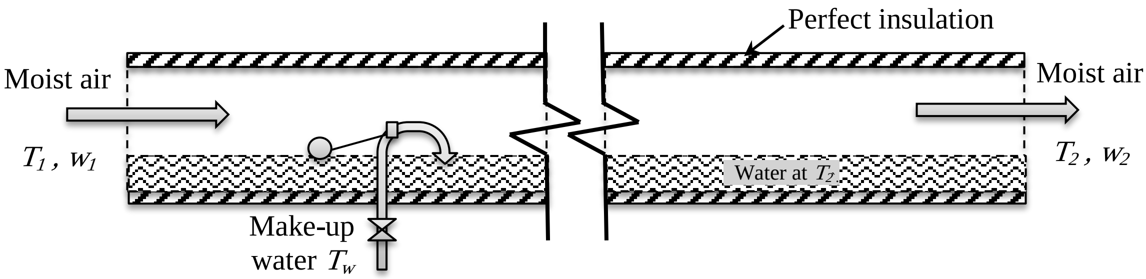


FIGURE (3) Adiabatic saturation process.

1- Sensible cooling:-

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. Figure (4) shows the sensible cooling process 1-2 on a psychrometric chart. The heat transfer rate during this process is given by:-

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

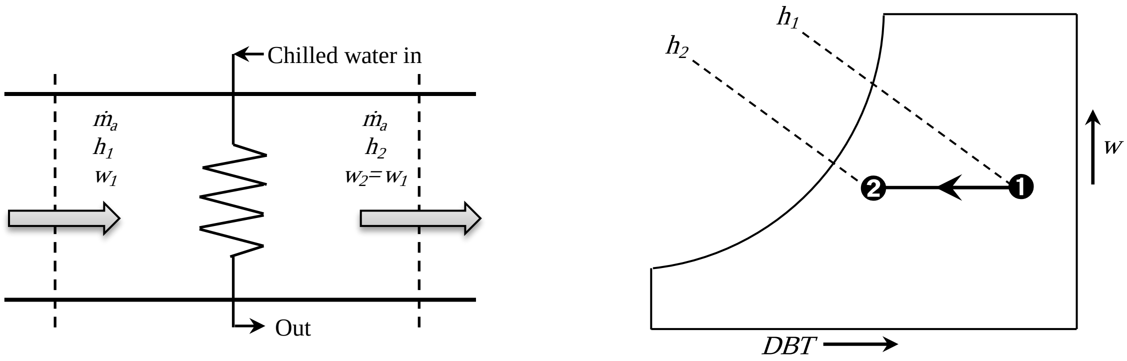
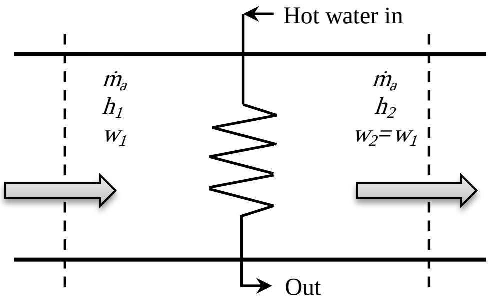


FIGURE (4) Sensible cooling process.

2- Sensible heating:-

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil, as shown in Figure (5). The heat transfer rate during this process is given by:-

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



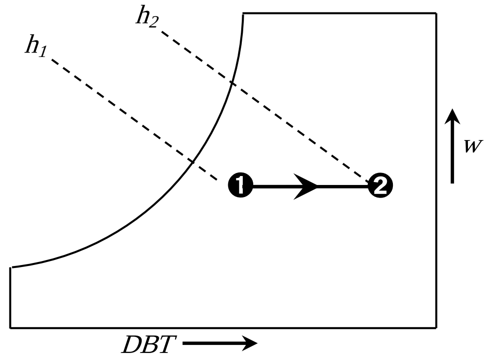


FIGURE (5) Sensible heating process.

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

Physical property	Sensible cooling	Sensible heating
DBT	**Decrease**	**Increase**
WBT	**Decrease**	**Increase**
DPT	**Constant**	**Constant**
w	**Constant**	**Constant**
v	**Decrease**	**Increase**
h	**Decrease**	**Increase**
R.H.	**Increase**	**Decrease**
Ps	**Constant**	**Constant**

Example 9:-

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

Solution:-

$Q_{sh} = \dot{m}_a (h_2 - h_1)$

h_1 = 41.88kJ/kg, h_2 = 62.31kJ/kg, & v_1 = *0.8439* m^3/kg from the psychrometric chart.

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

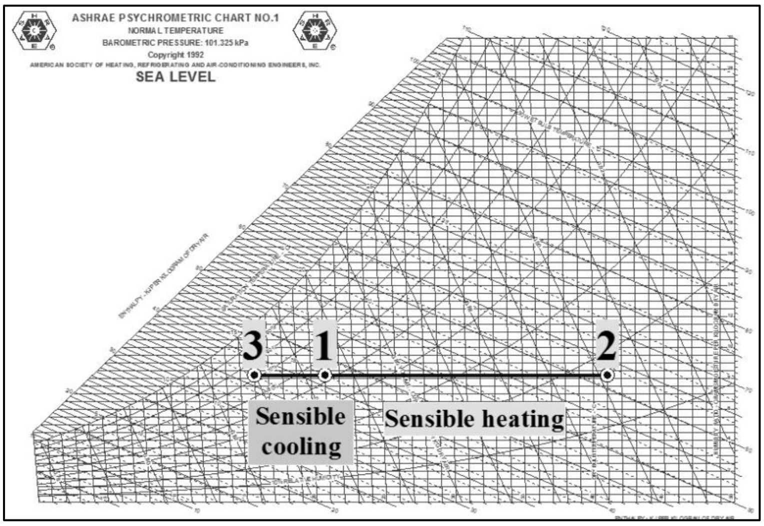
$Q_{1\text{-}2}$ = 1.777 (62.31 – 41.88) = 36.3kW

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

$Q = \dot{m}_w \, . \, c_{pw} \, (T_{wout} - T_{win})$

$36.3 = \dot{m}_w \times 4.186 \, (85 - 75)$

$\dot{m}_w = 0.867 \text{kg}_w/\text{sec}$



Example 10:-

If the moist-air mentioned in Example 9 is cooled sensibly by 5°C using cooler coil, what is the flow rate of chilled water necessary to effect this cooling if the flow return temperature of 10°C and 15°C satisfactory.

Solution:-

$Q_{sc} = \dot{m}_a \, (h_1 - h_3)$ (h_3 = 36.77kJ/kg)

$Q_{1\text{-}3}$ = 1.777 (41.88 – 36.77) = 9.1kW

Heat lost from air = heat gain by water

$9.1 = \dot{m}_w \times 4.186(15 - 10) \quad \rightarrow \quad \dot{m}_w = 0.434 \text{ kg}_w/\text{sec}$

3- Dehumidification:-

There are four methods whereby moist-air can be dehumidified:-

a- Cooling to a temperature below the dew point.
b- Adsorption.
c- Absorption.
d- Compression followed by cooling

The first method form is the required matter for this section. Cooling to a temperature below the dew point is done by passing the most air over a cooler coil or through an air washer, as shown in Figure (6). The dew point of air is when it is fully saturated i.e. at 100% saturation. When air is fully saturated it cannot hold any more moisture in the form of water vapor.

Figure (6) shows a sketch of psychrometric chart what happens when moist air is cooled and dehumidified. Since dehumidification is the aim, some of spray water or same part of cooler coil, must be at a temperature less than dew point temperature of the air entering equipment. The temperature (***T_c*** or ***ADP***) is the ***Apparatus Dew Point***. This term is use for both coils and washer, but in the case of coil alone, T_c is the mean coil surface temperature. It can be seen from the figure that the moisture content of air is reduced as well as its enthalpy and *DBT*. The percentage of saturation increases. It might be thought that the point 2 lie on the saturation curve, but there are no cooler coils or air washer have hundred percent efficient.

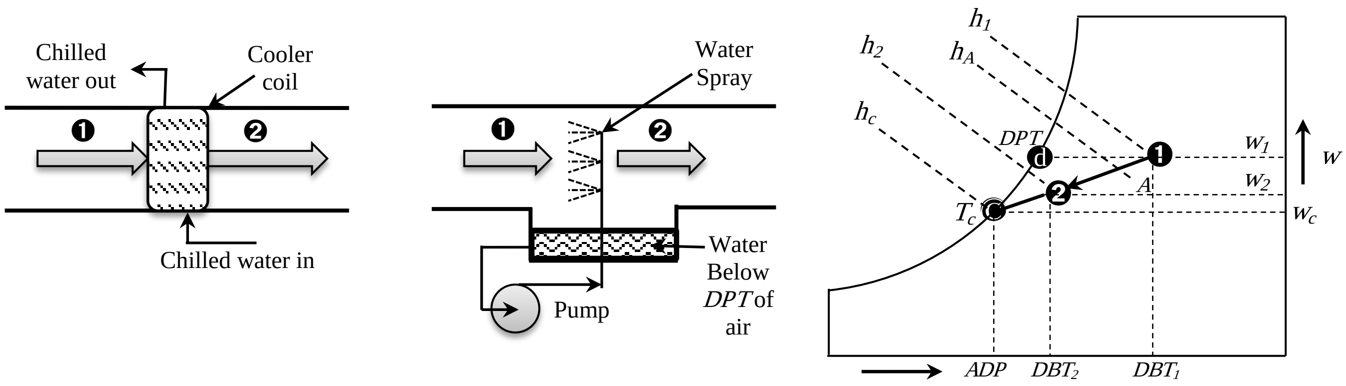


FIGURE (6) Cooling to a temperature below the dew point.

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

Mass balance:-

$$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a \qquad \text{For dry air}$$

$$\dot{m}_{a1} w_1 = \dot{m}_{a2} w_2 + \dot{m}_w \qquad \text{For water}$$

$$\dot{m}_w = \dot{m}_a (w_1 - w_2)$$

Heat balance:-

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + Q_t + \dot{m}_w h_{fc}$$

$$Q_t = \dot{m}(h_1 - h_2) - \dot{m}_w h_{fc}$$

$$Q_t = \dot{m}[(h_1 - h_2) - (w_1 - w_2) h_{fc}]$$

h_{fc}:- Enthalpy of saturated liquid at point c.

The 2nd term of the above equation is normally small compared to the other terms, so it can be neglected. Hence, $\boldsymbol{Q_t} = \dot{m}(h_1 - h_2)$

Latent heat (1-A):-

$$Q_L = \dot{m}_a(h_1 - h_A) = \dot{m}_a \cdot h_{fg}(w_1 - w_2)$$

Sensible heat (A-2):-

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

$$Q_t = Q_s + Q_L$$

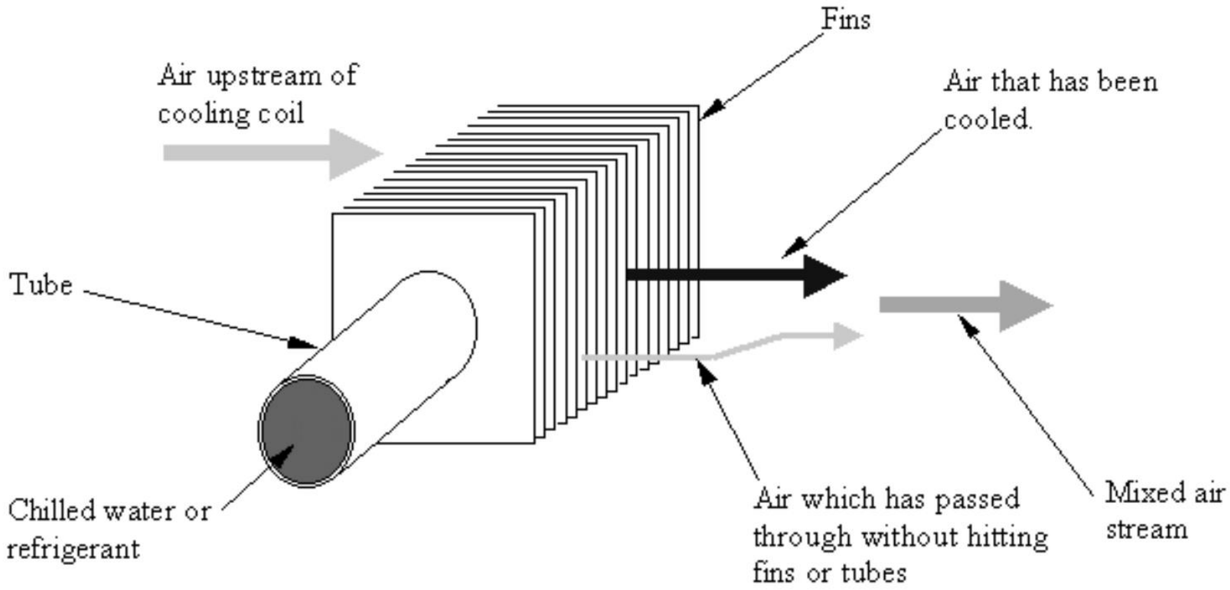
By separating the total heat transfer rate from the cooling coil into sensible and latent heat transfer rates, a useful parameter called Sensible Heat Factor (***SHF***) is defined. SHF is defined as the ratio of sensible to total heat transfer rate:-

$$SHF = \frac{Q_s}{Q_s + Q_L} = \frac{Q_s}{Q_t}$$

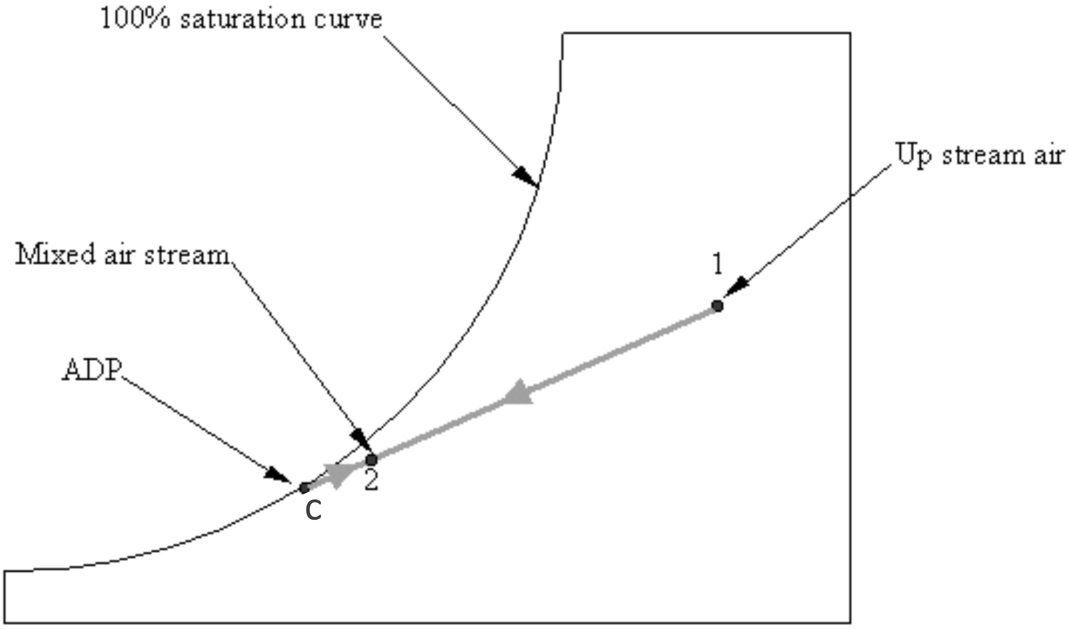
From the above equation, one can deduce that a SHF of 1.0 corresponds to no latent heat transfer and a SHF of 0 corresponds to no sensible heat transfer. A SHF of 0.75 to 0.80 is quite common in air conditioning systems in a normal dry-climate. A lower value of SHF, say 0.6, implies a high latent heat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

Cooling Coil Contact Factor:-

Some of the air going through a cooling coil does not come into contact with the tubes or fins of the cooling coil and is therefore not cooled to the *ADP* temperature. A mixing process therefore takes place as two air streams mix downstream of the cooling coil as shown in Figure (7). One air stream is cooled down to the *ADP* and the other air stream by-passes the coil surfaces to give an off-coil air temperature (mixed air stream) a little higher than the *ADP*. This may be looked upon as an inefficiency of the coil and is usually given as the cooling coil contact factor. The process is shown on the psychrometric chart below.



A SECTION OF COOLING COIL SHOWING AIR STREAMS



PSYCHROMETRIC CHART SHOWING COOLING COIL CONTACT FACTOR

FIGURE (7) Cooling Coil Contact Factor.

The contact factor can be calculated as:-

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

$$\beta = \frac{h_1 - h_2}{h_1 - h_c} = \frac{w_1 - w_2}{w_1 - w_c} = \frac{DBT_1 - DBT_2}{DBT_1 - DBT_c}$$

While the by-pass factor is:-

$$B.F. = \frac{Line\ 2 - c}{Line\ 1 - c}$$

$$B.F. = \frac{h_2 - h_c}{h_1 - h_c} = \frac{w_2 - w_c}{w_1 - w_c} = \frac{DBT_2 - DBT_c}{DBT_1 - DBT_c}$$

Also we have:- $\beta = 1 - B.F.$

Example 11:-

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

Solution:-

T_c = 11.5°C , h_1 = 60.5kJ/kg , h_2 = 32.5 kJ/kg , h_c = 31.5kJ/kg From the chart.

$$\beta = \frac{h_1 - h_2}{h_1 - h_c} = \frac{60.5 - 32.5}{60.5 - 31.5} = 0.9665$$

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

cooling load $= \frac{V}{v_1}(h_1 - h_2) = \frac{1.5}{0.87}(60.5 - 32.5) = 48.27kW$

4- Humidification:-

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

Humidification by water

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

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

Water injection:-

The simplest case to consider and one that provides the moist insight into change of state of the air stream subjected to humidification by the injection of water is where all the injected water is evaporated. Figure (8) shows what happens when total evaporation occur, air enters a spray chamber, all injected water being evaporated, and no falling to the bottom of the chamber to run to waste or to be circulated. The feed water temperature is important to release that, since total evaporation has occurred. State 2, 3 and 4 must be lie nearer to the saturation curve, but just how nearer will depend on the amount of water injected. Two equations, a heat balance and mass balance, provide the answer required:-

$h_1 + \dot{m}_w h_w = h_2$

$\dot{m}_{a1} + \dot{m}_w = \dot{m}_{a2}$

$w_2 = w_a + \dot{m}_w$ The associated kg of dry air be ignored.

$\dot{m}_w$:- the amount of feed water in *kgw/kga* flowing through the spray chamber.

Applying the heat balance:-

$$h_1 + \dot{m}_w h_w = h_2 = 1.005DBT_2 + w_2(2501.6 + 1.86DBT_2)$$

One thing is immediately apparent:-

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

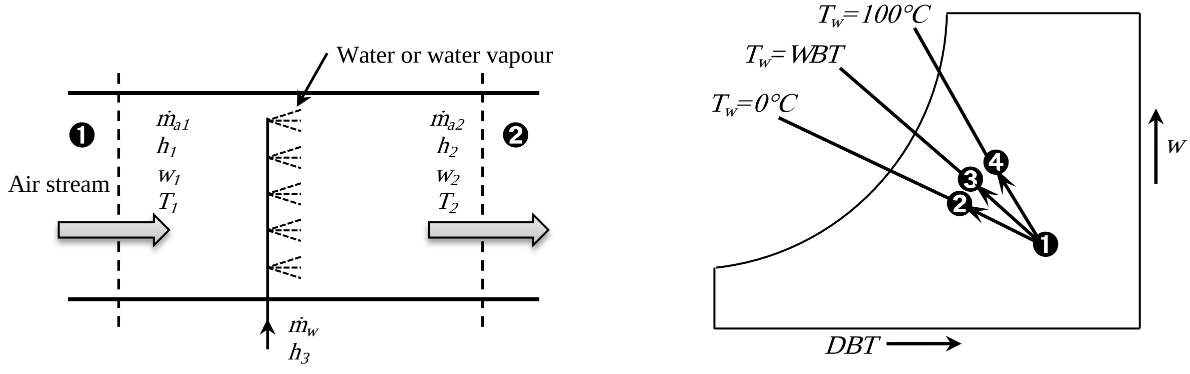


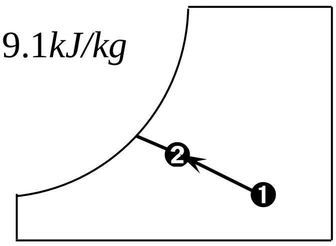
FIGURE (8) Humidification by water injection.

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

Example 12:-

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

Solution:-



$h_1 = 42 kJ/kg \;,\; w_1 = 8.314 gw/kga \;,\; h_w = h_{f100°C} = 419.1 kJ/kg$

$w_2 = w_1 + \dot{m}_w = 8.314 + 2 = 10.13\ gw/kga$

$h_2 = h_1 + \dot{m}_w h_w = 42 + 0.002 \times 419.1 = 42.838 kJ/kg$

$42.838 = 1.005 DBT_2 + 10.13 \times 10^{-3} (2501.6 + 1.86 DBT_2)$

$DBT_2 = 17.09°C$

Humidification by steam injection:-

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

$w_2 = w_1 + \dot{m}_s$

$\dot{m}_s$:- mass of steam injected in kg into one kg/s of dry air stream.

$h_2 = h_1 + \dot{m}_s h_s$

The change of state takes place almost along a line of constant *DBT* between limits defined by smallest and largest enthalpies of injected steam, provided the steam is in dry saturated condition. Figure (9) shows the possibility of steam injection The lowest possible enthalpy is for dry saturated steam at 100°C, the other extreme is provided by the steam which has maximum enthalpy of 2803 kJ/kg, which is exist at 30 bar and 234°C.

Packaged steam humidifier

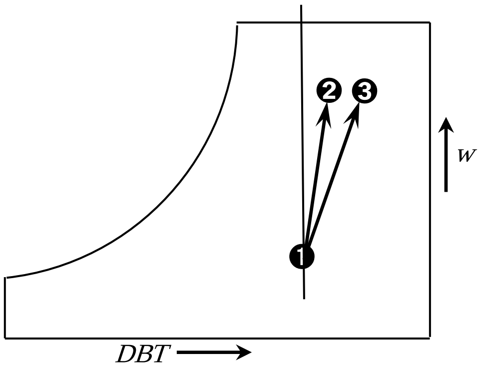


FIGURE (9) Humidification by steam injection.

Example 13:-

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

Solution:-

From the psychrometric chart:- $h_1 = 33.2$ kJ/kg , $w_1 = 1.937$ g/kga.

$h_s = 2675.8$ kJ/kg

$w_2 = 1.937 + 10 = 11.937$gw/kga

$h_2 = 33.2 + 0.01 \times 2675.8 = 59.9$kJ/kg

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

$DBT_2 = 29.2$°C

Example 14:-

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

Solution:-

From the psychrometeric chart:- $h_1 = 33.2$ kJ/kg , $w_1 = 1.937$ g/kga

The maximum enthalpy of steam is 2803 kJ/kg at 30 bar and 234°C saturated.

$h_3 = 33.2 + 0.01 \times 2803 = 61.2\text{kJ/kg}$

$61.2 = 1.005DBT_3 + 0.011937(2501.3 + 1.86DBT_3)$

$DBT_3 = 30.4°\text{C}$

It can be seen from Examples (13) and (14) for the range of states considered, the change in *DBT* is not very great, so, we can conclude that, the change of state following **steam injection is up a line of constant Dry Bulb Temperature.**

5- Mixing of air streams:-

Mixing of air streams at different states is commonly encountered in many processes, including in air conditioning. Depending upon the state of the individual streams, the mixing process can take place with or without condensation of moisture. Figure (10) shows an adiabatic mixing of two moist air streams during which no condensation of moisture takes place. As shown in the figure, when two air streams at state points 1 and 2 mix, the resulting mixture condition 3 can be obtained from mass and energy balance.

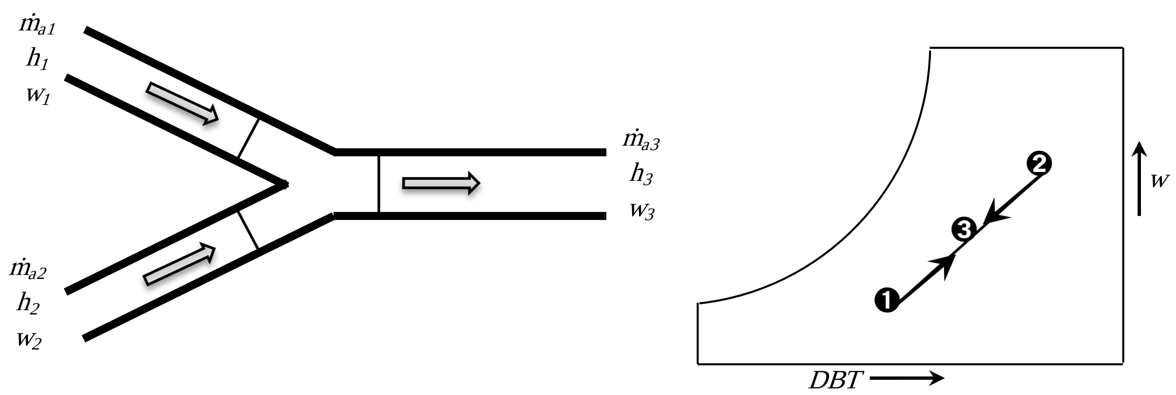


FIGURE (10) adiabatic mixing of two moist air streams.

From the mass balance of dry air and water vapour:-

$\dot{m}_{a1} + \dot{m}_{a2} = \dot{m}_{a3}$ Dry air

$w_1\dot{m}_{a1} + w_2\dot{m}_{a2} = w_3\dot{m}_{a3}$ Associated water vapour

$w_1\dot{m}_{a1} + w_2\dot{m}_{a2} = w_3(\dot{m}_{a1} + \dot{m}_{a2})$

$$w_3 = \frac{w_1\dot{m}_{a1} + w_2\dot{m}_{a2}}{\dot{m}_{a1} + \dot{m}_{a2}}$$

Making use of conservation of energy:-

$h_1\dot{m}_{a1} + h_2\dot{m}_{a2} = h_3\dot{m}_{a3}$ Associated water vapour

$$h_3 = \frac{h_1\dot{m}_{a1} + h_2\dot{m}_{a2}}{\dot{m}_{a1} + \dot{m}_{a2}}$$

Also you can use:- $\dfrac{\dot{m}_{a1}}{\dot{m}_{a2}} = \dfrac{w_2 - w_3}{w_3 - w_1} = \dfrac{h_2 - h_3}{h_3 - h_1}$

Example 15:-

Moist air at state 50°C *DBT*, 32°C *WBT* and 101.325 kPa barometric pressure mixes with moist air at 5°C *DBT*, 1°C *WBT* and 101.325 kPa barometric pressure. If the mass of dry air are 3kg/s and 2 kg/s respectively. Calculate for the mixture the *DBT*, enthalpy and moisture content by using:- **a-** Equations and table. **b-** Psychrometric chart.

Solution:-

- air-stream (1):- ***DBT*** = 50°C, ***WBT*** = 32°C, ***Pat*** = 101.325kPa, ***Pssw*** = 4.754 kPa

$Ps = 4.754 - 101.325 \times 6.66 \times 10^{-4}(50 - 32) = 3.54\text{kPa}$

$$w_1 = 0.622 \frac{3.54}{101.325-3.54} = 0.0225 kgw/kga$$

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

$h_1 = 1.005 \times 50 + 0.0225(2501.3 + 1.86 \times 50) = 108.6$ kJ/kg

- air-stream (2):- ***DBT*** = 5°C, ***WBT*** = 1°C, ***Pat*** = 101.325kPa, ***Pssw*** = 0.6566 kPa

$Ps = 0.6566 - 101.325 \times 6.66 \times 10^{-4}(5 - 1) = 0.3866\text{kPa}$

$$w_2 = 0.622 \frac{0.3866}{101.325-0.3866} = 2.3822 \times 10^{-3}\ kgw/kga$$

$h_2 = 1.005 \times 5 + 2.3822 \times 10^{-3}(2501.3 + 1.86 \times 5) = 11$ kJ/kg

- air-mixture (3):-

$$w_3 = \frac{w_1\dot{m}_{a1} + w_2\dot{m}_{a2}}{\dot{m}_{a1} + \dot{m}_{a2}} = \frac{0.0225 \times 3 + 2.3822 \times 10^{-3} \times 2}{5}$$
$$= 0.01446\ kgw/kga$$

$$h_3 = \frac{h_1\dot{m}_{a1} + h_2\dot{m}_{a2}}{\dot{m}_{a1} + \dot{m}_{a2}} = \frac{108.6 \times 3 + 11 \times 2}{5} = 69.56\ kJ/kg$$

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

$69.56 = 1.005DBT_3 + 36.168 + 0.0219DBT_3$

$DBT_3 = 32.36°C$

But if the DBT_3 were calculated by proportion according to the mass of dry air in the two mixing air streams a slightly answer result.

$$DBT_3 = \frac{DBT_1\dot{m}_{a1} + DBT_2\dot{m}_{a2}}{\dot{m}_{a1} + \dot{m}_{a2}} = \frac{50 \times 3 + 5 \times 2}{5} = 32°C$$

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

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

Part (**b**) of the example is a homework for the students.

Example 16:-

Saturated air leaving the cooling section of an air-conditioning system at 14°C at a rate of 50 m^3/min is mixed adiabatically with the outside air at 32°C and 60 percent relative humidity at a rate of 20 m^3/min. Assuming that the mixing process occurs at a pressure of 1 atm, determine the moisture content, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture.

Solution:-

The properties of each inlet stream are determined from the psychrometric chart to be:-

h_1 = 39.4 kJ/kg dry air , w_1 = 0.010 kgw/kga , v_1 = 0.826 m^3/kg dry air

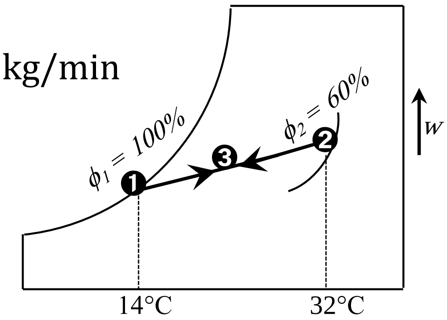
h_2 = 79.0 kJ/kg dry air , w_2 = 0.0182 kgw/kga , v_2 = 0.889 m^3/kg dry air

The mass flow rates of dry air in each stream are:-

$$\dot{m}_{a1} = \frac{V_1}{v_1} = \frac{50}{0.826} = 60.5\ \text{kg/min}$$

$$\dot{m}_{a2} = \frac{V_2}{v_2} = \frac{20}{0.889} = 22.5 \text{ kg/min}$$

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = 60.5 + 22.5 = 83 \text{ kg/min}$$



The moisture content and the enthalpy of the mixture can be determined from this:-

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{w_2 - w_3}{w_3 - w_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{60.5}{22.5} = \frac{0.0182 - w_3}{w_3 - 0.010} = \frac{79 - h_3}{h_3 - 39.4}$$

Which yield:- $w_3 = 0.0122$ kgw/kga , $h_3 = 50.1$ kJ/kg dry air

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:-

$DBT_3 = 19°C$, $\phi_3 = 89\%$, $v_3 = 0.844$ m^3/kg dry air

Finally, the volume flow rate of the mixture is determined from:-

$V_3 = \dot{m}_{a3}\, v_3 = 83 \times 0.844 = 70.1$ m^3/min

6- Cooling and dehumidification with reheat:-

Moist air at state **A** passes over finned tube of a cooler coil through which chilled water is flowing (see Figure (11)). The amount of dehumidification carried out is controlled by a dew point thermostat, **C1**, positioned after the coil. This thermostat regulates the amount of chilled water flowing through the coil by means of three way mixing valve **R1**. Air leaves the coil at state **B**, with moisture content suitable for the proper removal of latent heat gains occurring in the room being conditioned. The moisture content has been reduced from $\boldsymbol{w_a}$ to $\boldsymbol{w_b}$ and cooler coil has a mean surface temperature of $\boldsymbol{t_c}$.

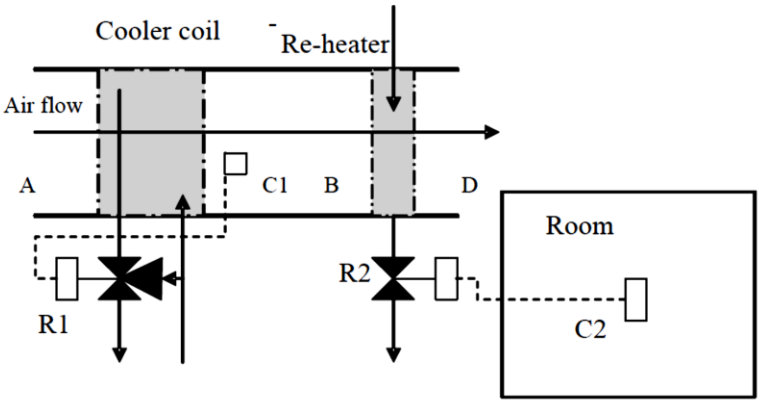
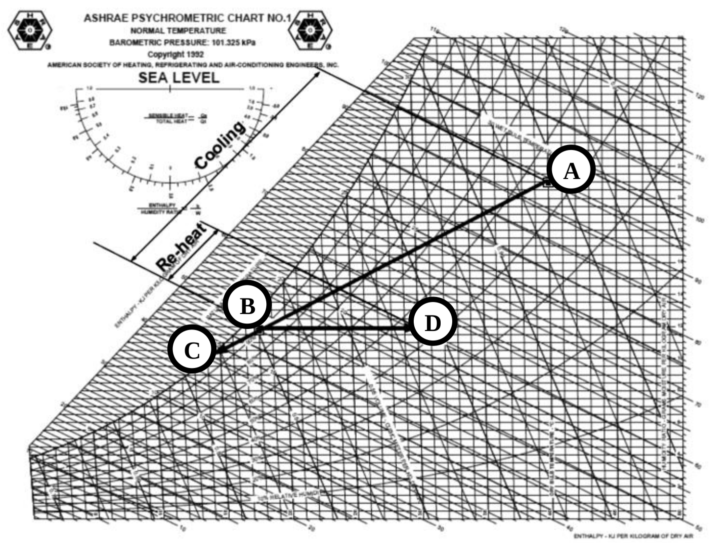


FIGURE (11) Cooling and dehumidification with reheat.

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



Example 17:-

Moist air at a state of 30°C *DBT*, 50% *RH* and a barometric pressure is 101.325 kPa. flows over a cooler coil and leaves it at a state of 12°C *DBT* and 0.0082 kg per kg of dry air. The air supplied to the space condition at sensible load of 2.4 kW and a temperature of 20°C *DBT* if the sensible load reduced in the space to 1.3 kW with constant latent load. Calculate:-

- **i.** Mass of dry air which must be supplied to the space.
- **ii.** Cooling load.
- **iii.** The temperature of the air supplied to the space.
- **iv.** Re – heat.

Solution:-

By using Psychrometric chart:- h_1 = 65.1 kJ/kg , h_2 = 32.9 kJ/kg , h_3 = 40.9 kJ/kg

i. $Q_s = \dot{m}_a \times C_{pa} \times \Delta T \rightarrow 2.4 = \dot{m}_a \times 1.005 \times (20 - 12) \rightarrow \dot{m}_a = 0.298$ kg/s
ii. Cooling load $= \dot{m}_a\,(h_1 - h_2) = 0.298 \times (65.1 - 32.9) = 9.59$ kW
iii. $Q_s = \dot{m}_a \times C_{pa} \times \Delta T \rightarrow 1.3 = 0.298 \times 1.005 \times (20 - T_3) \rightarrow T_3 = 15.66\ ^{\circ}C$
iv. Re – heat load $= \dot{m}_a\,(h_3 - h_2) = 0.298 \times (40.9 - 32.9) = 2.384$ kW

7- Pre-heating and humidification with re-heat:-

Air-conditioning plants which handle fresh air only are faced in winter with the task of increasing both the moisture content and the temperature of they supply to the conditioned space. Humidification is needed because the outside air in winter may have a very low moisture content, and if this air were to be introduced directly to the room there would be a low moisture content there as well, and when the air is heated to higher temperature its relative humidity may become very low. Therefore the room condition will be far from human comfort. A popular and effective approach is to preheat the air (**1-2**), pass it through air washer (**2-3**), where it undergoes adiabatic saturation, and then to reheat it to the temperature at which it must be supplied to the room. Figure (12) shows a typical plant. Opening the valve **R1** in the return pipeline from the pre-heater increases the heating output of the battery. Similarly، opening the control valve **R2**, associated with the re-heater, allows air at a higher temperature to be delivered to the room being conditioned. **C1** and **C2** are room humidistat and room thermostat, respectively.

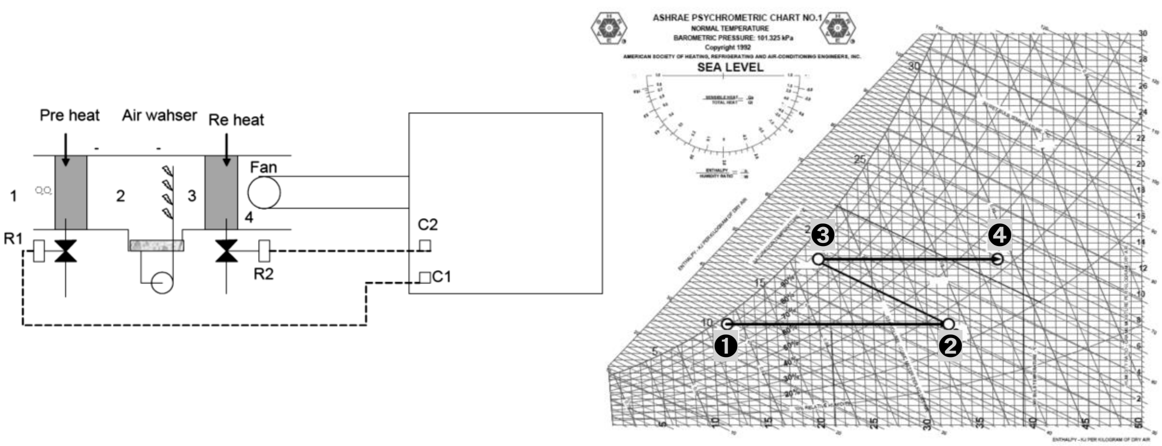


FIGURE (12) Pre-heating and humidification with re-heat.

Example 18:-

Air is pre–heat from 2°C *DBT* and 0.0038 kg per kg of dry air to 23.75°C *DBT*. It is then passed through an air washer having a humidifying efficiency of 90% and using recirculated spray water, the air leaves the air washer with 2.5m^3/s. The air is again reheated sensibly to obtain final desired condition. If the sensible heat losses from re–heating is 22kW and maintained temperature is 21°C *DBT*. Calculate:-

1- The total pre–heating required.
2- The temperature to which the air leaves the air washer.
3- The temperature to which the air should be supplied.
4- The total re–heating required.

Solution:-

From chart:- $h_1 = 11.55$ kJ/kg , $h_2 = 33.65$ kJ/kg , $v = 0.8465$ m^3/kg

$\dot{m}_a = 2.5/0.8465$

1- $Q_s = \dot{m}_a (h_2 - h_1) \rightarrow (2.5/0.8465) \times (33.65 - 11.55) = 65.268$ kW

2- For the humidifying efficiency we have:- $\eta = (w_3 - w_2) / (w_5 - w_2)$

$0.9 = (w_3 - 0.0038) / (0.00866 - 0.0038)$

$w_3 = 0.008174$ kg$_m$/kg$_a$

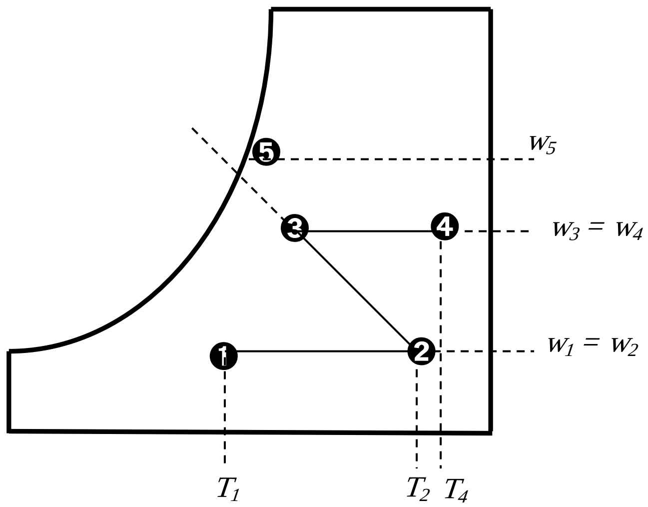
$T_3 = 12.7^\circ$C (from chart)

OR $\eta = (T_2 - T_3) / (T_2 - T_5) \rightarrow 0.9 = (23.75 - T_3) / (23.75 - 11.5)$

$T_3 = 12.725^\circ$C

3- $Q_s = \dot{m}_a \times C_{pa} \times \Delta T \rightarrow 22 = 2.953 \times 1.005 (T_4 - 21) \rightarrow T_4 = 28.25^\circ$C

4- $Q_s = \dot{m}_a \times (h_4 - h_3) = 2.953 \times (49.2 - 34.3) = 43.99$ kW



8- Summer and Winter Cycles:-

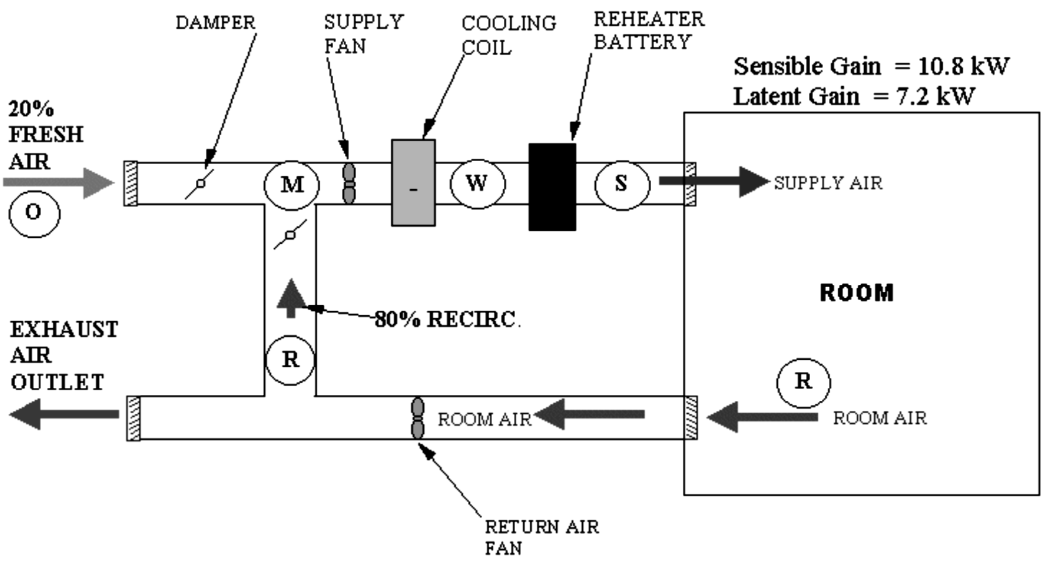
There are four basic processes for summer and winter air conditioning systems. The following basic processes are explained:-

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

Example 19:- (Summer Cycle)

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

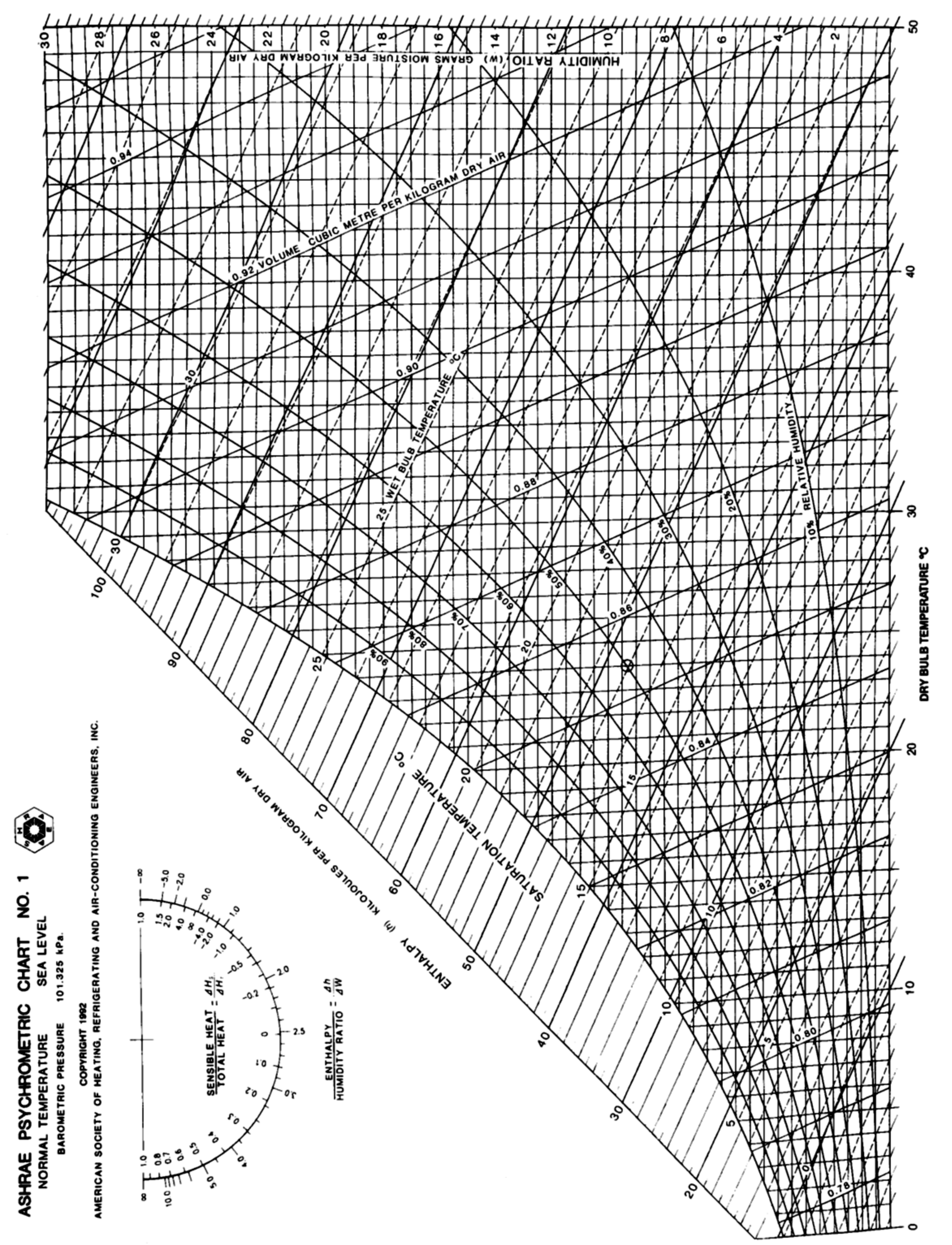
DATA:-
Outdoor condition is 28°C, 80% saturation.
The outdoor air and recirculated air ratio is 20% / 80%.
The Apparatus Dew Point *ADP* is 8°C
Neglect the cooling coil contact factor.

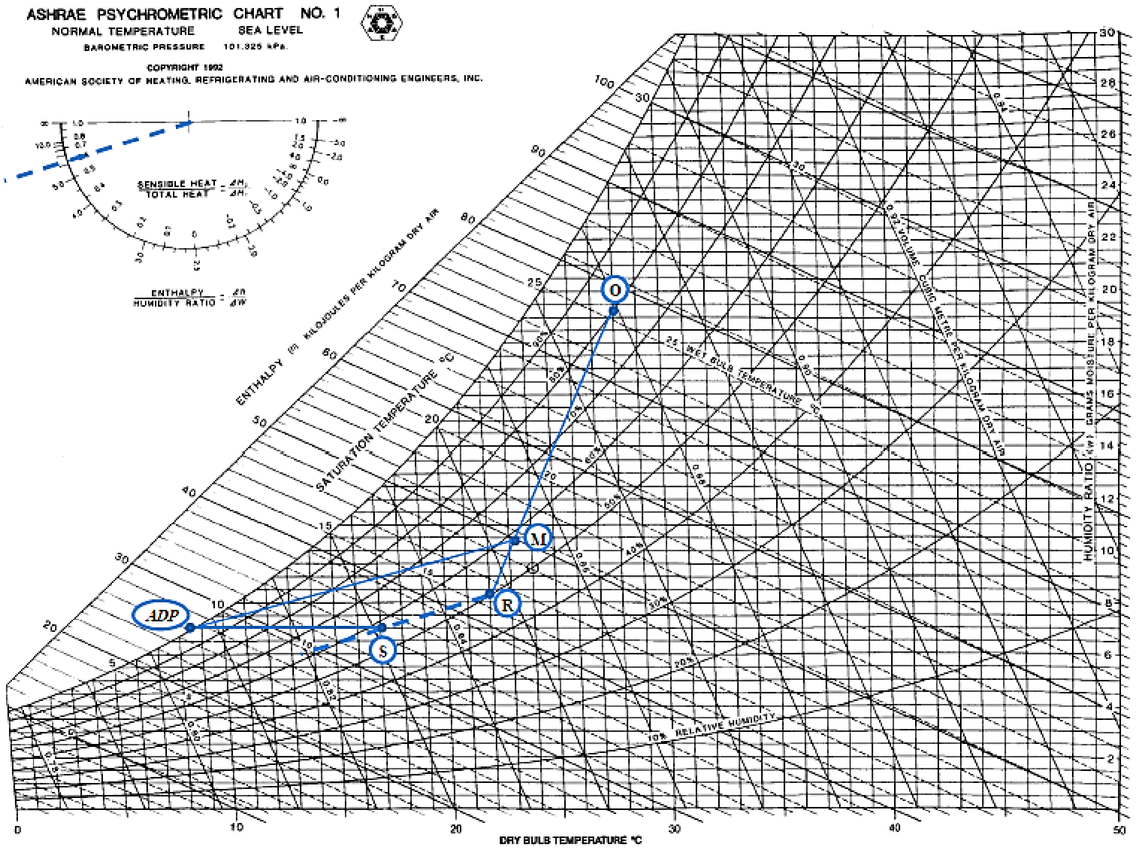


Solution:-

Mixing conditions:-

$$DBT_M = \frac{DBT_R \dot{m}_R + DBT_O \dot{m}_O}{\dot{m}_R + \dot{m}_O} = \frac{22 \times 0.8 + 28 \times 0.2}{0.8 + 0.2} = 23.2°C$$


ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE 101.325 kPa.
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔHt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY (h) KILOJOULES PER KILOGRAM DRY AIR
SATURATION TEMPERATURE °C
WET BULB TEMPERATURE °C
VOLUME CUBIC METRE PER KILOGRAM DRY AIR
HUMIDITY RATIO (W) GRAMS MOISTURE PER KILOGRAM DRY AIR
RELATIVE HUMIDITY
DRY BULB TEMPERATURE °C



Locate point **M** by plot a vertical line from *DBT* of 23.2°C, the intersection of the vertical line with the line **OR** located point **M**. From **M** draw a line to the ***ADP*** of 8°C.

Find Room Sensible Heat Factor (***RSHF***) as follow:-

$$RSHF = \frac{Q_{SR}}{Q_{SR} + Q_{LR}} = \frac{10.8}{10.8 + 7.2} = 0.6$$

Draw line parallel to the line of ***RSHF*** from **R**, The condition **S** must be lie on the line of ***RSHF***. Thus the supply conditions are *DBT* = 17°C, R.H= 58%, *w* = 7 *gw/kgair*

The mass flow rate of air can be calculated as:-

$$\dot{m}_a = \frac{Q_S}{C_{Pa}(DBT_R - DBT_S)} = \frac{10.8}{1.005\ (22 - 17)} = 2.149\ kg/s$$

Cooling coil load:- $Q_T = \dot{m}_a\,(h_M - h_C) = 2.149 \times (50 - 26.1) = 51.36\ kW$

Heater load:- $Q_T = \dot{m}_a\,(h_S - h_C) = 2.149 \times (34.8 - 26.1) = 18.69\ kW$

Example 20:- (Winter Cycle)

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

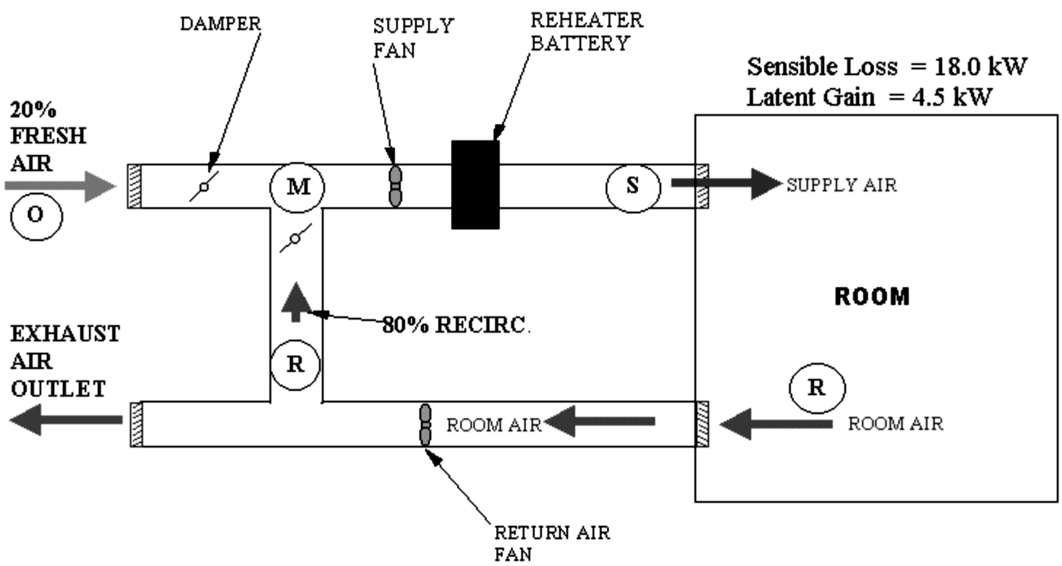
Indoor condition: 21°C dry-bulb temperature, 50% saturation.

Outdoor condition: 0°C *DBT*, 80% saturation.

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

No preheating or humidification takes place in this simplified example.

Solution:-



Mixing conditions:-

$$DBT_M = \frac{DBT_R \dot{m}_R + DBT_O \dot{m}_O}{\dot{m}_R + \dot{m}_O} = \frac{21 \times 0.8 + 0 \times 0.2}{0.8 + 0.2} = 16.8°C$$

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

Find Room Sensible Heat Factor (***RSHF***) as follow:-

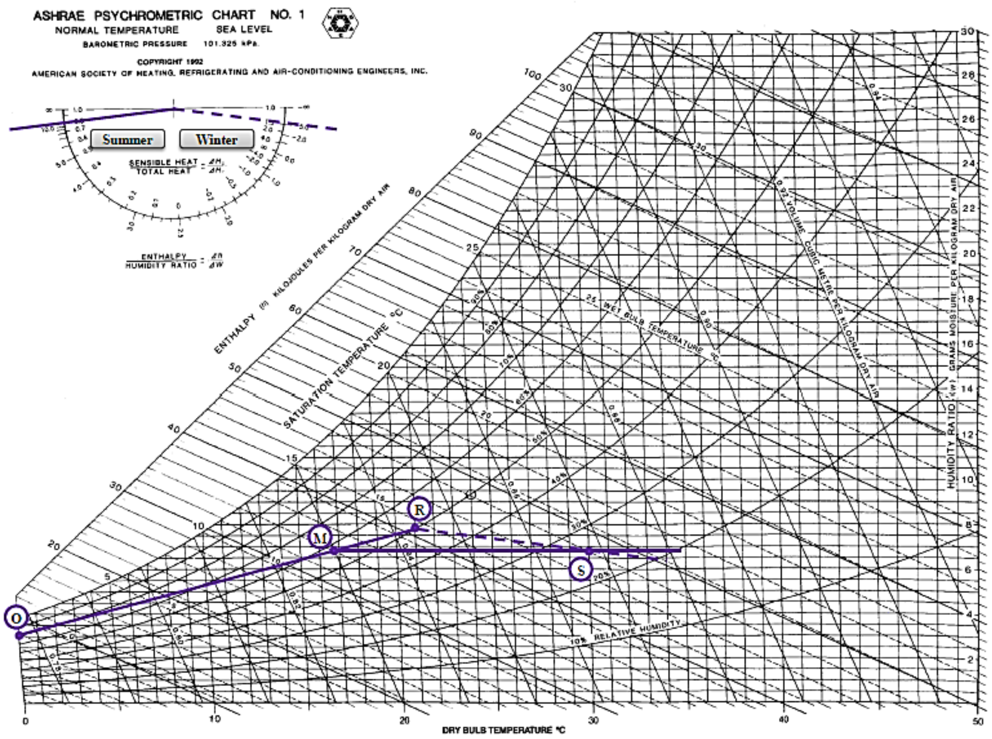
$$RSHF = \frac{Q_{SR}}{Q_{SR} + Q_{LR}} = \frac{18}{18 + 4.5} = 0.8$$

Draw line parallel to the line of ***RSHF*** from **R**, The condition **S** must be lie on the line of ***RSHF***. Thus the supply conditions are *DBT* = 30°C, R.H= 25%, *w* = 6.8 *gw/kgair*.

The mass flow rate of air can be calculated as:-

$$\dot{m}_a = \frac{Q_S}{C_{Pa}(DBT_R - DBT_S)} = \frac{18}{1.005\ (30 - 21)} = 1.99\ kg/s$$

Heater load:- $Q_T = \dot{m}_a\ (h_S - h_M) = 1.99 \times (48 - 34) = 27.86\ kW$



✓ **Note:-** Figure (13) shows the various Air-conditioning processes on the psychrometric chart.

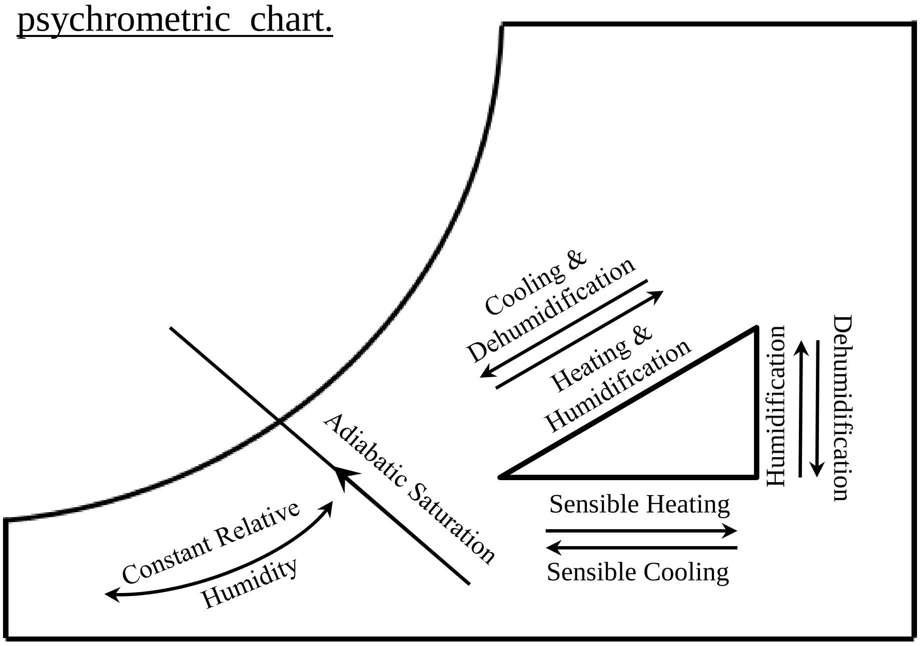


FIGURE (13) Various Air-conditioning processes.

Sheet No. (1)

Q1) For a sample of air having 22ºC *DBT*, relative humidity 30 percent at barometric pressure of 101.325 kPa, calculate:- 1.Vapour pressure, 2. Moisture content, 3. Vapour density, and 4. Enthalpy. Verify your result by Psychrometric chart.

Q2) 300 m^3/min of fresh air at 30ºC (*DBT*) dry bulb temperature and 50% *R.H.* is to be mixed with 800 m^3/min of recirculated air at 22ºC (*DBT*) dry bulb temperature and 10 ºC dew point temperature. Determine the enthalpy, specific volume, Moisture content, and dew point temperature of the mixture. (***Ans.*** h_3 = 47.86 *kJ/kg of dry air*).

Q3) In a heating application, moist air enters a steam heating coil at 10ºC, 50% *R.H.* and leaves at 30ºC. Determine the sensible heat transfer, if mass flow rate of air is 100kg of dry air per second.

Q4) The air enters a duct at 10 ºC and 80% *R.H.* at the rate of 150 m^3/min and is heated to 30 ºC without adding or removing any moisture. The pressure remains constant at 1 atmosphere. Determine the relative humidity of air at exit from the duct and the rate of heat transfer. (***Ans.*** *R.H.* ≈ 23.5% and Q = 3611.4 *kJ/min*).

Q5) The atmospheric air at 30ºC dry bulb temperature and 75% relative humidity enters a cooling coil at the rate 200 m^3/min. the coil dew point temperature is 14ºC and the by-pass factor of the coil is 0.1. Determine: 1. the temperature of air leaving the cooling coil; 2. the capacity of the cooling coil in tones of refrigeration and in kilowatt; 3. the amount of water vapour removed per minute; and 4. the sensible heat factor for the process.

Q6) 200 m^3 *of air per minute* is passed through the adiabatic humidifier. The condition of air at inlet is 40ºC dry bulb temperature and 15% relative humidity and the outlet condition is 25ºC dry bulb temperature and 20ºC wet bulb temperature. Find the dew point temperature and the amount of water vapour added to the air per minute.

Q7) The atmospheric air at 25ºC dry bulb temperature and 12ºC wet bulb temperatures is flowing at the rate of 100 m^3/min through the duct. The dry saturated steam at 100ºC is injected into the air steam at the rate of 72 kg per hour. Calculate the specific humidity and enthalpy of the leaving air. Also determine the dry bulb temperature, wet bulb temperature and relative humidity of the leaving air.

Q8) Air at 10°C dry bulb temperature and 90% relative humidity is to be heated and humidified to 35°C dry bulb temperature and 22.5°C wet bulb temperature. The air is pre-heated sensibly before passing to the air washer in which water is recalculated. The relative humidity of the air coming out of the air washer is 90%. This air is again reheated sensibly to obtain the final desired condition. Find:- 1.The temperature to which the air should be preheated, 2. The total heating required, 3.The make-up water required in the air washer.

Q9) A stream of outdoor air is mixed with a stream of return air in an air-conditioning system that operates at 101 kPa pressure. The flow rate of outdoor air is 2 kg/s, and its condition is 35°C dry-bulb temperature and 25°C wet-bulb temperature. The flow rate of return air is 3 kg/s, and its condition is 24°C and 50 percent relative humidity. Determine (a) the enthalpy of the mixture, (b) the moisture content of the mixture, (c) the dry-bulb temperature of the mixture from the properties determined in parts (a) and (b) and (d) the dry-bulb temperature by weighted average of the dry-bulb temperatures of the entering streams. (***Ans.*** h_3 = 58.777 *kJ/kg*, w_3 = 0.1194 *kg/kg*, DBT_3 = 28.6°C and 28.4°C).

Q10) The air conditions at the intake of an air compressor are 28°C, 50 percent relative humidity, and 101 kPa. The air is compressed to 400 kPa, then sent to an intercooler. If condensation of water vapor from the compressed air is to be prevented, what is the minimum temperature to which the air can be cooled in the intercooler?

Q11) Return air from a conditioned space at 21°C, 50% saturation, and a mass flow of 20 kg/s, mixes with outside air at 28°C dry bulb and 20°C wet bulb, flowing at 3 kg/s. What is the condition of the mixture?

Q12) Saturated air leaving the cooling section of an air-conditioning system at 14°C at a rate of 50 m^3/min is mixed adiabatically with the outside air at 32°C and 60 percent relative humidity at a rate of 20 m^3/min. Assuming that the mixing process occurs at a pressure of 1 atm, determine the specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture.

REFRIGERATION

Refrigeration may be defined as the process of achieving and maintaining a temperature below that of the surroundings, the aim being to cool some product or space to the required temperature.

In olden days refrigeration was achieved by natural means such as the use of ice or evaporative cooling. In earlier times, ice was either transported from colder regions or Harvested in winter and stored in ice houses for summer use. In Europe, America and other countries a number of icehouses were built to store ice. In 1806, Frederic Tudor, (who was later called as the "ice king") began the trade in ice by cutting it from the Hudson River and ponds of Massachusetts (in USA) and exporting it to various countries.

As mentioned above, refrigeration deals with cooling of bodies or fluids to temperatures lower than those of surroundings. This involves transfer heat from a cooler, low-energy reservoir, to a warmer, high-energy reservoir (see Figure (14)).

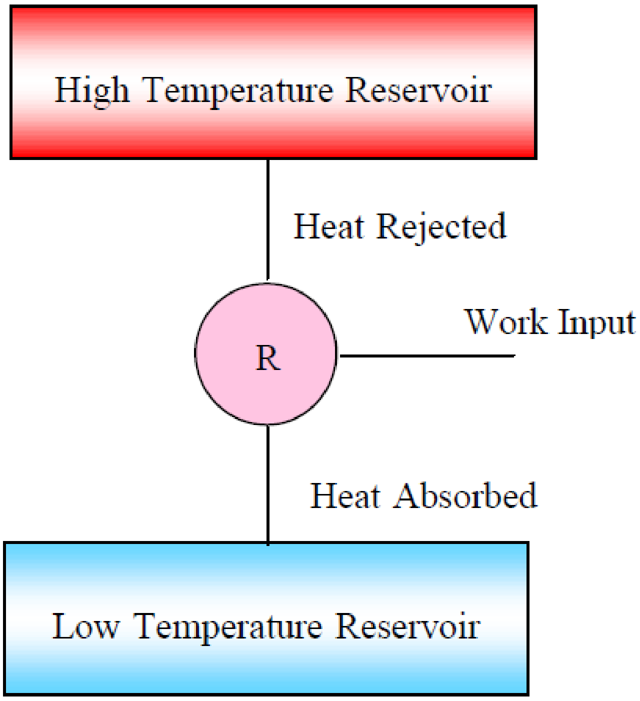


FIGURE (14) Schematic representation of refrigeration system.

In olden days, the main purpose of refrigeration was to produce ice, which was used for cooling beverages, food preservation and refrigerated transport etc.

Now-a-days refrigeration and air conditioning find so many applications that they have become very essential for mankind, and without refrigeration and air conditioning the basic fabric of the society will be adversely affected. Refrigeration and air conditioning are generally treated in a single subject due to the fact that one of the most important applications of refrigeration is in cooling and dehumidification as required for summer air conditioning. Of course, refrigeration is required for many applications other than air conditioning, and air conditioning also involves processes other than cooling and dehumidification. Figure (15) shows the relation between refrigeration and air conditioning in a pictorial form.

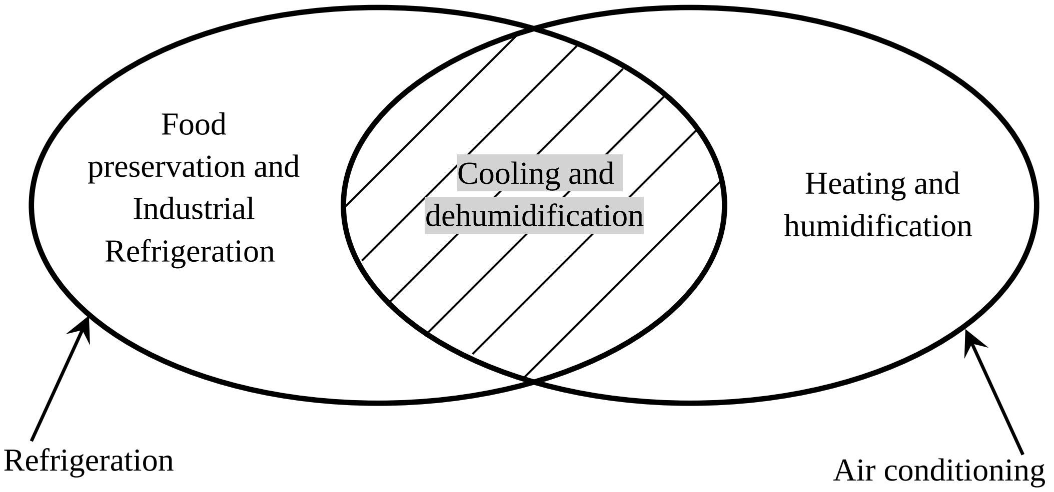


FIGURE (15) Relation between refrigeration and air conditioning.

The major applications of refrigeration can be grouped into the following five major equally important areas:-

1- Food processing, preservation and distribution.

2- Chemical and process industries.

3- Special Applications (manufacturing processes, applications in medicine, construction units … etc.).

4- Liquefaction of gases.

5- Comfort air-conditioning.

Units of refrigeration:-

The basic unit of refrigeration is the (Ton of Refrigeration).

Ton of Refrigeration:- A ton of refrigeration is the heat flux produced by the fusion of one ton of ice at 0 °C over 24 hours period (i.e., 1 ton of ice melting in 24 hr).

1 TR = 2000 Ib = 907.2 kg

Latent heat of fusion = 144 Btu/Ib = 334.9 kJ/kg

1 TR = (907.2 kg X 334.9 kJ/kg) / (24 hr X 3600 sec)

1 TR = 3.516 kW

1 TR = 12000 Btu/hr

Refrigerators And Heat Pumps:-

The heat flows in the direction of decreasing temperature, that is, from high-temperature regions to low-temperature ones. This heat-transfer process occurs in nature without requiring any devices. The reverse process, however, cannot occur by itself. The transfer of heat from a low-temperature region to a high-temperature one requires special devices called **refrigerators.**

Refrigerators are cyclic devices, and the working fluids used in the refrigeration cycles are called **refrigerants.** A refrigerator is shown schematically in Figure (16-a). Here Q_L is the magnitude of the heat removed from the refrigerated space at temperature T_L ,Q_H is the magnitude of the heat rejected to the warm space at temperature T_H , and $W_{net,in}$ is the net work input to the refrigerator.

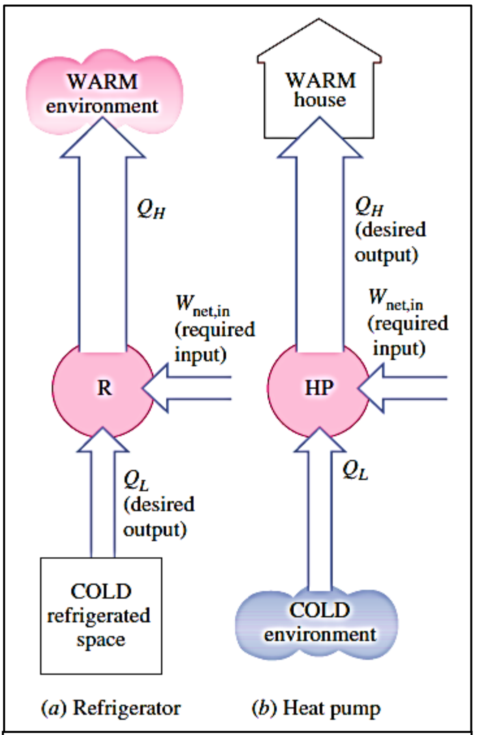


FIGURE (16) Refrigerators and Heat Pumps.

Another device that transfers heat from a low-temperature medium to a high-temperature one is the **heat pump.** Refrigerators and heat pumps are essentially the same devices; they differ in their objectives only. The objective of a refrigerator is to maintain the refrigerated space at a low temperature by removing heat from it. Discharging this heat to a higher-temperature medium is merely a necessary part of the operation, not the purpose. The objective of a heat pump, however, is to maintain a heated space at a high temperature. This is accomplished by absorbing heat from a low-temperature source, such as well water or cold outside air in winter, and supplying this heat to a warmer medium such as a house (Figure (16-b)).

Reversed Carnot Cycle:-

Refrigeration **Carno**t cycle in which the heat is absorbed from the cold reservoir and rejected to the hot reservoir. When it is a reversible cycle, all four processes that comprise the Carnot cycle can be reversed. Reversing the cycle does also reverse the directions of any heat and work interactions. The result is a cycle that operates in the counterclockwise direction on a *T-s* diagram, which is called the **reversed Carnot cycle.** A refrigerator or heat pump that operates on the reversed Carnot cycle is called a **Carnot refrigerator** or a **Carnot heat pump.**

Processes of Refrigeration Carnot cycle:-

Figure (17) shows the processes of refrigeration Carnot cycle:-

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

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

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

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

Work input (process 1-2) = $Q_H - Q_L$

Heat rejected (process 2-3) $Q_H = T_H (S_2 - S_3)$

Heat absorbed (process 4-1) $Q_L = T_L (S_1 - S_4)$

Since $S_3 = S_4$ & $S_2 = S_1 \rightarrow \quad Q_L = T_L (S_2 - S_3)$

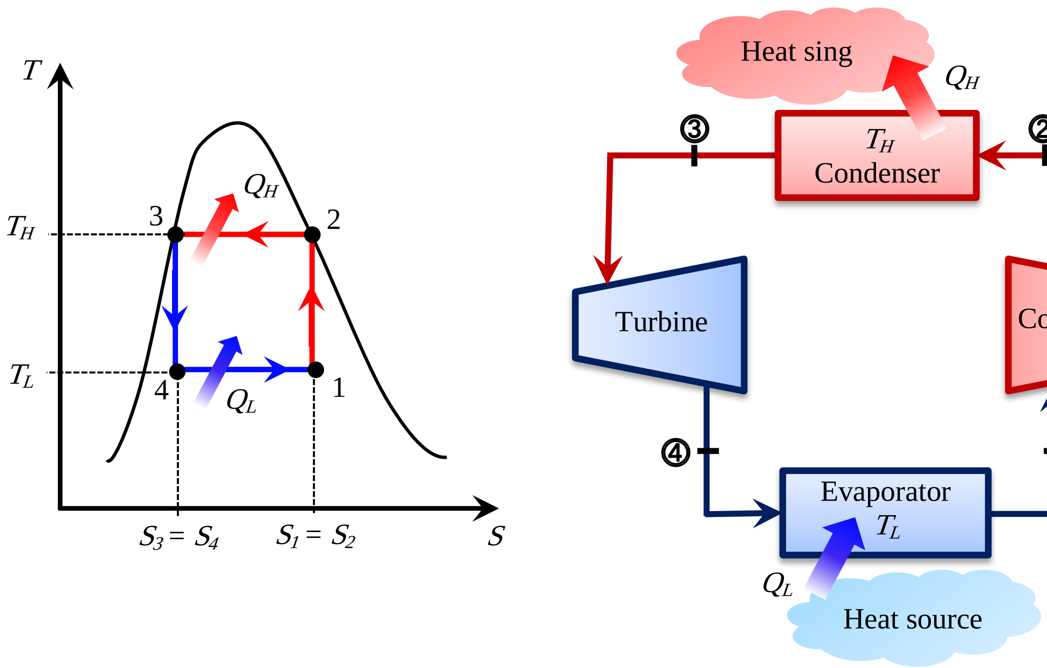


FIGURE (17) Processes of refrigeration Carnot cycle.

Coefficient Of Performance (COP):-

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

$$COP_r = \frac{Q_L}{W} = \frac{T_L\,(S_2 - S_3)}{T_H\,(S_2 - S_3) - T_L\,(S_2 - S_3)}$$

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

The performance of heat pumps is expressed in terms of coefficient of performance (COP) as:-

$$COP_{hp} = \frac{Q_H}{W} = \frac{T_H}{T_H - T_L}$$

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

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

Example 21:-

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

Solution:-

$$COP = \frac{T_L}{T_H - T_L} = \frac{-30 + 273}{32 - (-30)} = 3.91$$

actual $COP_r = 0.75 \times 3.91 = 2.939$

$$COP_r = \frac{Q_L}{W} \qquad \rightarrow \qquad 2.939 = \frac{Q_L}{1}$$

$Q_L = 2.939\ kW_{\ of\ refrigeration} / kW_{\ of\ work\ input}$

Example 22:-

A heat pump works on a reversed Carnot cycle. The temperature in the condenser coils is 27°C and that in the evaporator coils is -23°C. For a work input of 1 kW, how much is the heat pumped?

Solution:-

$$COP_{hp} = \frac{Q_H}{W} = \frac{T_H}{T_H - T_L} = \frac{300}{300 - 250} = 6$$

$Q_H = 6 \times 1 = 6$ kW

VAPOUR COMPRESSION CYCLE

Simple vapour compression cycle:-

A simple vapour compression refrigeration system consists of the following equipments:-

① Compressor, ② Condenser, ③ Expansion valve, and ④ Evaporator.

The schematic diagram of the arrangement is as shown in Figure (18). The low temperature, low pressure vapour at state 1 is compressed by a compressor to high temperature and pressure vapour at state 2. This vapor is condensed into high pressure vapour at state 3 in the condenser and then passes through the expansion valve. Here, the vapour is throttled down to a low pressure liquid (at state 4) and passed on to an evaporator, where it absorbs heat from the surroundings from the circulating fluid (being refrigerated) and vaporizes into low pressure vapour at state 4. The cycle then repeats. The exchange of energy is as follows:-

i. Compressor requires work, *W*. The work is supplied to the system from the surroundings.
ii. During condensation, heat Q_C the equivalent of latent heat of condensation is lost from the refrigerator.
iii. There is no exchange of heat during throttling process through the expansion valve as this process occurs at constant enthalpy.
iv. During evaporation, heat Q_E equivalent to latent heat of vaporization is absorbed by the refrigerant.

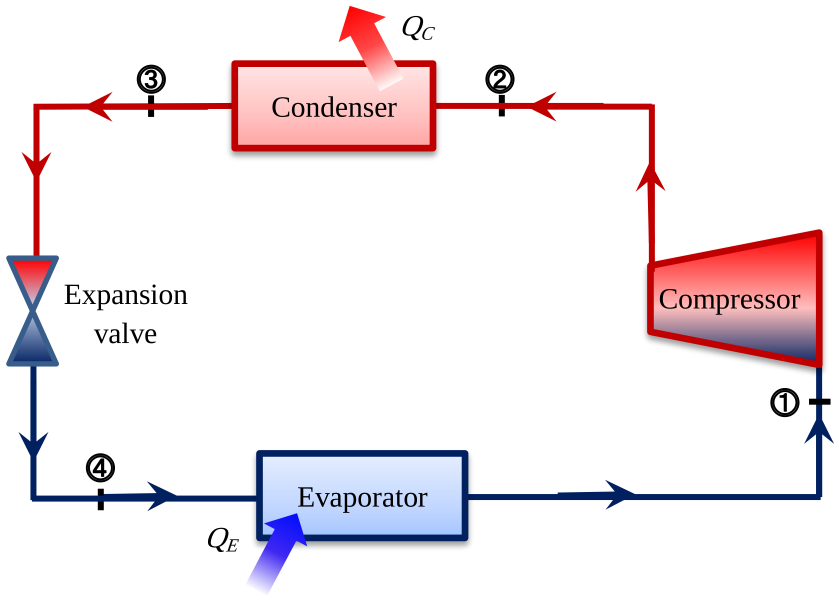
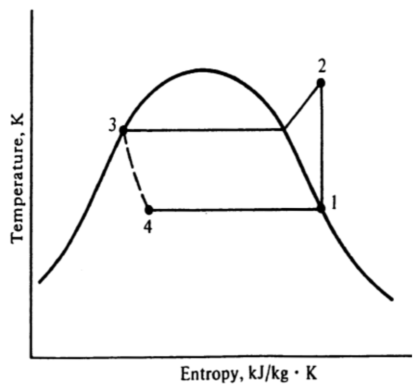


FIGURE (18) Simple vapour compression cycle.

The standard vapour compression cycle is shown on the temperature-entropy diagram in the side Figure. The processes constituting the standard vapour compression cycle are:-



(1-2) Reversible and adiabatic compression from saturated vapour to the condenser pressure.

(2-3) Reversible rejection of heat at constant pressure, causing desuperheating and condensation of the refrigerant.

(3-4) Irreversible expansion at constant enthalpy from saturated liquid to the evaporator pressure.

(4-1) Reversible addition of heat at constant pressure causing evaporation to saturated vapour.

The pressure-enthalpy diagram is the usual graphic means of presenting refrigerant properties. In refrigeration practice, the enthalpy is one of the most important properties sought, and the pressure can usually be determined most easily. A skeleton pressure-enthalpy diagram is shown in Figure (19). The pressure is the ordinate and the enthalpy the abscissa.

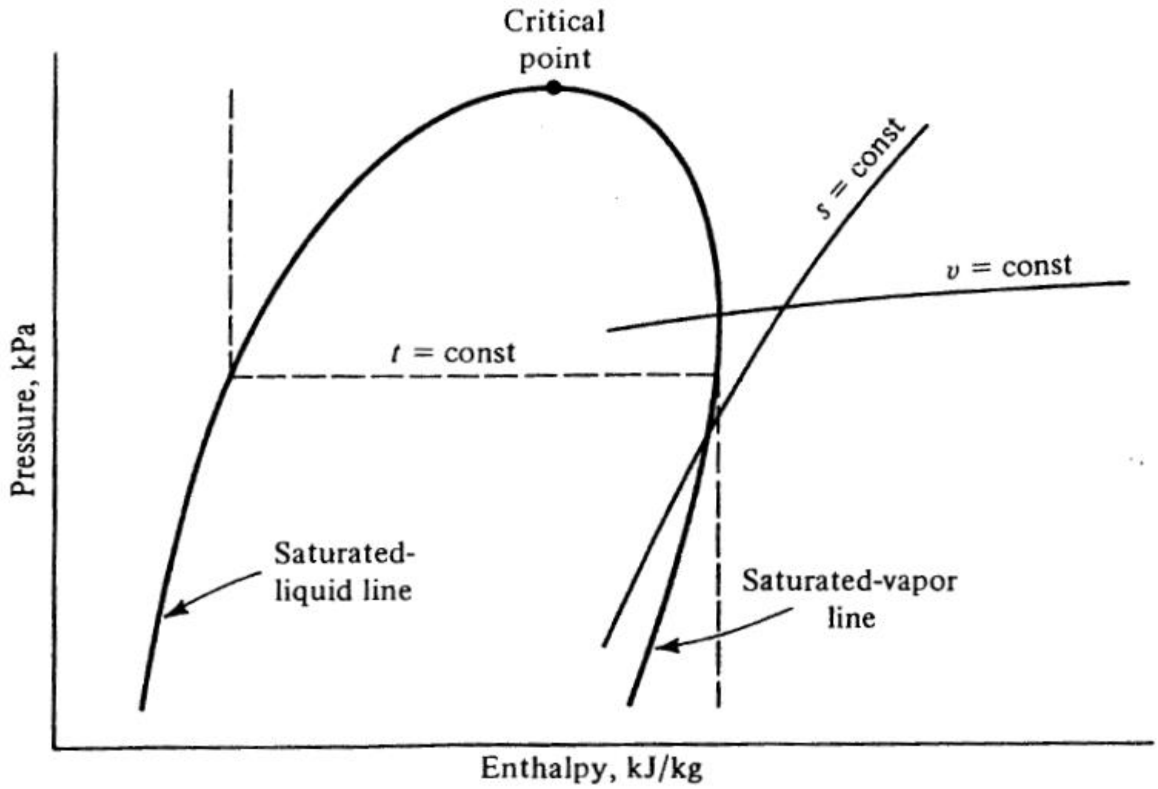


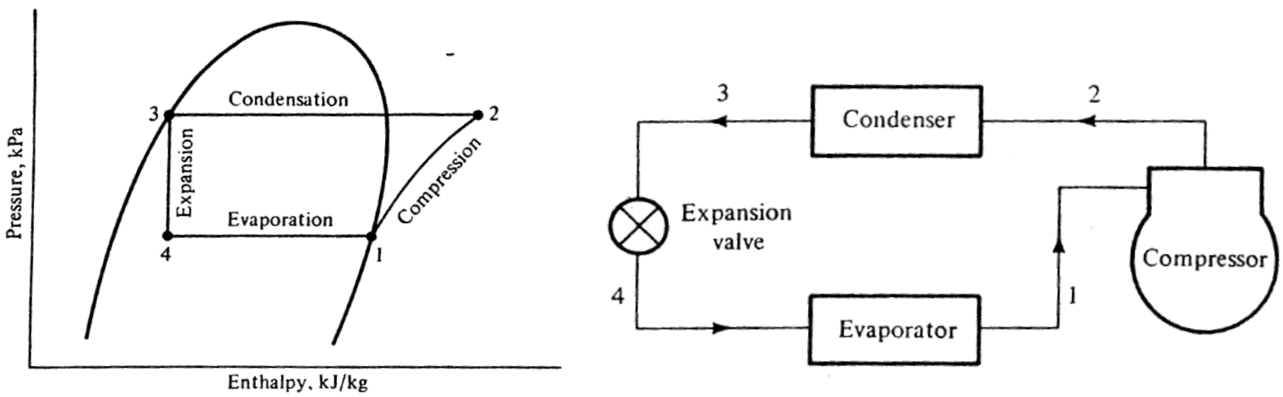
FIGURE (19) The pressure-enthalpy diagram.

With the saturated-vapour and saturated-liquid lines as the reference, lines of constant temperature, entropy, and specific volume appear on the diagram. The constant temperature line is horizontal in the mixture region because here the temperature must correspond with the saturation pressure. The subcooled-liquid or compressed-liquid region is to the left of the saturated-liquid line. In this region the constant-temperature line is practically vertical. The temperature of a compressed liquid therefore determines the enthalpy and not the pressure. This statement conforms to the standard practice in using steam tables at moderate pressures. To find the enthalpy of liquid water that is subcooled, the enthalpy is read as the enthalpy of saturated liquid at the existing temperature, even though the actual pressure is higher than the saturation pressure. The superheat region is to the right of the saturated-vapour line. In the superheat region the line of constant temperature drops first slightly to the right and then vertically. When the line of constant temperature becomes vertical, Δh = (const) (Δt), the typical relationship of enthalpy and temperature of a perfect gas.

The line of constant specific volume slopes upward to the right. Lines of higher specific volumes are found at progressively lower pressures.

The line of constant entropy runs upward to the right. A reversible and adiabatic compression, which is isentropic, shows the expected increase in enthalpy as the pressure increases during a compression.

What would be the appearance of the standard vapor-compression cycle on the pressure-enthalpy diagram? Figure (20-(a)) shows the processes which constitute the cycle, and Figure (20-(b)) is a schematic diagram of the equipment. Process (1-2) is the isentropic compression along the constant-entropy line from saturated vapour to the condenser pressure. Process (2-3) is the constant-pressure desuperheating and condensation, which is a straight horizontal line on the pressure-enthalpy diagram. The throttling process, (3-4), is one of constant enthalpy and therefore is vertical on the chart. Finally, the evaporation process (4-1) is a straight horizontal line because the flow of refrigerant through the evaporator is assumed to be at a constant pressure.



(a) **(b)**

FIGURE (20) (a) The standard vapour compression cycle on the pressure-enthalpy diagram; (b) Flow diagram.

Performance of the standard vapour compression cycle:-

With the help of the pressure-enthalpy diagram, the significant quantities of the standard vapour compression cycle will be determined. These quantities are the work of compression, the heat rejection rate, the refrigerating effect, the coefficient of performance, the volume rate of flow per kilowatt of refrigeration, and the power per kilowatt of refrigeration.

The work of compression in kilojoules per kilogram is the change in enthalpy in process (1-2) of Figure (20-(a)) or $h_2 - h_1$. This relation derives from the steady-flow energy equation:-

$$h_1 + Q = h_2 + W$$

where changes in kinetic and potential energy are negligible. Because in the adiabatic compression the heat transfer Q is zero, the work W equals $h_2 - h_1$. Knowledge of the work of compression is important because it may be one of the largest operating costs of the system.

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

The heat rejection in kilojoules per kilogram is the heat transferred from the refrigerant in process (2-3), which is $h_2 - h_3$. This knowledge also comes from the steady-flow energy equation, in which the kinetic energy, potential energy, and work terms drop out. The value of the heat rejection is used in sizing the condenser and calculating the required flow quantities of the condenser cooling fluid.

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

The process described by initial and final state points (3-4) occur in the throttling valve when the pressure of the liquid is reduced from condensation pressure to evaporation pressure as the liquid passes through the throttle expansion valve. This process is throttling type adiabatic expansion in which enthalpy of working substance remains the same, i.e $h_3 = h_4$.

The refrigerating effect in kilojoules per kilogram is the heat transferred in process (4-1), or $h_1 - h_4$. Knowledge of the magnitude of the term is necessary because performing this process is the ultimate purpose of the entire system.

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

The coefficient of performance of the standard vapour compression cycle is the refrigerating effect divided by the work of compression:-

$$\boldsymbol{COP} = \frac{Refrigeration\, Effect}{Work\ of\ Compressor} = \frac{Q_{Evap.}}{W_{Comp.}} = \frac{h_1 - h_4}{h_2 - h_1}$$

Sometimes the volume flow rate is computed at the compressor inlet or state point 1. The volume flow rate is a rough indication of the physical size of the compressor. The greater the magnitude of the term, the greater the displacement of the compressor in cubic meters per second must be.

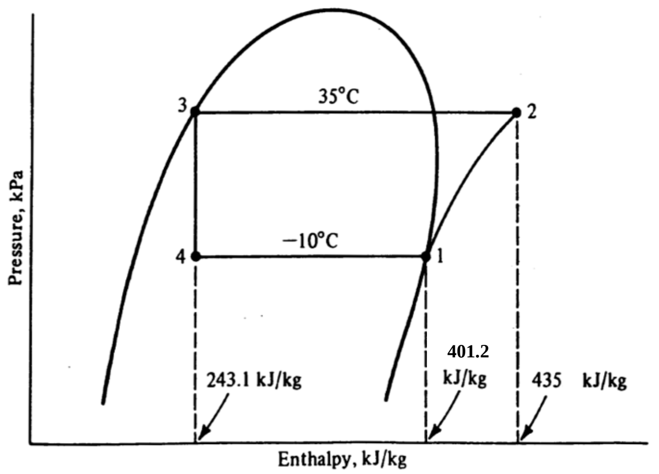
Example 23:-

A standard vapour compression cycle developing 50 kW of refrigeration using refrigerant 22 operates with a condensing temperature of 35°C and an evaporating temperature of -10°C. Calculate ***(a)*** the refrigerating effect in kilojoules per kilogram, ***(b)*** the circulation rate of refrigerant in kilograms per second, **(c)** the power required by the compressor in kilowatts, ***(d)*** the coefficient of performance, ***(e)*** the volume flow rate measured at the compressor suction, and ***(f)*** the compressor discharge temperature.

Solution:-

As the first step in the solution, sketch the pressure-enthalpy diagram and determine from R22 P-h Diagram the enthalpies at key points. The value of h_1 is the enthalpy of saturated vapour at -10°C, which is 401.2 kJ /kg.

To find h_2 move at a constant entropy from point 1 until reaching the saturation pressure corresponding to 35°C. This condensing pressure is 1354 kPa, and the value of h_2 is 435 kJ/kg.



The values of h_3 and h_4 are identical and are equal to the enthalpy of saturated liquid at 35°C, which is 243.1 kJ/kg. Therefore

h_1 = 401.2 kJ/kg, h_2 = 435.2 kJ/kg, and $h_3 = h_4$ = 243.1 kJ/kg

(a) The refrigerating effect is:- $h_1 - h_4$ = 401.2 – 243.l = 158.1 kJ/kg

(b) The circulating rate of refrigerant can be calculated by dividing the refrigerating capacity by the refrigerating effect:-

$$Flow\ rate = \frac{50\ kW}{158.1\ kJ/kg} = 0.316\ kg/s$$

(c) The power required by the compressor is the work of compression per kilogram multiplied by the refrigerant flow rate:-

Compressor power = (0.316 kg/s) (435.2 – 401.2 kJ/kg)= 10.7 kW

(d) The coefficient of performance is the refrigerating rate divided by the compressor power

$$\boldsymbol{COP} = \frac{50\ kW}{10.7\ kW} = 4.67$$

(e) The volume rate of flow at the compressor inlet requires knowledge of the specific volume of the refrigerant at point 1. From R22 Table or Chart this value is 0.0652 m^3/kg, and so

Volume flow rate = (0.316 kg/s) (0.0652 m^3/kg) = 0.0206 m^3/s = 20.6 L/s

(f) The compressor discharge temperature is the temperature of superheated vapor at point 2 which from R22 chart is found to be 57°C.

Example 24:-

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

Solution:-

From R-134a p-h diagram we can find the enthalpies at each point as follow:-

h_1 = 387 kJ/kg, h_2 = 434 kJ/kg, and $h_3 = h_4$ = 258 kJ/kg

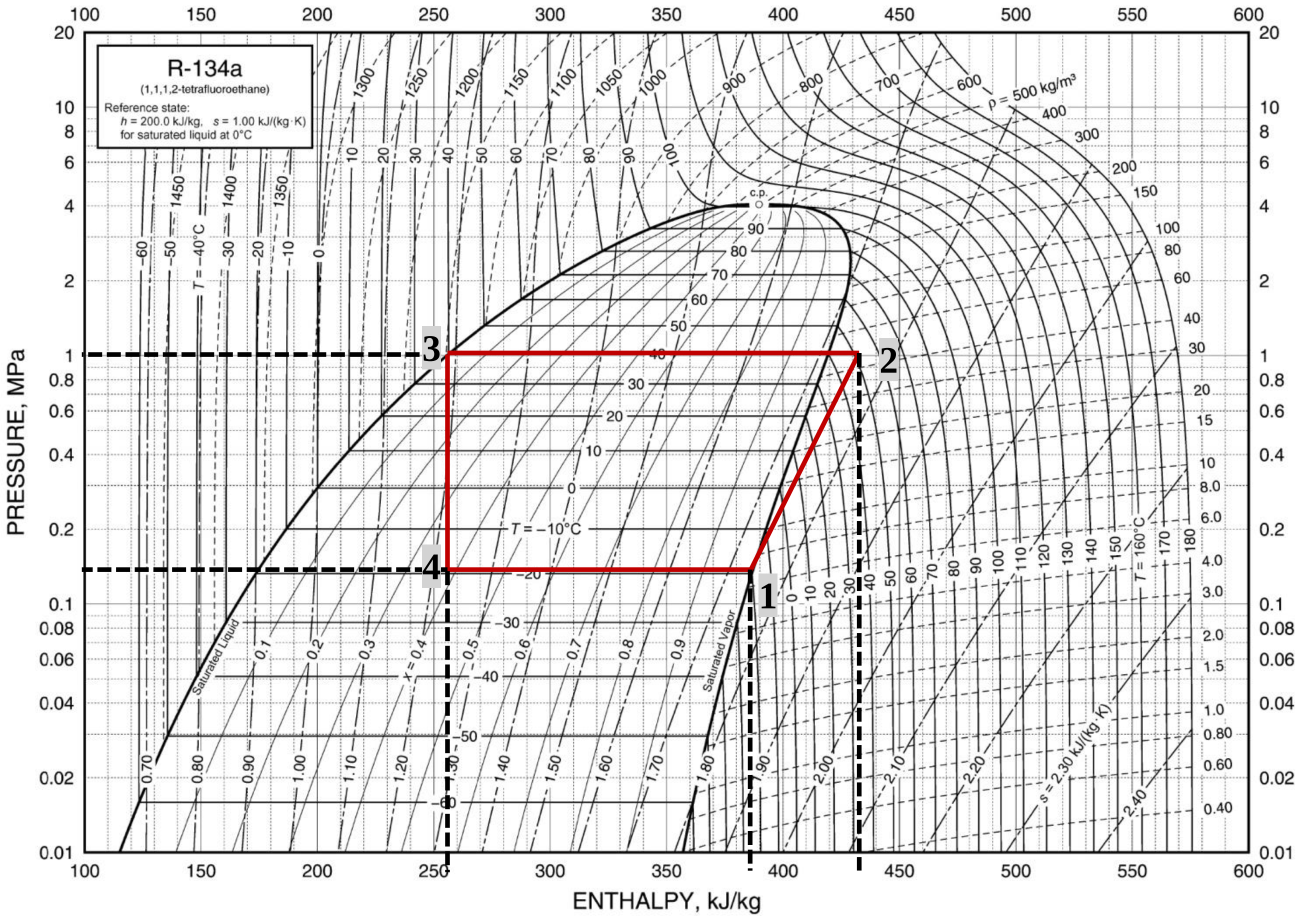
Work input to compressor ($W_{comp.}$) = $h_2 - h_1$ = 434 – 387 = 47 kJ/kg

Power input to the compressor = $\dot{m}$ ($h_2 - h_1$) = 0.05 (434 – 387) = 2.35 kW

Refrigeration effect (Q_{evap}) = $h_1 - h_4$ = 387 – 258 = 129 kJ/kg

Refrigeration effect in kW = $\dot{m}$ ($h_1 - h_4$) = 0.05 (387 – 258) = 1.45 kW

$$\boldsymbol{COP} = \frac{Q_{Evap.}}{W_{Comp.}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{129}{47} = 2.744$$



Ideal and Actual vapour compression cycle:-

The Ideal cycles considered so far are internally reversible and no change of refrigerant state takes place in the connecting pipelines. However, in actual *VCRS* several irreversibilities exist. These are due to:-

1. Pressure drops in evaporator and condenser.
2. Pressure drop across suction and discharge valves of the compressor.
3. Heat transfer in compressor.
4. Pressure drop and heat transfer in connecting pipe lines.

Figure (21) shows the actual *VCRS* cycle on P-h and T-s diagrams indicating various irreversibilities.

Process	State	Process	State
Pressure drop in evaporator.	4-1d	Pressure drop across discharge valve.	2-2a
Superheat of vapour in evaporator.	1d-1c	Pressure drop in the delivery line.	2a-2b
Useless superheat in suction line.	1c-1b	Desuperheating of vapour in delivery pipe.	2b-2c
Suction line pressure drop.	1b-1a	Pressure drop in the condenser.	2b-3
Pressure drop across suction valve.	1a-1	Subcooling of liquid refrigerant.	3-3a
Non-isentropic compression.	1-2	Heat gain in liquid line.	3a-3b

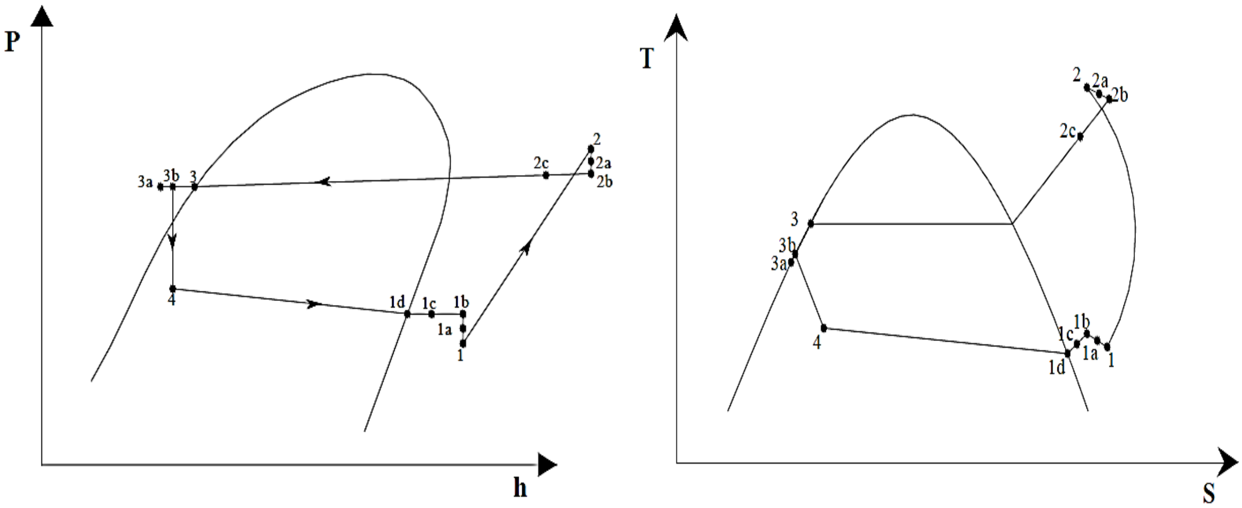


FIGURE (21) Actual *VCRS* cycle on P-h and T-s diagrams.

Factors affecting vapour compression cycle performance:-

a) The effect of evaporator and condenser temperatures on *COP* of the *VCR* cycle:-

Figure (22) shows the effect of evaporator and condenser temperatures on *COP* of the *VCR* cycle. As expected, for a given condenser temperature the *COP* increases rapidly with evaporator temperature, particularly at low condensing temperatures. For a given evaporator temperature, the *COP* decreases as condenser temperature increases. However, the effect of condenser temperature becomes marginal at low evaporator temperatures.

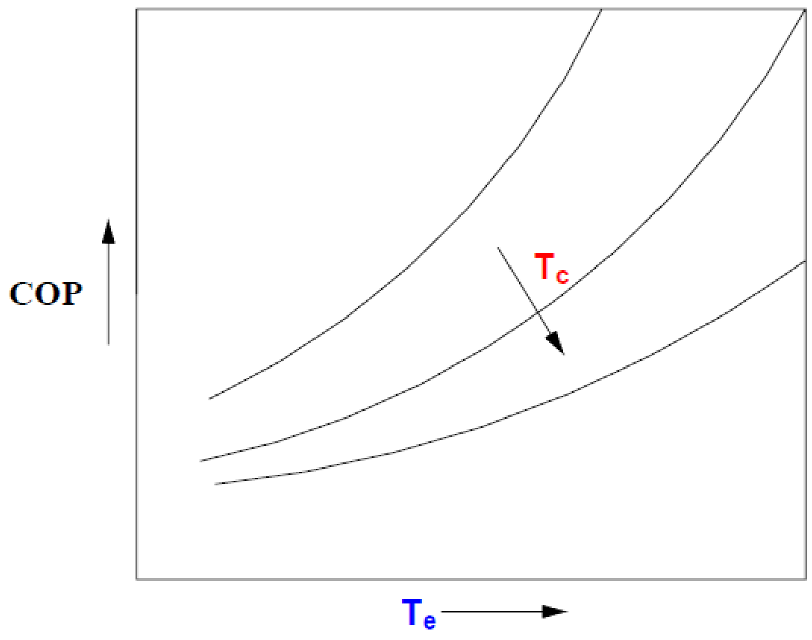


FIGURE (22) Effect of evaporator and condenser temperatures on *COP* of the *VCR* cycle.

b) The effect of pressure loss:-

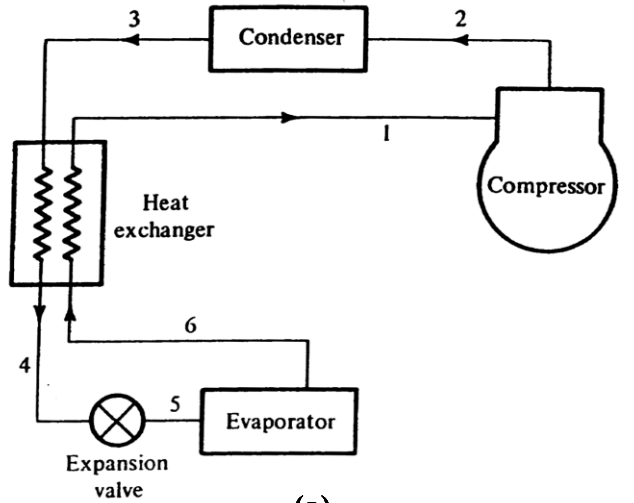
During the flow of the refrigerant through the piping, evaporator, condenser, receiver, valves and pipe lines, there is pressure drop due to internal fluid friction. Figure (21) shows a P-h and T-c representation of an actual cycle with pressure drops occurring in various components.

We can conclude that the refrigerating effect is decreased and work required is increased when there is decrease in suction pressure. The net effect is to reduce the refrigerating capacity of the system (with the same amount of refrigerant flow) and the *COP*.

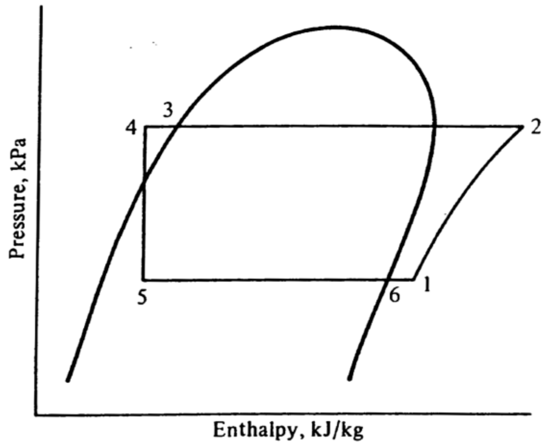
c) The effect of subcooling and superheating:-

It is possible to cool the refrigerant liquid in the condenser to a few degrees lower than the condensing temperature by adding extra area for heat transfer. In such a case, the exit condition of the condenser will be in the subcooled liquid region. Hence this process is known as ***subcooling***. the effect of subcooling is to increase the refrigerating effect. Thus sub-cooling results in increase of *COP*. Similarly, the temperature of heat source will be a few degrees higher than the evaporator temperature, hence the vapour at the exit of the evaporator can be superheated by a few degrees. If the superheating of refrigerant takes place due to heat transfer with the refrigerated space (low temperature heat source) then it is called as ***useful superheating*** as it increases the refrigeration effect but it give a low value of *COP*. On the other hand, it is possible for the refrigerant vapour to become superheated by exchanging heat with the surroundings as it flows through the connecting pipelines. Such a superheating is called as useless superheating as it does not increase refrigeration effect.

Some refrigeration systems use a ***liquid-to-suction heat exchanger***, which subcools the liquid from the condenser with suction vapour coming from the evaporator. The arrangement is shown in Figure (23-(a)) and the corresponding pressure-enthalpy diagram in Figure (23-(b)).



(a)



(b)

FIGURE (23) (a) Refrigeration system with a heat exchanger. (b) Pressure-enthalpy diagram of the system using a heat exchanger.

Saturated liquid at point 3 coming from the condenser is cooled to point 4 by means of vapour at point 6 being heated to point 1. From a heat balance,

$$h_3 - h_4 = h_1 - h_6$$

The refrigerating effect is either $h_6 - h_5$ or $h_1 - h_3$

Figure (24) shows a cutaway view of a liquid-to-suction heat exchanger.

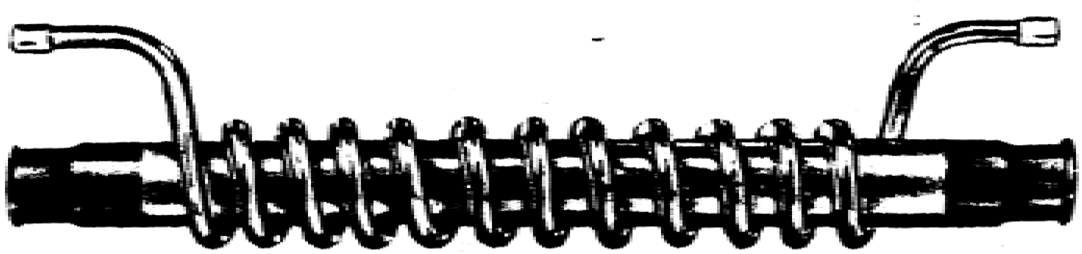


FIGURE (24) A liquid-to-suction heat exchanger.

Compared with the standard vapour-compression cycle, the system using the heat exchanger may seem to have obvious advantages because of the increased refrigerating effect. Both the capacity and the coefficient of performance may seem to be improved. This is not necessarily true, however. Even though the refrigerating effect is increased, the compression is pushed farther out into the superheat region, where the work of compression in kilojoules per kilogram is greater than it is close to the saturated-vapour line.

The heat exchanger is important because of two reasons:-

i. The vapour entering the compressor must be superheated to ensure that no liquid enters the compressor.

ii. To subcool the liquid from the condenser to prevent bubbles of vapour from impeding the flow of refrigerant through the expansion valve.

Example 25:-

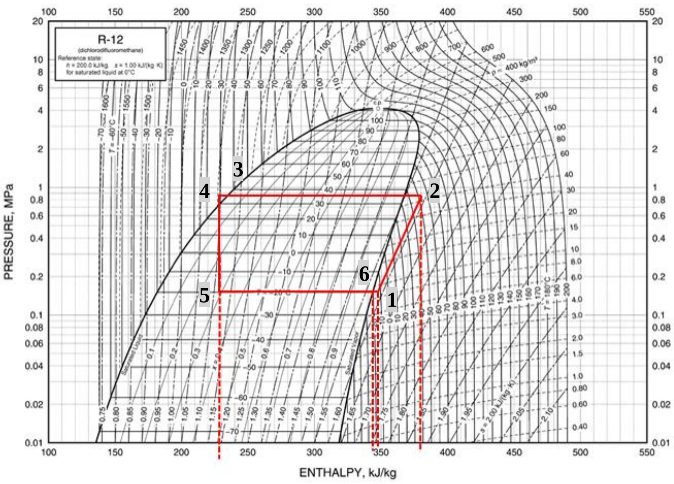
A compression refrigeration cycle used R-12 has a cooling effect of 40 kW. The condenser temperature is 35°C and the evaporator temperature is -20°C. The vapour is superheated by 6°C before it enters the compressor while it is subcooled by 5°C. Calculate the following:-

i. Mass flowrate of the refrigerant.

ii. Work done in the cycle.

iii. *COP*.

Solution:-



Pressure-Enthalpy Diagram for Refrigerant 12

From R-12 P-h diagram we find:-

h_1 = 346.5 kJ/kg , h_2 = 378 kJ/kg,

h_3 = 233.5 kJ/kg, $h_4 = h_5$ = 228.5 kJ/kg,

and h_6 = 342.6 kJ/kg.

i. $$\dot{m} = \frac{Cooling\ effect}{(h_6 - h_5)} = \frac{40}{(342.6-228.5)}$$

$$\dot{m} = \frac{40}{114.1} = 0.35\ kg/sec$$

ii. $W_{comp.} = \dot{m}\,(h_2 - h_1) = 0.35\,(378 - 346.5) = 11.025$ kW

iii. $$COP = \frac{Cooling\ effect}{W_{comp.}} = \frac{40}{11.025} = 3.63$$

Example 26:-

A refrigerant 22 vapour compression system includes a liquid-to-suction heat exchanger in the system. The heat exchanger warms saturated vapour coming from the evaporator from -10 to 5°C with liquid which comes from the condenser at 30°C. The compressions are isentropic in both cases listed below.

(a) Calculate the coefficient of performance of the system without the heat exchanger but with the condensing temperature at 30°C and an evaporating temperature of -10°C.

(b) Calculate the coefficient of performance of the system with the heat exchanger?

(c) If the compressor is capable of pumping 12.0 L/s measured at the compressor suction, what is the refrigeration capacity of the system without the heat exchanger?

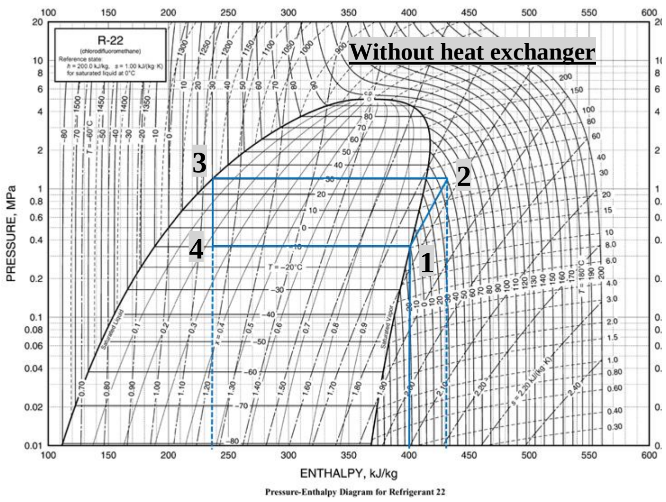
(d) With the same compressor capacity as in (c), what is the refrigerating capacity of the system with the heat exchanger?

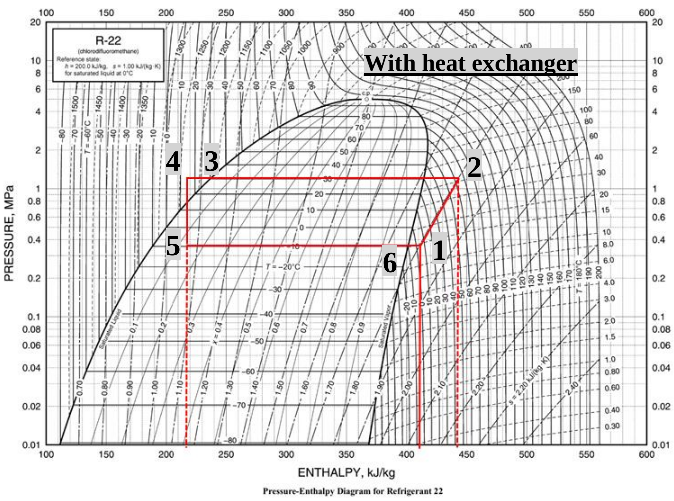
Solution:-

(a) Without heat exchanger:- From R22 P-h diagram

h_1 = 401.5 kJ/kg , h_2 = 431.5 kJ/kg , $h_3 = h_4$ = 236.6 kJ/kg

$$\boldsymbol{COP} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{401.5 - 236.6}{431.5 - 401.5} = 5.5$$





(b) With heat exchanger:- From R22 P-h diagram

h_6 = 401.5 kJ/kg, h_1 = 412 kJ/kg, h_2 = 443 kJ/kg, h_3 = 236.6 kJ/kg.

$$\boldsymbol{COP} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{412 - 236.6}{443 - 412} = 5.65$$

(c) Refrigerating capacity without heat exchanger:-

At 1, v = 65.359 L/kg

Refrigerating Capacity = (12 L/s) / (65.3399 L/kg) ╳ ($h_1 - h_4$)

= 0.184 ╳ (401.5 – 236.6) = 30.3 kW

(d) Refrigerating capacity with heat exchanger:-

At 1, v = 70.275 L/kg

Refrigerating Capacity = (12 L/s) / (70.275 L/kg) ╳ ($h_1 - h_3$)

= 0.17 ╳ (412 – 236.6) = 29.9 kW

Multipressure systems:-

A multipressure system is a refrigeration system that has two or more low-side pressures. The low-side pressure is the pressure of the refrigerant between the expansion valve and the intake of the compressor. A multipressure system is distinguished from the single-pressure system, which has but one low-side pressure. A multipressure system may be found, for example, in a dairy where one evaporator operates at -35°C to harden ice cream while another evaporator operates at 2°C to cool milk. Another typical application might be in a process industry where a two- or three-stage compression arrangement serves an evaporator operating at a low temperature of -20°C or lower.

This lecture considers only multipressure systems having two low-side pressures, but the principles developed here will apply to more than two low-side pressures. Two functions often integral to multipressure systems are the removal of flash gas and intercooling. They will be discussed first. Then several combinations of multiple evaporators and compressors will be analyzed.

(a) Removal of flash gas:-

A saving in the power requirement of a refrigeration system results if the flash gas that develops in the throttling process between the condenser and evaporator is removed and recompressed before complete expansion. When saturated liquid expands through an expansion valve, the fraction of vapour or flash gas progressively increases. The expansion process shown on the pressure-enthalpy diagram in Figure (25) takes place from 1 to 2. The state point, as the expansion proceeds, moves into a region of a greater fraction of vapour.

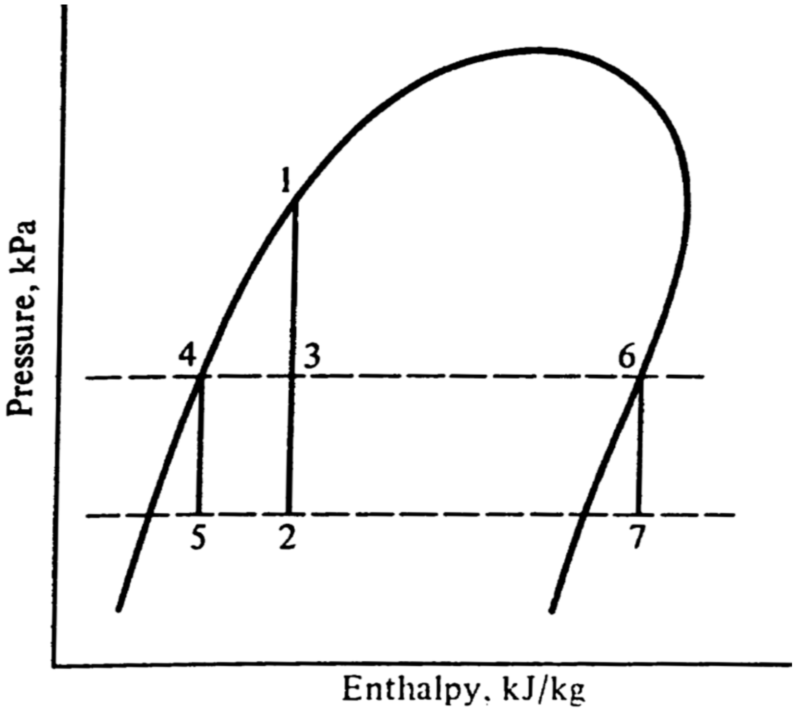


FIGURE (25) Expansion process showing replacement of process 3-2 with the combination of 4-5 and 6-7.

The end point of the expansion, 2, could have been achieved by interrupting the expansion at 3 and separating the liquid and vapour phases, which are 4 and 6, respectively. The expansion could then continue by expanding the liquid at 4 and the vapour at 6 to the final pressure, giving 5 and 7, respectively. The combination of refrigerant at states 5 and 7 gives point 2.

Inspection of the expansion from 6 to 7 confirms that it is wasteful. In the first place, the refrigerant at 7 can do no refrigerating; in the second place, work will be required to compress the vapour back to the pressure it had at 6. Why not perform part of the expansion, separate the liquid from the vapour, continue expanding the liquid, and recompress the vapour without further expansion? The equipment to achieve this separation is called a ***flash tank*** (see Figure (26)). The expansion from 1 to 3 takes place through a float valve, which serves the further purpose of maintaining a constant level in the flash tank. To recompress the vapour at 6, a compressor must be available with a suction pressure of 6. Thus two compressors are needed in the system.

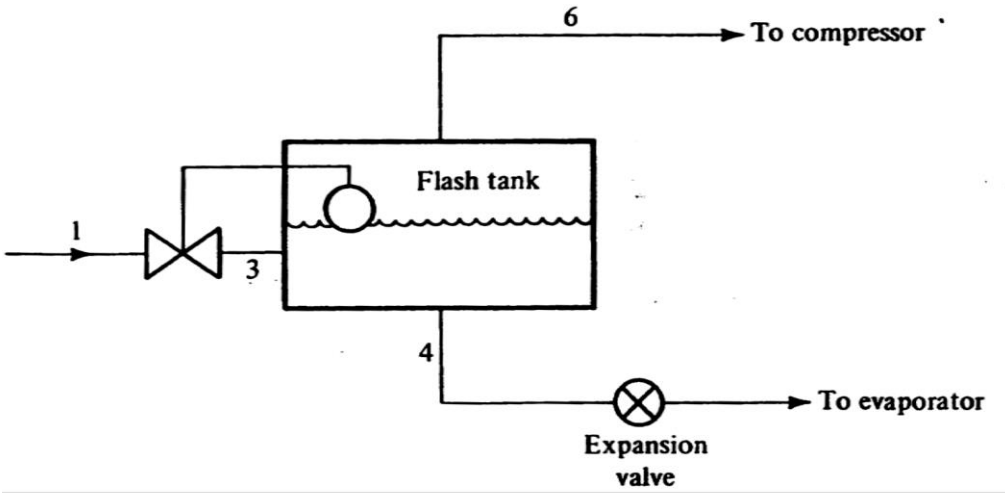


FIGURE (26) Flash tank for removing flash gas during expansion process.

The flash tank must separate liquid refrigerant from vapour. The separation occurs when the upward velocity of the vapour is low enough for the liquid particles to drop back into the tank. Normally vapour velocities less than 1 m/s will provide adequate separation. This velocity is found by dividing the volume flow of the vapour by the surface area of the liquid.

(b) Inter-cooler:-

lntercooling between two stages of compression reduces the work of compression per kilogram of vapour. In two-stage compression of air, for example, an intercooling from point 2 to 4 on the pressure-displacement diagram of Figure (27) saves some work. If the processes are reversible, the saving is represented by the shaded area in Figure (27).

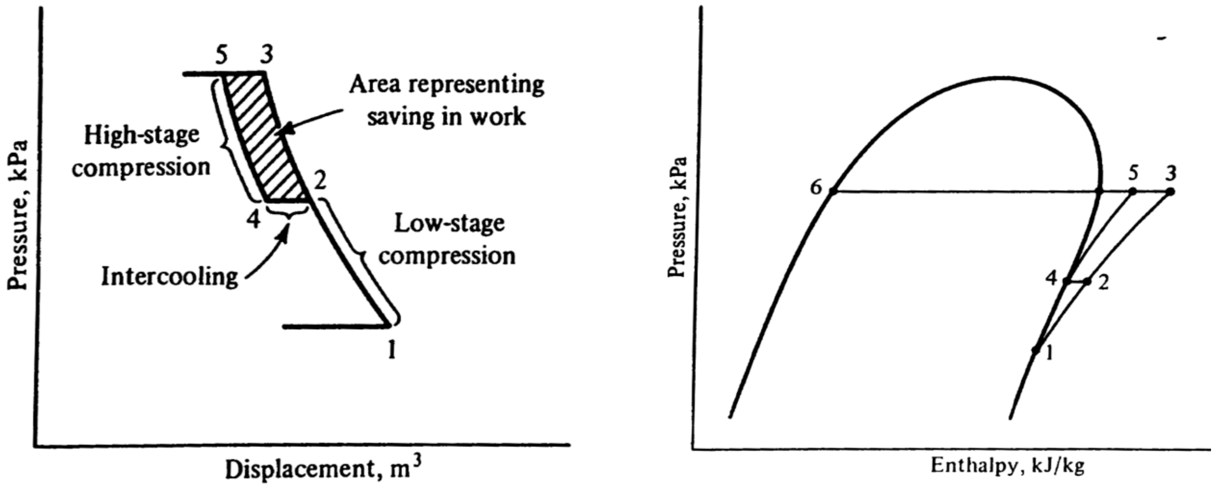


FIGURE (27) Intercooling of a refrigerant in two-stage compression..

Figure (27) shows how compression with intercooling appears on the pressure-enthalpy diagram of a refrigerant. Processes 1-2-3 and 4-5 are on lines of constant entropy, but process 2-3 falls on a flatter curve than process 4-5. Between the same two pressures, therefore, process 4-5 shows a smaller increase in enthalpy, which indicates that less work is required than in 2-3.

Where subscript 1 refers to the entrance and 2 to the exit of the compressor. Between two given pressures, the work of compression is proportional to the specific volume of entering gas. The specific volume at 2 in Figure (27) is greater than it is at 4; so the work required for compressing from 2 to 3 is greater than in compressing from 4 to 5.

Intercooling in a refrigeration system can be accomplished with a water-cooled heat exchanger or by using refrigerant. (Figure (28-a and b)). The water-cooled intercooler may be satisfactory for two-stage air compression, but for refrigerant compression the water is usually not cold enough. The alternate method of Figure (28-b) uses liquid refrigerant from the condenser to do the intercooling. Discharge gas from the low-stage compressor bubbles through the liquid in the intercooler. Refrigerant leaves the intercooler at 4 as saturated vapour.

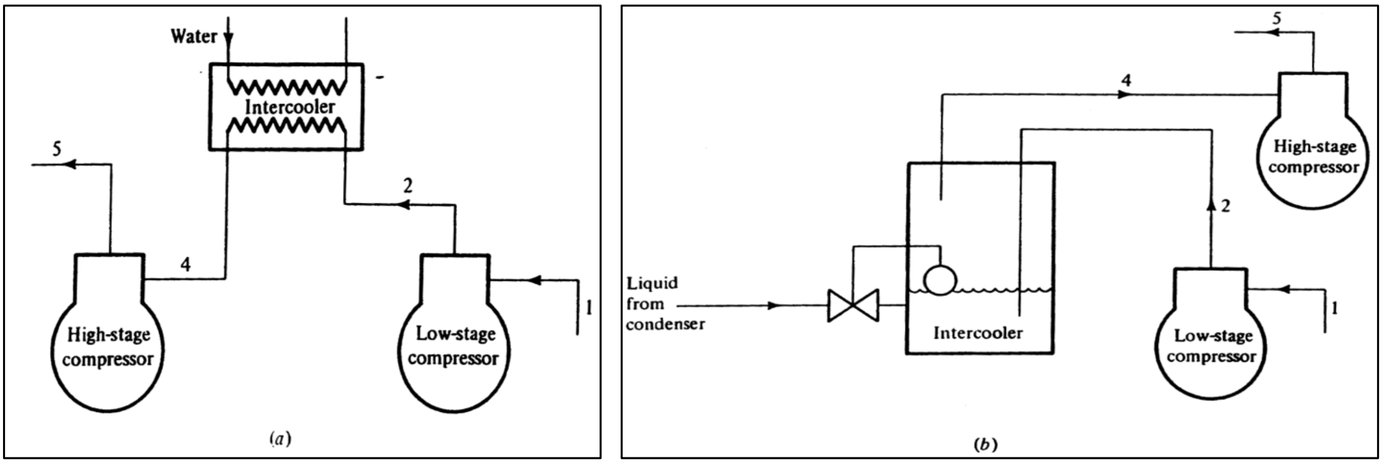


FIGURE (28) Intercooling with (a) a water-cooled heat exchanger, and (b) liquid refrigerant.

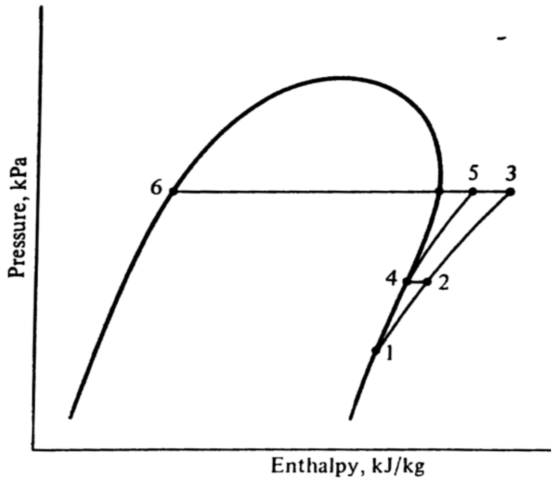
Intercooling with liquid refrigerant will usually decrease the total power requirements when ammonia is the refrigerant but not when refrigerant 12 or 22 is used.

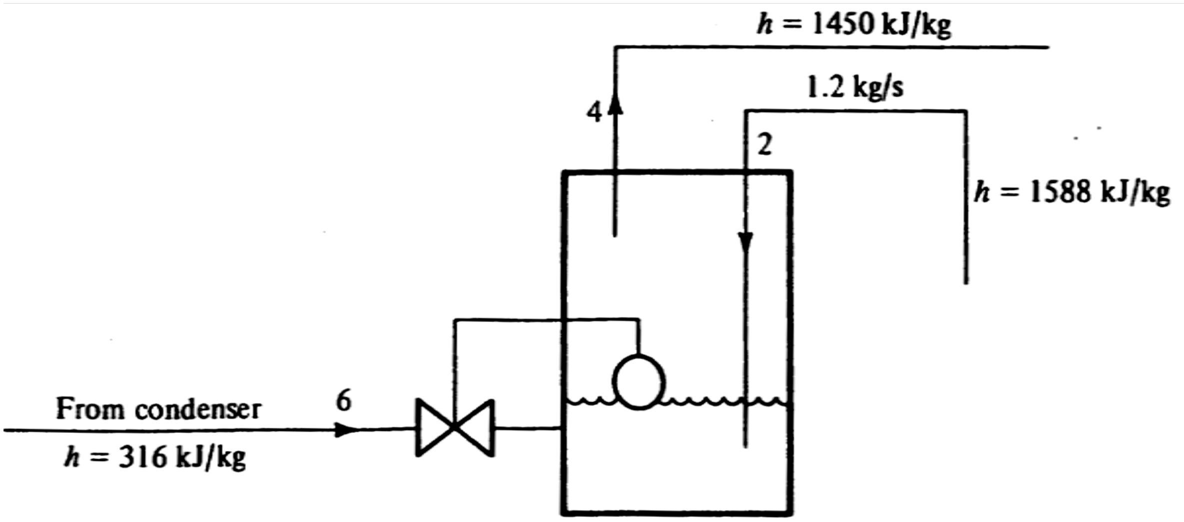
Example 27:-

Calculate the power needed to compress 1.2 kg/s of ammonia from saturated vapor at 80 kPa to 1000 kPa ***(a)*** by single-stage compression and ***(b)*** by two-stage compression with intercooling by liquid refrigerant at 300 kPa.

Solution:-

Table (2) shows the summary of the calculations with the subscripts referring to state points in side Figure. The high-stage compressor in the intercooled system must compress 1.2 kg/s plus the flow rate of refrigerant that evaporates to desuperheat the gas at 2. The flow rate of ammonia compressed in the high stage can be calculated by making a heat and a mass balance about the intercooler, as shown in below Figure.





Heat balance:-

$$\dot{m}_6(316\ \text{kJ/kg}) + (1.2\ \text{kg/s})(1588\ \text{kJ/kg}) = \dot{m}_4(1450\ \text{kJ/kg})$$

Mass balance:-

$$\dot{m}_6 + 1.2 = \dot{m}_4$$

Solving gives $\dot{m}_4 = 1.346$ kg/s

TABLE 2:- Comparison of ammonia compression with and without intercooling.

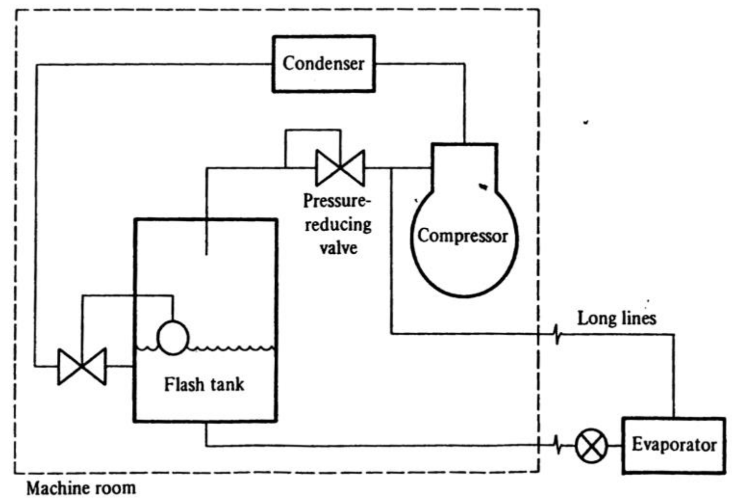
Data	Without intercooling, processes 1-2 and 2-3	With intercooling, processes 1-2, 2-4, and 4-5
h_2 - h_1 , kJ /kg	1588 - 1410	1588 -1410
$h_3 – h_2$, kJ /kg	1800 - 1588	. . . .
$h_5 – h_4$, kJ /kg	. . . .	1628 - 1450
Flow rate, kg/s, 1 to 2	1.2	1.2
2 to 3	1.2	. . . .
4 to *5*	. . . .	1.346
Power required, kW, 1 to 2	213.6	213.6
2 to 3	254.4	. . . .
4 to *5*	. . . .	239.6
Total power, kW	468.0	453.2

Intercooling the ammonia with liquid refrigerant reduced the power requirement from 468 to 453.2 kW.

(c) Single evaporator and single compressor:-

The flash tank and intercooler appear in most multipressure systems and will now be examined in various compressor evaporator combinations.

With one compressor and one evaporator the flash tank may function as shown schematically in the side Figure. A pressure-reducing valve throttles the flash gas from the intermediate pressure to the evaporator pressure. The throttling is necessary because there is no compressor available with a high suction pressure. Calculations would show that the flash tank does not improve the performance of the system. The only reason for using the flash tank would be to keep the flash gas in the machine room rather than sending it to the evaporator. The flash gas in the evaporator tubes and long suction line does no refrigeration but does increase the pressure drop. This system is used infrequently.



(d) Single compressor and two evaporators:-

In many situations one compressor serves two evaporators having different temperature requirements. An example is an industry which needs low-temperature refrigeration for a process and which must also provide air conditioning for some offices. Figure (29-a) shows one method of arranging this system, and Figure (29-b) shows the corresponding pressure-enthalpy diagram. In Figure (29-a) the air-conditioning evaporator operates at -10°C even though a higher temperature in this evaporator would cool the air sufficiently. Furthermore, difficulties may arise when an evaporator operates at an unnecessarily low temperature: an evaporator which cools air for air conditioning may collect frost, which blocks the flow of air; an evaporator which chills a liquid may freeze the liquid; and an evaporator which cools air for a room where meat or produce is stored may dehumidify the air so much that the products will be dehydrated.

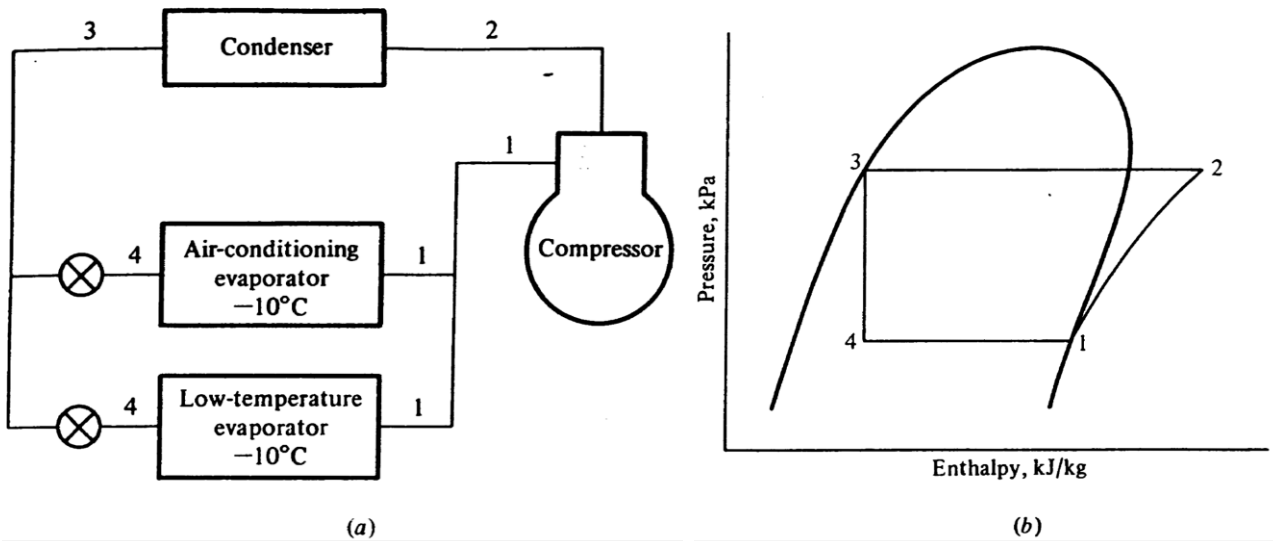
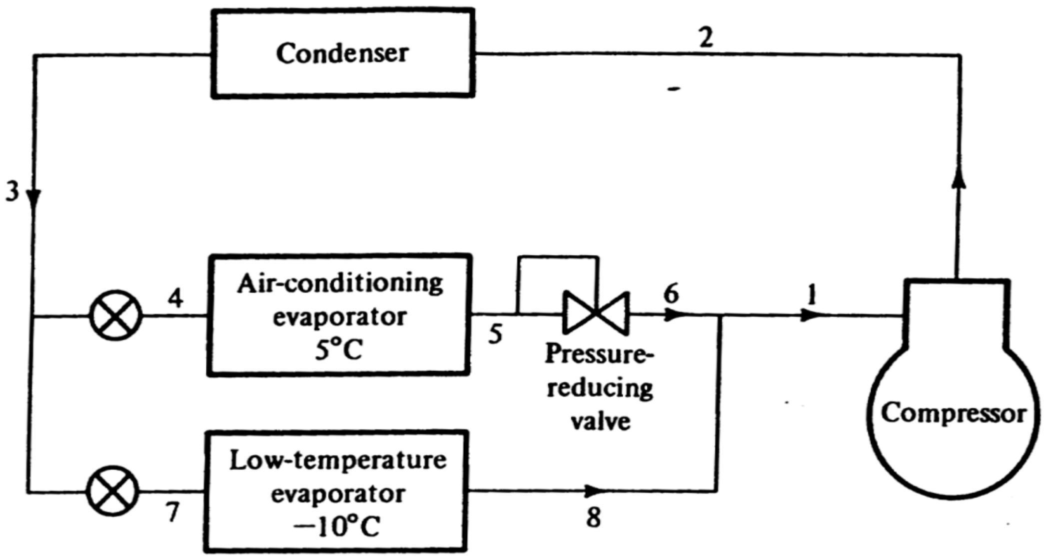


FIGURE (29) ***(a)*** One compressor and two evaporators with the air conditioning evaporator operating at -10°C. ***(b)*** Pressure-enthalpy diagram for system of *(a)*.

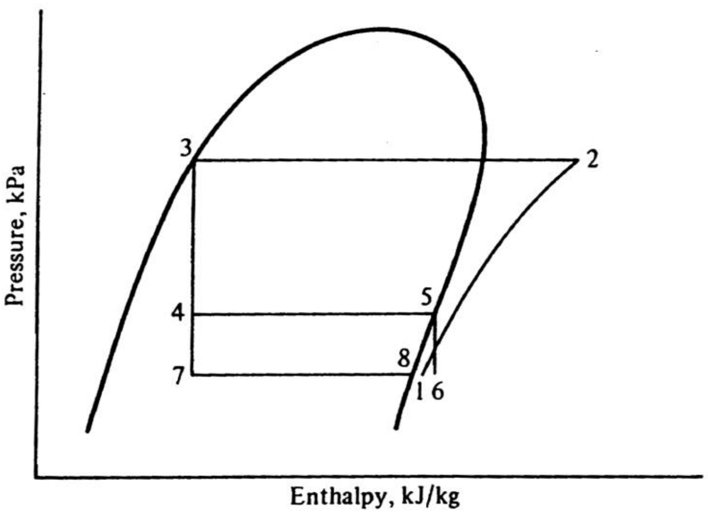
To overcome the drawbacks of the system in Figure (29-a), a revision may be made as shown in Figure (30-a). A pressure-reducing valve installed after the high-temperature evaporator regulates the pressure and maintains a temperature in the air-conditioning evaporator of 5°C, for example. Figure (30-b) shows the corresponding pressure enthalpy diagram. Differences in performance between the systems in Figures (29-a) and (30-a) are as follows. In the system of Figure

(30-a), the refrigerating effect in the high-temperature evaporator is greater than it is in the system of Figure (29-a). This is an advantage for the system of Figure (30-a). To counterbalance this advantage, the compression in Figure (30-b) occurs farther out in the superheat region than in Figure (29-b). The system of Figure (30-a) therefore demands more work per kilogram of refrigerant.

From a power standpoint, the systems are practically a standoff, but for proper operation of the high-temperature evaporator the system of Figure (30-a) is preferred.



(a)



(b)

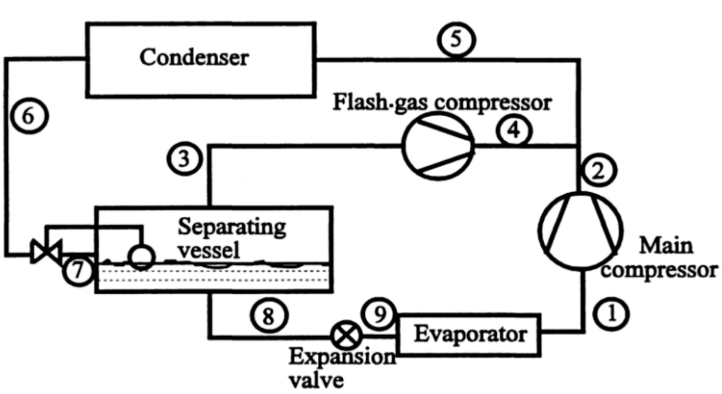
FIGURE (30) ***(a)*** One compressor and two evaporators with a pressure-reducing valve to maintain a high temperature in the air-conditioning evaporator. ***(b)*** Pressure-enthalpy diagram for system of (a).

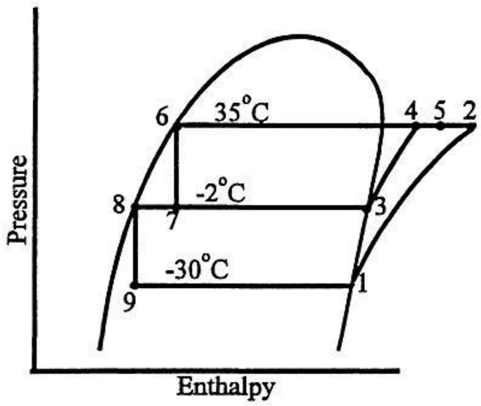
(e) Two compressors and single evaporator:-

Two-stage compression with intercooling and removal of flash gas is often the ideal way to serve one low-temperature evaporator. This system requires less power than with a single compressor, and often the saving in power will justify the cost of the extra equipment.

Example 28:-

An ammonia system operating with an evaporating temperature of -30°C and a condensing temperature of 35°C separates flash gas at a temperature of -2°C and delivers it to the condenser through a separate compressor. If the refrigerating capacity is 200 kW, what are the power requirements if the system operates ***(a)*** single stage, and ***(b)*** with flash-gas removal?





Solution:-

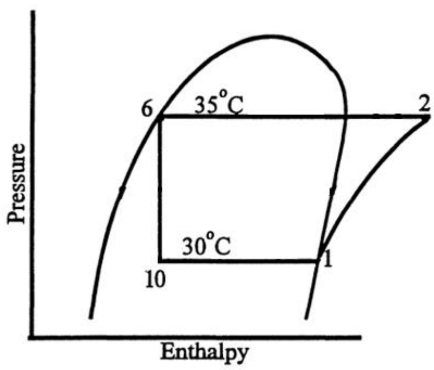
From R717 P-h diagram:-

h_1 = 1423.6 kJ/kg ; h_2 = 1796.9 kJ/kg ; h_3 = 1459.7 kJ/kg ; h_4 = 1633.5 kJ/kg ;
h_6 = 366.4 kJ/kg ; h_7 = 366.4 kJ/kg ; h_8 = 190.7 kJ/kg ; h_9 = 190.7 kJ/kg

(a) Single-stage:- In single-stage compression the enthalpies are as follows:-

h_1 = 1423.6 kJ/kg ; h_2 = 1796.9 kJ/kg ;

$h_6 = h_{10}$ = 366.4 kJ/kg ; Q_e =200kW



The mass rate of flow through the system is:-

$$\dot{m} = \frac{Q_e}{h_1 - h_{10}} = \frac{200}{1423.6 - 366.4} = 0.189 kg/s$$

and the power requirement is:-

$$\dot{W} = \dot{m}(h_2 - h_1) = (0.189kg/s)(1796.9 - 1423.6)kJ/kg = 70.55kW$$

***(b)** With flash-gas removal:-*

The mass rate of flow through the evaporator, $\dot{m}_1$, is:

$$\dot{m}_1 = \frac{Q_e}{h_1 - h_9} = \frac{200}{1423.6 - 190.7} = 0.162kg/s$$

and $\dot{m}_8 = \dot{m}_9 = \dot{m}_1 = \dot{m}_2 = 0.162kg/s$

The mass rate of flow of flash gas, $\dot{m}_3$, can be calculated from a mass and energy balance about the separating vessel:-

$$\dot{m}_7 = \dot{m}_3 + \dot{m}_8 \Rightarrow \dot{m}_7 = \dot{m}_3 + 0.162kg/s$$

and $\dot{m}_7 h_7 = \dot{m}_3 h_3 + \dot{m}_8 h_8$

$$\Rightarrow (\dot{m}_3 + 0.162)366.4 = \dot{m}_3 1459.7 + (0.162)(190.7)$$

$$\dot{m}_3 = 0.026kg/s$$

The power requirements of the compressors are:-

Main compressor:-

$$\dot{W}_1 = \dot{m}_1(h_2 - h_1) = (0.162kg/s)(1796.9 - 1423.6)kJ/kg = 60.47kW$$

Flash-gas compressor:-

$$\dot{W}_2 = \dot{m}_3(h_4 - h_3) = (0.026kg/s)(1633.5 - 1459.7)kJ/kg = 4.52kW$$

The total compressor power = $\dot{W}_1 + \dot{W}_2 = 60.47 + 4.52 = 64.99kW$

The percentage saving resulting from flash-gas removal is:-

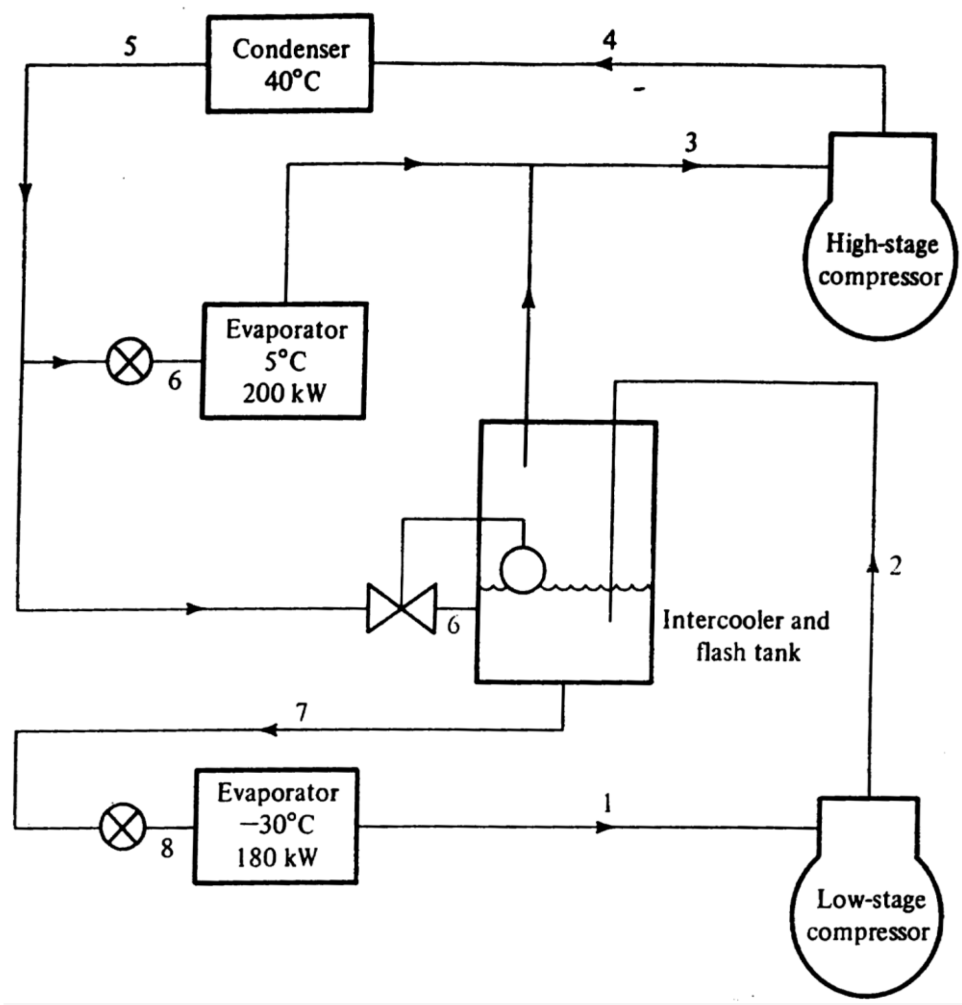
$$= \left(\frac{70.55 - 64.99}{70.55}\right) \times 100\% = 7.9\%$$

(f) Two compressors and two evaporators:-

The system which has two evaporators operating at different temperatures is common in industrial refrigeration. A dairy cooling milk and manufacturing ice cream has been mentioned. A frozen-food plant may require two evaporators at different temperatures, one at -40°C to quick-freeze the food and the other at -25°C to hold the food after it is frozen. Process and chemical industries often require different temperatures of refrigeration in various sections of the plant. Evaporators at two different temperatures can be handled efficiently by a two-stage system which employs intercooling and removal of flash gas.

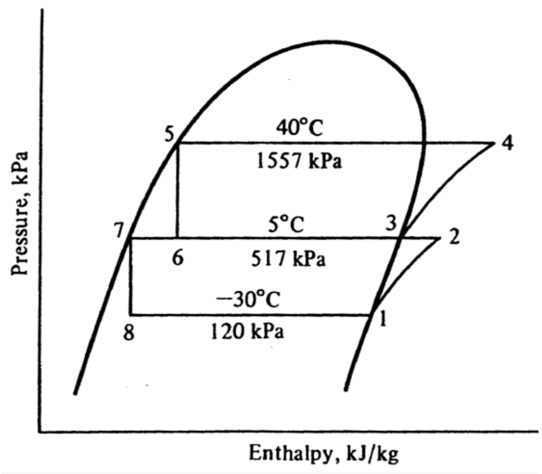
Example 29:-

In an ammonia system one evaporator is to provide 180 kW of refrigeration at -30°C and another evaporator is to provide 200 kW at 5°C. The system uses two-stage compression with intercooling and is arranged as in Figure below. The condensing temperature is 40°C. Calculate the power required by the compressors.



Solution:-

Sketch the pressure-enthalpy diagram of the cycle as in the side Figure. The discharge pressure of the low-stage compressor and the suction pressure of the high-stage compressor are the same as the pressure in the 5°C evaporator. Next determine the enthalpies at the state points.



$h_1 = h_g$ at -30°C = 1423 kJ/kg

$h_2 = h$ at 517 kPa after isentropic compression = 1630 kJ/kg

$h_3 = h_g$ at 5°C = 1467 kJ/kg

$h_4 = h$ at 1557 kPa after isentropic compression = 1625 kJ/kg

$h_5 = h_f$ at 40°C = 390.6 kJ/kg $h_6 = h_5$ = 390.6 kJ/kg

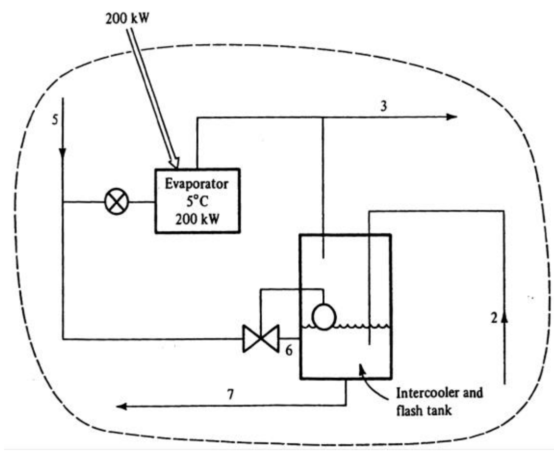
$h_7 = h_f$ at 5°C = 223 kJ/kg $h_8 = h_7$ = 223 kJ/kg.

The mass rates of flow are:-

$$\dot{m}_1 = \frac{Q_{e1}}{h_1 - h_8} = \frac{180\ kW}{1423 - 223} = 0.15\ kg/sec$$

$\dot{m}_7 = \dot{m}_8 = \dot{m}_2 = \dot{m}_1$ = 0.15 kg/sec

Probably the simplest way to calculate the mass rate of flow handled by the high-stage compressor is to make a heat and mass balance about both the high temperature evaporator and the intercooler, as shown in Figure below.



Heat balance:-

$\dot{m}_5 h_5$ + 200 kW + $\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$

Mass balance:-

$\dot{m}_2 = \dot{m}_7$ = 0.150 kg/s

Therefore $\dot{m}_5 = \dot{m}_3$

Combining gives

390.6 $\dot{m}_3$ + 200 + 0.15(1630) = 1467 $\dot{m}_3$ + 0.15(223)

Solving leads to:- $\dot{m}_3 = 0.382$ kg/sec

The power required by the compressors can now be calculated:-

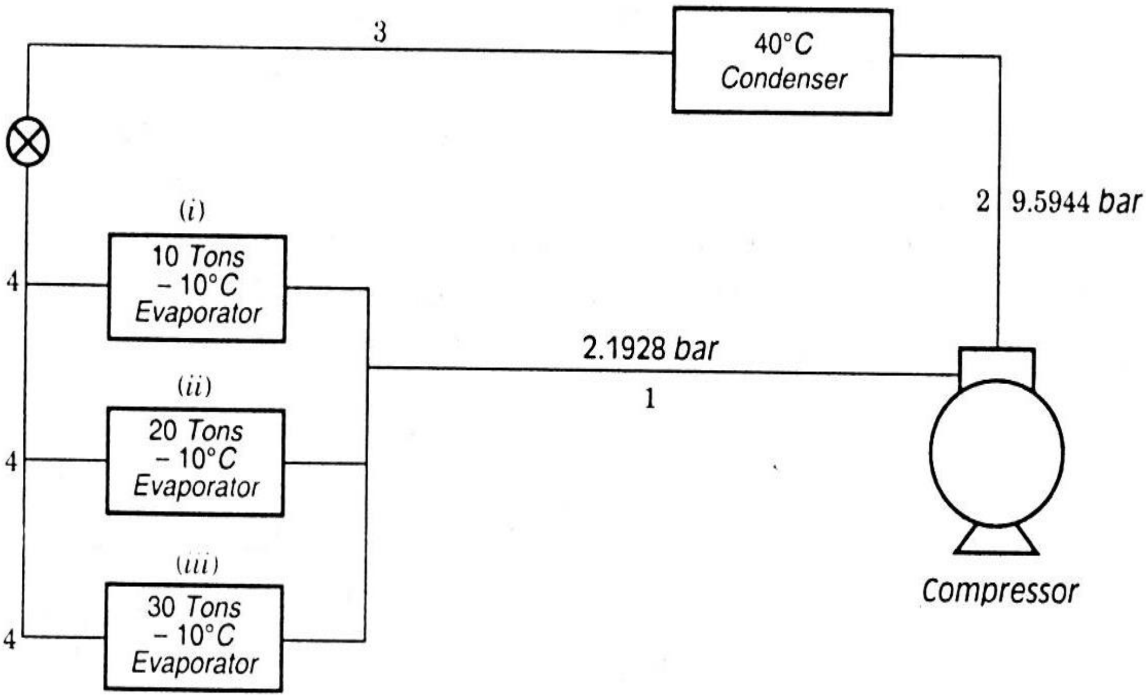
Low-stage power:- 0.15(1630 - 1423) = 31.1 kW

High-stage power:- 0.382(1625 - 1467) = 60.4 kW

Total = 91.5 kW

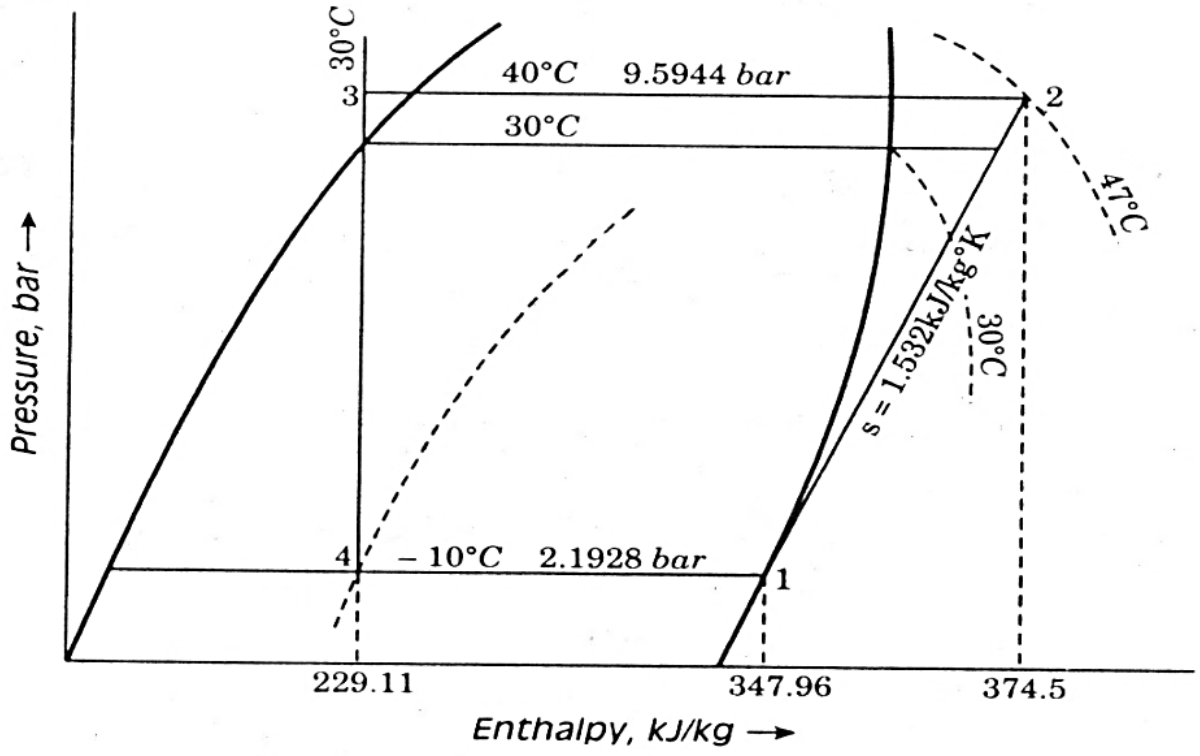
Example 30:-

A single compressor system using Freon-12 has three evaporators of 10 ton, 20 ton and 30 ton capacity, all operating at same temperature of -10°C. The condenser pressure is 9.5944 bar (40°C) and the liquid is subcooled in the condenser by 10°C. The discharge from the evaporator is by saturated and compression is to be assumed isentropic. Determine ***(i)*** the refrigerating effect in kJ/kg of flow rate, ***(ii)*** rate of refrigerant in kg/min, and ***(iii)*** the theoretical power required.



Solution:-

The flow diagram and the P-h representation is shown in the side Figure. The values obtained from P-h diagram have been labeled on chart representation.



(i) Refrigeration effect = ($h_1 - h_4$) = 347.96 – 229.11 = 118.85 kJ/kg

(ii) Mass flow rate in 60 ton evaporator, 10 + 20 + 30 = 60 ton

$\dot{m}$ = (60 X 210) / ($h_1 - h_4$) = (60 X 210) / 118.85 = 106 kg/min

(iii) Theoretical power = [Flow rate X ($h_2 - h_1$)] / 60

= [106 X (374.5 – 347.96)] / 60 = 46.88 kW

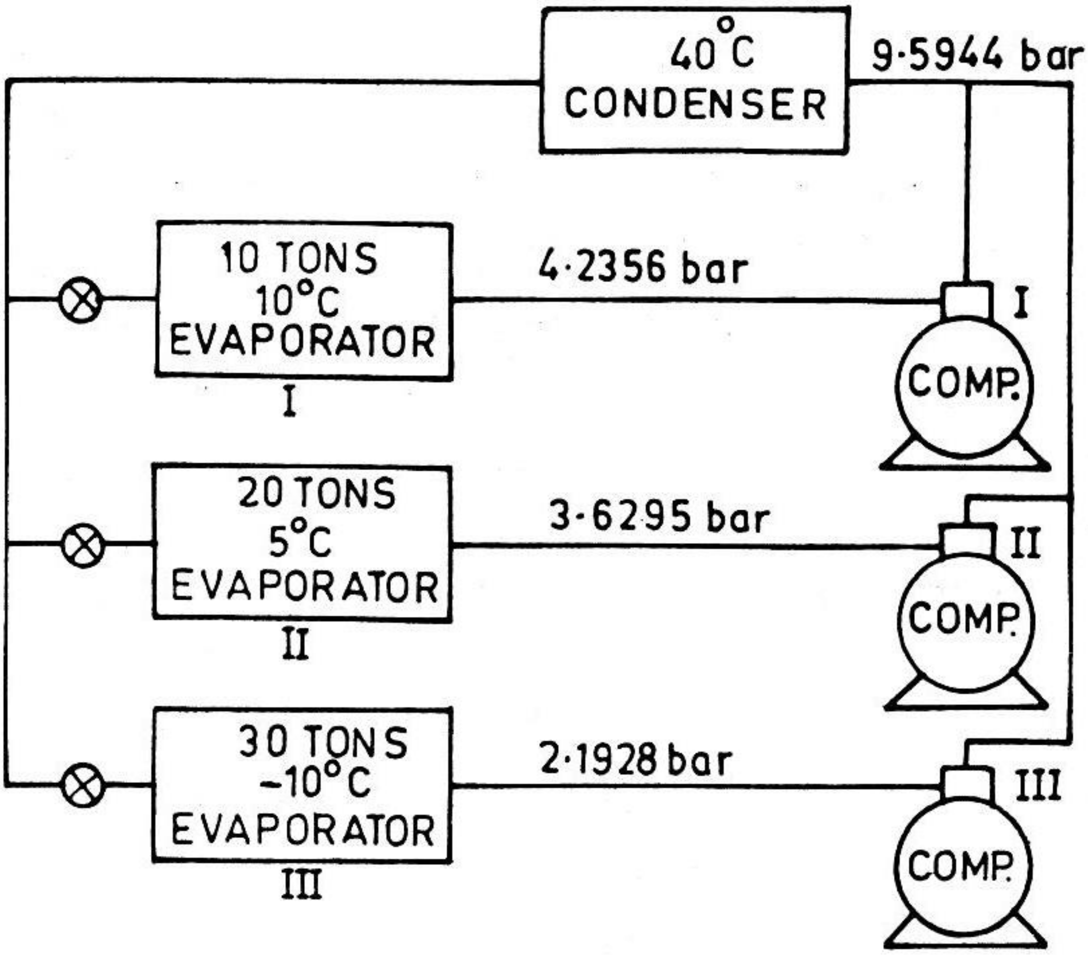
Example 31:-

A refrigeration plant comprises three evaporators 10 ton at 10°C, 20 ton at 5°C and 30 ton at -10°C with individual expansion valves and individual compressors but one condenser operating at 40°C and subcooling the liquid to 30°C. All the evaporators discharge dry saturated refrigerant R-12 to the compressor. Compression in each compressor may be assumed isentropic. Determine:-

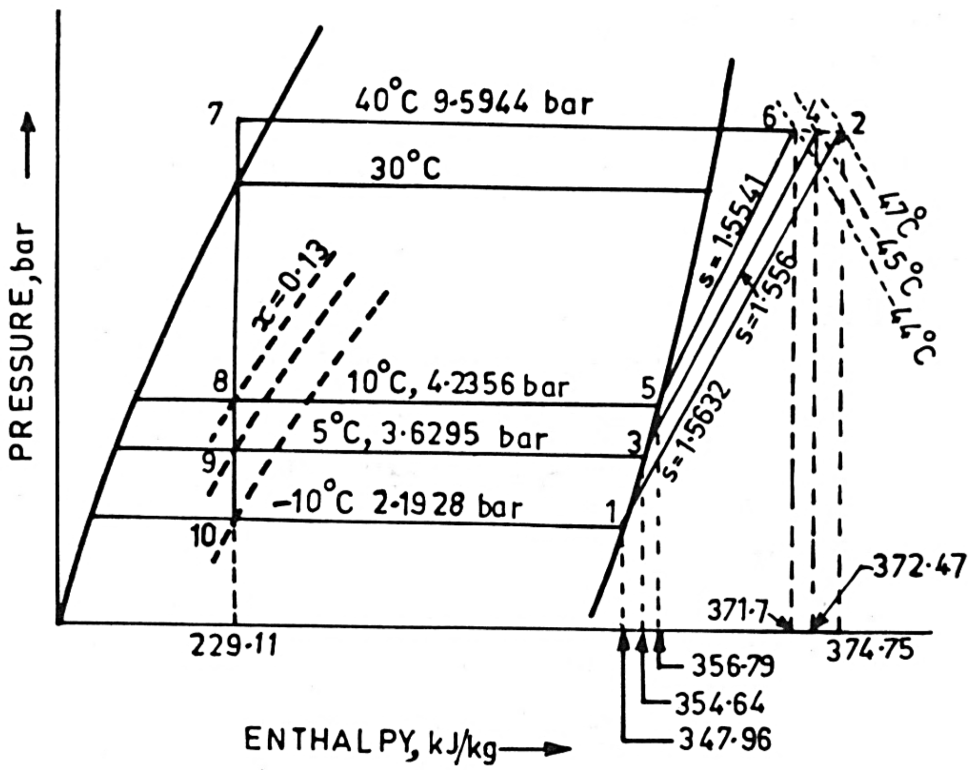
i) Refrigeration effect in each evaporator.

ii) Mass flow rate in each evaporator.

iii) Compressor power in kW.


40°C
CONDENSER
9·5944 bar
10 TONS
10°C
EVAPORATOR
I
4·2356 bar
I
COMP.
20 TONS
5°C
EVAPORATOR
II
3·6295 bar
II
COMP.
30 TONS
−10°C
EVAPORATOR
III
2·1928 bar
III
COMP.

Solution:-


PRESSURE, bar
40°C 9·5944 bar
30°C
x=0·13
10°C, 4·2356 bar
5°C, 3·6295 bar
−10°C 2·1928 bar
s=1·5541
s=1·556
s=1·5632
47°C
45°C
44°C
7
8
9
10
1
2
3
4
5
6
229·11
371·7
372·47
374·75
356·79
354·64
347·96
ENTHALPY, kJ/kg

Evaporator III:-

Refrigerating effect = $(h_1 - h_{10})$ = (347.96 – 229.11) = 118.85 kJ/kg

Mass flow rate ($\dot{m}_3$) = (Ton capacity X 210) / $(h_1 - h_{10})$

$\dot{m}_3$ = 30 X 210 / 118.85 = 53 kg/min

Power of compressor (P_{III}) = $\dot{m}_3$ X $(h_2 - h_1)$ / 60

P_{III} = 53 X (374.75 – 347.96) / 60 = 23.66 kW

Evaporator II:-

Refrigerating effect = $(h_3 - h_9)$ = (354.64 – 229.11) = 125.53 kJ/kg

Mass flow rate ($\dot{m}_2$) = (Ton capacity X 210) / $(h_2 - h_9)$

$\dot{m}_2$ = 20 X 210 / 125.53 = 33.45 kg/min

Power of compressor (P_{II}) = $\dot{m}_2$ X $(h_4 - h_3)$ / 60

P_{II} = 33.45 X (372.47 – 354.64) / 60 = 9.94 kW

Evaporator I:-

Refrigerating effect = $(h_5 - h_8)$ = (356.79 – 229.11) = 127.68 kJ/kg

Mass flow rate ($\dot{m}_1$) = (Ton capacity X 210) / $(h_5 - h_8)$

$\dot{m}_1$ = 10 X 210 / 127.68 = 16.45 kg/min

Power of compressor (P_I) = $\dot{m}_1$ X $(h_6 - h_5)$ / 60

P_I = 16.45 X (371.7 – 356.79) / 60 = 4.088 kW

Total power of the compressors (P) = $P_I + P_{II} + P_{II}$

P = 4.088 + 9.94 + 23.66 = 37.688 kW

Sheet No. (2)

Q1) A machine working on a Carnot cycle operates between -13°C and 32°C. Determine the C.O.P. when it is operated as (1) a refrigerating machine. (2) a heat pump.

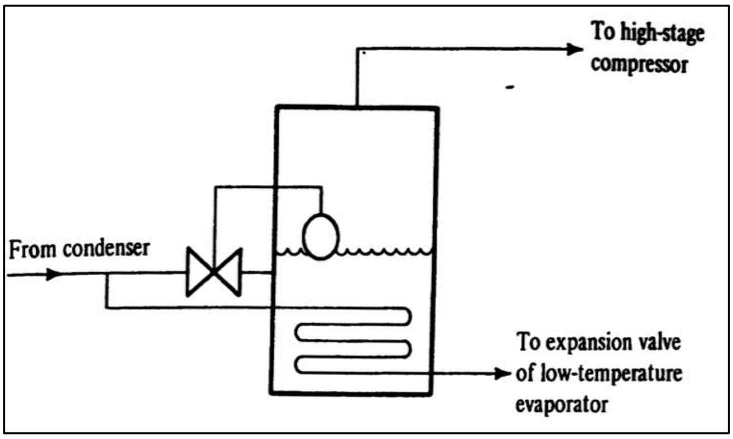
Q2) A Carnot refrigeration cycle absorbs heat at -12°C and rejects it at 40°C. **(a)** Calculate the coefficient of performance of this refrigeration cycle. **(b)** If the cycle is absorbing 15 kW at the -12°C temperature, how much power is required? **(c)** If a Carnot heat pump operates between the same temperatures as the above refrigeration cycle, what is the performance factor? **(d)** What is the rate of heat rejection at the 40°C temperature if the heat pump absorbs 15 kW at the -12°C temperature?

(***Ans. (a)*** *5.02*, ***(b)*** *3kW*, ***(d)*** *18kW*).

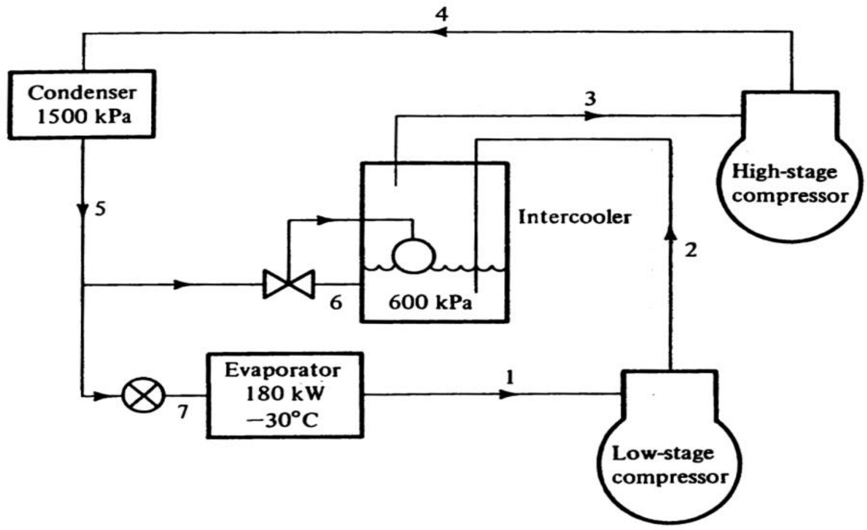
Q3) If in a standard vapor-compression cycle using refrigerant 22 the evaporating temperature is -5°C and the condensing temperature is 30°C, sketch the cycle on pressure-enthalpy coordinates and calculate **(a)** the work of compression, **(b)** the refrigerating effect, and **(c)** the heat rejected in the condenser, all in kilojoules per kilogram, and **(d)** the coefficient of performance. (***Ans. (d)*** *6.43*).

Q4) A refrigeration system using refrigerant 22 is to have a refrigerating capacity of 80 kW. The cycle is a standard vapor-compression cycle in which the evaporating temperature is -8°C and the condensing temperature 42°C. **(a)** Determine the volume flow of refrigerant measured in cubic meters per second at the inlet to the compressor. **(b)** Calculate the power required by the compressor. (***Ans. (a)*** *0.0325 m^3/s and* ***(b)*** *19.442 kW*).

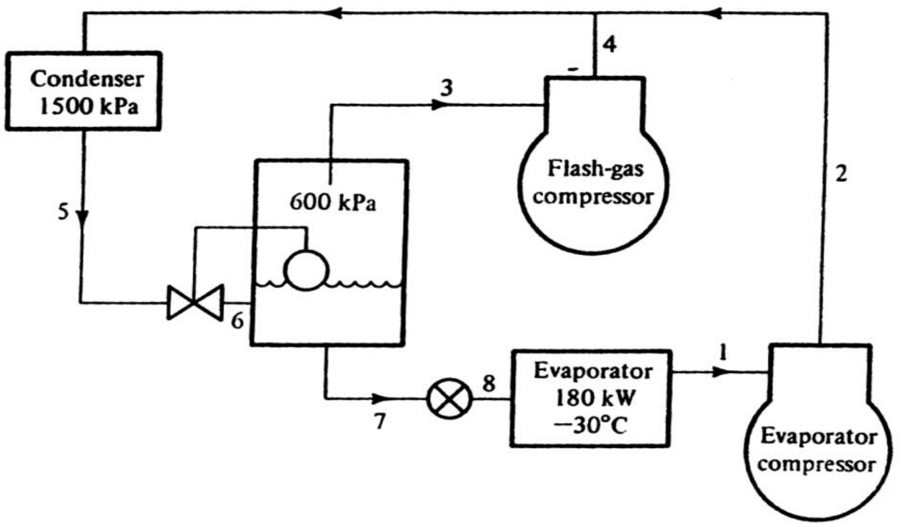
Q5) A liquid subcooler as shown in the Figure receives liquid ammonia at 30°C and subcools 0.6 kg/s to 5°C. Saturated vapor leaves the subcooler for the high-stage compressor at -1°C. Calculate the flow rate of ammonia that evaporates to cool the liquid. (***Ans.*** 0.0575 kg/s).



Q6) In a refrigerant 22 refrigeration system the capacity is 180 kW at a temperature of -30°C. The vapor from the evaporator is pumped by one compressor to the condensing pressure of 1500 kPa. Later the system is revised to a two-stage compression operating on the cycle shown in the Figure with intercooling but no removal of flash gas at 600 kPa. **(a)** Calculate the power required by the single compressor in the original system. **(b)** Calculate the total power required by the two compressors in the revised system. (***Ans. (a)*** *71.23 kW and* ***(b)*** *70.73 kW*).

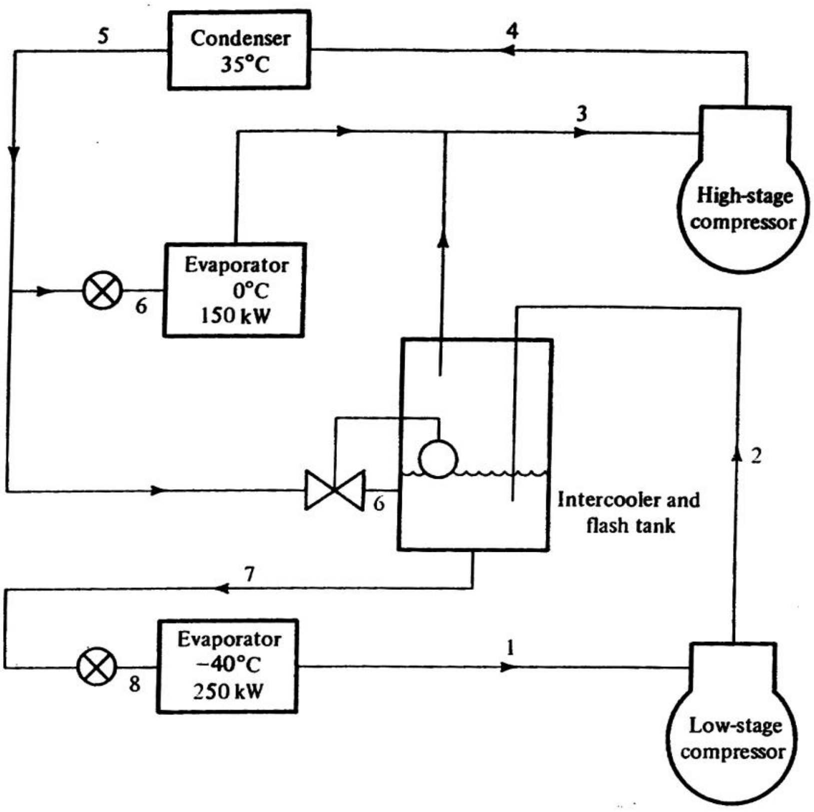


Q7) A refrigerant 22 system has a capacity of 180 kW at an evaporating temperature of -30°C when the condensing pressure is 1500 kPa. **(a)** Compute the power requirement for a system with a single compressor. **(b)** Compute the total power required by the two compressors in the system shown in the Figure where there is no intercooling but there is flash-gas removal at 600 kPa. (***Ans. (b)*** *61.06 kW*).



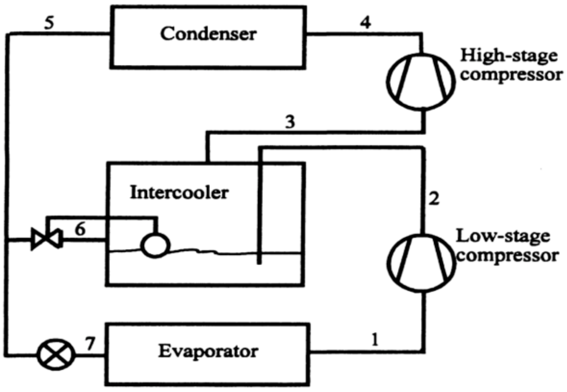
Q8) **A** two-stage ammonia system using flash-gas removal and intercooling operates on the cycle shown in the Figure. The condensing temperature is 35°C. The saturation temperature of the intermediate-temperature evaporator is 0°C, and its capacity is 150 kW. The saturation temperature of the low-temperature evaporator is -40°C, and its capacity is 250 kW. What is the rate of refrigerant compressed by the high-stage compressor?

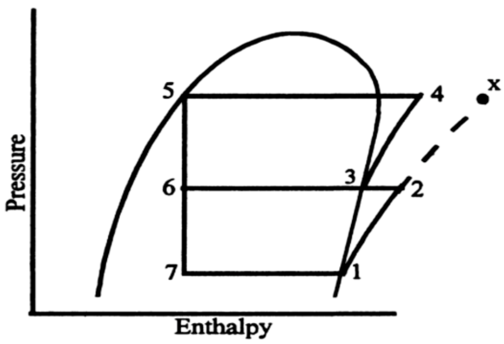
(***Ans.*** *0.411 kg/s*).



Q9) The intercooling system shown in Figure below operates with ammonia at the following saturation temperatures:- evaporating, -35°C; intermediate, 0°C; and condensing, 35°C. What is the saving in power of the intercooled cycle, expressed in percent, compared to single-stage operation?

(***Ans.*** *5.7%*).





REFRIGERANTS

Refrigerants:- A refrigerant is the **primary** working fluid used for absorbing and transmitting heat in a refrigeration system. Refrigerants absorb heat at a low temperature and low pressure and release heat at a higher temperature and pressure. Most refrigerants undergo phase changes during heat absorption (evaporation) and heat releasing (condensation).

Cooling Media:- A cooling medium is the working fluid cooled by the refrigerant to transport the cooling effect between a central plant and remote cooling units and terminals. In a large, centralized system, it is often more economical to use a coolant medium that can be pumped to remote locations where cooling is required. Chilled water, brine, and glycol are used as cooling media in many refrigeration systems. The cooling medium is often called a **secondary refrigerant**, because it obviates extensive circulation of the primary refrigerant.

The thermodynamic efficiency of a refrigeration system depends mainly on its operating temperatures. However, important practical issues such as the system design, size, initial and operating costs, safety, reliability, and serviceability etc. depend very much on the type of refrigerant selected for a given application. Due to several environmental issues such as ozone layer depletion and global warming and their relation to the various refrigerants used, the selection of suitable refrigerant has become one of the most important issues in recent times. Replacement of an existing refrigerant by a completely new refrigerant, for whatever reason, is an expensive proposition as it may call for several changes in the design and manufacturing of refrigeration systems. Hence it is very important to understand the issues related to the selection and use of refrigerants. In principle, any fluid can be used as a refrigerant. Air used in an air cycle refrigeration system can also be considered as a refrigerant. However, in this lecture the attention is mainly focused on those fluids that can be used as refrigerants in vapour compression refrigeration systems only.

In the early days only four refrigerants, Air, ammonia (NH_3), Carbon dioxide (CO_2), Sulphur dioxide (SO_2), possessing chemical, physical and thermodynamic properties permitting their efficient application and service in the practical design of refrigeration equipment were used. All the refrigerants change from liquid state to vapour state during the process.

The primary refrigerants are grouped as follows:-

(*i*) Halocarbon compounds. In this group are included refrigerants which contain one or more of three halogens, chlorine and bromine and they are sold in the market under the names as *Freon, Genetron, Isotron,* and *Areton*. Since the refrigerants belonging to this group have outstanding merits over the other group's refrigerants, therefore they find wide field of application in domestic, commercial and industrial purposes.

The list of the halocarbon-refrigerants commonly used is given below:-

R-10 — Carbon tetrachloride (CCl_4)

R-11 — Trichloro-monofluoro methane (CCl_3F)

R-12 — Dichloro-difluoro methane (CCl_2F_2)

R-13 — Mono-bromotrifluoro methane ($CBrF_3$)

R-21 — Dichloro monofluoro methane ($CHCl_2F$)

R-22 — Mono chloro difluoro methane ($CHClF_2$)

R-30 — Methylene-chloride (CH_2Cl_2)

R-40 — Methyle chloride (CH_3Cl)

R-41 — Methyle fluoride (CH_3F)

R-100— Ethyl chloride (C_2H_5Cl)

R-113— Trichloro trifluoroethane ($C_2F_3Cl_3$)

R-114— Tetra-fluoro dichloroethane ($Cl_2F_4Cl_2$)

R-152— Difluoro-ethane ($C_2H_6F_2$)

(*ii*) Azeotropes. The refrigerants belonging to this group consists of mixtures of different substances. These substances cannot be separated into components by distillations. They possess fixed thermodynamic properties and do not undergo any separation with changes in temperature and pressure. An azeotrope behaves like a simple substance.

Example. R-500. It contains 73.8% of (R-12) and 26.2% of (R-152).

(*iii*) Hydrocarbons. Most of the refrigerants of this group are organic compounds. Several hydrocarbons are used successfully in commercial and industrial installations. Most of them possess satisfactory thermodynamic properties but are highly inflammable. Some of the important refrigerants of this group are:-

R-50 — Methane (CH_4)

R-170— Ethane (C_2H_6)

R-290— Propane (C_2H_8)

R-600— Butane (C_4H_{10})

R-601— Isobentane [$CH(CH_3)_3$]

(*iv*) Inorganic compounds. Before the introduction of hydrocarbon group these refrigerants were most commonly used for all purposes. The important refrigerants of this group are:-

R-717— Ammonia (NH_3)

R-718— Water (H_2O)

R-729— Air (mixture of O_2, N_2, CO_2 etc.)

R-744— Carbon dioxide (CO_2)

R-764— Sulphur dioxide (SO_2)

(*v*) Unsaturated organic compound. The refrigerants belonging to this group possess ethylene or propylene as their constituents. They are:-

R-1120 — Trichloroethylene ($C_3H_4Cl_3$)

R-1130 — Dichloroethylene ($C_2H_4Cl_2$)

R-1150 — Ethylene (C_3H_6)

R-1270 — Propylene.

Desirable properties of an ideal refrigerant:-

An ideal refrigerant should possess the following properties:-

1. Thermodynamic properties:-

a) Low boiling point.

b) Low freezing point.

c) Positive pressures (but not very high) in condenser and evaporator.

d) High saturation temperature.

e) High latent heat of vapourisation.

2. Chemical Properties:-

a) Non-toxicity.

b) Non-flammable and non-explosive.

c) Non-corrosiveness.

d) Chemical stability in reacting.

e) No effect on the quality of stored (food and other) products like flowers, with other materials *i.e.*, furs and fabrics.

f) Non-irritating and odourless.

3. Physical Properties:-

a) Low specific volume of vapour.

b) Low specific heat.

c) High thermal conductivity.

d) Low viscosity.

e) High electrical insulation.

4. Other Properties:-

a) Ease of leakage location.

b) Availability and low cost.

c) Ease of handling.

d) High C.O.P.

e) Low power consumption per tonne of refrigeration.

f) Low pressure ratio and pressure difference.

Comparison between different refrigerants:-

Synthetic refrigerants that were commonly used for refrigeration, cold storage and air conditioning applications are:- R-11 (CFC-11), R-12 (CFC-12), R-22 (HCFC-22), R-502 (CFC-12+HCFC-22) etc. However, these refrigerants have to be phased out due to their Ozone Depletion Potential (ODP). The synthetic replacements for the older refrigerants are: R-134a (HFC-134a) and blends of HFCs. Generally, synthetic refrigerants are non-toxic and non-flammable. However, compared to the natural refrigerants the synthetic refrigerants offer lower performance and they also have higher Global Warming Potential (GWP). As a result, the synthetic refrigerants face an uncertain future. The most commonly used natural refrigerant is ammonia. This is also one of the oldest known refrigerants. Ammonia has good thermodynamic, thermophysical and environmental properties. However, it is toxic and is not compatible with some of the common materials of construction such as copper, which somewhat restricts its application. Other natural refrigerants that are being suggested are hydrocarbons (HCs) and carbon di-oxide (R-744). Though these refrigerants have some specific problems owing to their eco-friendliness, they are being studied widely and are likely to play a prominent role in future.

Table (3) shows a list of refrigerants being replaced and their replacements.

TABLE 3:- Refrigerants, their applications and substitutes.

Refrigerant	Application	Substitute suggested Retrofit(R)/New (N)
R 11(CFC) NBP = 23.7°C h_{fg} at NBP =182.5 kJ/kg T_{cr} =197.98°C C_p/C_v = 1.13 ODP = 1.0 GWP = 3500	Large air conditioning systems Industrial heat pumps.	R 123 (R,N) R 141b (N) R 245fa (N) n-pentane (R,N)
R 12 (CFC) NBP = -29.8°C h_{fg} at NBP=165.8 kJ/kg T_{cr} =112.04°C C_p/C_v = 1.126 ODP = 1.0 GWP = 7300	Domestic refrigerators Small air conditioners Water coolers Small cold storages.	R 22 (R,N) R 134a (R,N) R 227ea (N) R 401A,R 401B (R,N) R 411A,R 411B (R,N) R 717 (N)

R 22 (HCFC) NBP = -40.8°C h_{fg} at NBP=233.2 kJ/kg T_{cr} =96.02°C C_p/C_v = 1.166 ODP = 0.05 GWP = 1500	Air conditioning systems Cold storages.	R 410A, R 410B (N) R 417A (R,N) R 407C (R,N) R 507,R 507A (R,N) R 404A (R,N) R 717 (N)
R 134a (HFC) NBP = -26.15°C h_{fg} at NBP=222.5 kJ/kg T_{cr} =101.06°C C_p/C_v = 1.102 ODP = 0.0 GWP = 1200	Used as replacement for R 12 in domestic refrigerators, water coolers, automobile A/Cs etc.	No replacement required * Immiscible in mineral oils * Highly hygroscopic
R 717 (NH_3) NBP = -33.35°C h_{fg} at NBP=1368.9 kJ/kg T_{cr} =133.0°C C_p/C_v = 1.31 ODP = 0.0 GWP = 0.0	Cold storages Ice plants Food processing Frozen food cabinets.	No replacement required * Toxic and flammable * Incompatible with copper * Highly efficient * Inexpensive and available
R 744 (CO_2) NBP = -78.4°C h_{fg} at 40°C=321.3 kJ/kg T_{cr} =31.1°C C_p/C_v = 1.3 ODP = 0.0 GWP = 1.0	Cold storages Air conditioning systems Simultaneous cooling and heating (Trans critical cycle).	No replacement required * Very low critical temperature * Eco-friendly * Inexpensive and available
R718 (H_2O) NBP = 100°C h_{fg} at NBP=2257.9 kJ/kg T_{cr} =374.15°C C_p/C_v = 1.33 ODP = 0.0 GWP = 1.0	Absorption systems Steam jet systems.	No replacement required * High NBP * High freezing point * Large specific volume * Eco-friendly * Inexpensive and available

Numbering of Refrigerants:-

A numbering system for refrigerants was developed for hydrocarbons and halocarbons. According to ANSI/ASHRAE Standard 34-1992, the first digit is the number of unsaturated carbon–carbon bonds in the compound. This digit is omitted if the number is zero. The second digit is the number of carbon atoms minus one. This is also omitted if the number is zero. The third digit denotes the number of hydrogen atoms plus one. The last digit indicates the number of fluorine atoms. For example, the chemical formula for refrigerant R-123 is $CHCl_2CF_3$. In this compound:-

No unsaturated carbon–carbon bonds, first digit is 0

There are two carbon atoms, second digit is 2 – 1 = 1

There is one hydrogen atom, third digit is 1 + 1 = 2

There are three fluorine atoms, last digit is 3

The general chemical formula for the

Refrigerants is given as CmHnClpFq in which:-

$$n + p + q = 2(m+1)$$

m :- number of carbon atoms → (m -1)

n :- number of hydrogen atoms → (n +1)

p :- number chlorine atoms

q :- number halogen atoms like fluorine / bromine / iodine.

As shown above the number for the particular refrigerant is given by:-

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

Example 32:-

Dichloro-Monochloro –Methane

The chemical formula → $CHClF_2$

R → Refrigerant

m → 1

$n \rightarrow 1$

$P \rightarrow 1$ and

$q \rightarrow 2$

$n + p + q = 2\ (m+1)$

$1+1+2 = 2(1+1)$

$4 = 4$

It satisfies the balance equation so that the chemical formula is correct. The Designation for the above refrigerant stands that:- R (1-1) (1+1)2 = **R22**.

Example 33:-

Difluoro –Methane

The chemical formula→CH_2F_2

R → Refrigerant

$m \rightarrow 1$

$n \rightarrow 2$ and

$q \rightarrow 2$

$n + p + q = 2\ (m+1)$

$2 + 0 + 2 = 2(1+1)$

$4 = 4$

It satisfies the balance equation so that the chemical formula is correct. The Designation for the above refrigerant stands that:- R (1-1) (2+1)2 = **R32**.

Example 34:-

The chemical formula → $CBrF_3$

R → Refrigerant

m → 1

n → 0 and

q → 3

The Designation for the above refrigerant stands that:-

R (1-1) (0+1) 3 = **R13 B_1**.

Example 35:-

The chemical formula → $CClF_2CClF_2$

R → Refrigerant

m → 2

n → 0 and

q → 4

The Designation for the above refrigerant stands that:-

R (2-1) (0+1)4 = **R114**.

Example 36:-

The chemical formula for **Unsaturated organic compound** is →

CHCl = CHCl

R → Refrigerant

m → 2

n → 2 and

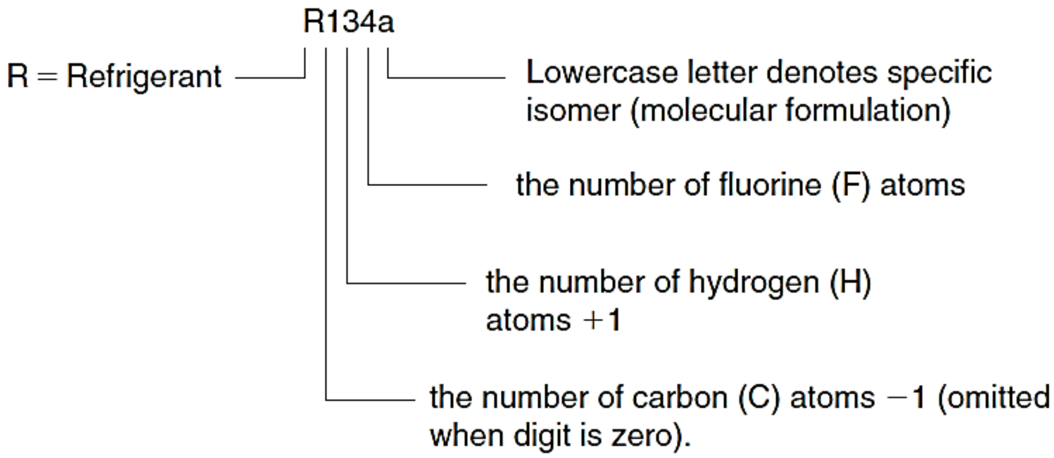
q → 0

The Designation for the above refrigerant stands that:-

R (2-1) (2+1)0 = **R1130**. (Adding 1000 for **Unsaturated organic compound).**

Example 37:-

CH_2FCF_3 OR $C_2H_2F_4$



Example 38:-

Find the chemical formula of R12.

Solution:-

As the number is 2 digit number, it is derived from methane base.

The digit "2" repesent number of flourine atoms.

The number of hydrogen atoms = 1 – 1 = 0.

To balance the methane, four monoatoms are required therefore number of chlorine atoms will be:-

$n + p + q = 2\ (m+1)$

$0 + p + 2 = 2\ (1+1)$

$p = 4 – 2 = 2$

Chemical formula of R12 is :-

$CmHnClpFq \rightarrow\ CCl_2F_2$.

Note:- inorganic compounds are in the 700 series. Identifi cation numbers are formed by adding the relative molecular mass of components to 700. For example, **R717** corresponds to ammonia which has a molecular mass of 17.