Refrigeration & Air-Conditioning

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Index

Module 1 - Fundamentals of Refrigeration

Lesson 1

Basic Refrigeration Cycle And Concepts, Standard Rating Of Refrigerating Machines

1.1. INTRODUCTION

The American Society of Heating, Air conditioning and Refrigerating Engineers (ASHARE) defines refrigeration as the science of providing and maintaining temperatures below the immediate surrounding temperatures. Thus it is a process of removing heat from a medium. In other words, the term “refrigeration” is used to denote maintenance of a system or body at a temperature lower than that of its surroundings. In other words it is the process of cooling a substance below the initial temperature of the substance.

1.2. WHAT IS AIR CONDITIONING?

Air conditioning refers the control of environmental conditions of air depending on the use of air conditioning. Air-conditioning signifies the control of an atmospheric environment either for human or to carry out industrial or scientific process efficiently. The purpose of air-conditioning is to supply sufficient volume of condition air having a specific amount of water vapour at a required temperature within a selected space. The space may be a small compartment such as a research laboratory, computer laboratory or big area like cinema hall, shopping centre, air port etc. Thus, air conditioning refers the control of temperature, relative humidity, quality of air and distribution of air depending upon the application of air conditioning. The environmental air conditioned in terms of temperature, humidity, purity of air results in greater comfort to occupants when applied to public places, offices and factories. Air conditioning designed for industrial purpose has several benefits including better control of product quality and efficiency.

1.3. IMPORTANCE OF REFRIGERATION IN DAIRY INDUSTRY

Refrigeration is a basic requirement for the processing and storage of milk and milk products as majority of dairy products are perishable in nature. The need of refrigeration is indicated below.

1. Chilling of milk at producers' level by employing bulk milk coolers and at milk chilling centers is the first requirement in dairy industry. Immediate cooling of milk to about 2-3 °C is very important to reduce the multiplications of micro-organisms and to get low bacterial count in the milk and milk products.
2. Processing of milk using either batch pasteurizer or HTST plant requires chilled water or any other cooling medium for cooling of milk.
3. Manufacture of many products requires refrigeration. e.g. butter, ice-cream etc.
4. Storage of milk and milk products requires maintaining low temperature in the cold storages depending on the type of product to be stored. e.g. milk is stored at around 3-4 °C while ice-cream is stored at -30 °C temperature.
5. Transportation of many products requires refrigerated vehicles to maintain the quality of products.
6. Low temperature storage is required for distribution of products as well as at the consumers' level.

Thus, refrigeration is a basic requirement from production to consumption level for dairy and food products. The concept of cold chain of transportation has been accepted in order to supply quality products to the consumers.

1.4. METHODS OF REFRIGERATION

1.4.1. Natural Methods

- Use of water: If a material is cooled below the initial temperature by using water, the process is called refrigeration. This method has limitations as desired value of low temperature water may not be available and it is not possible to reduce the temperature of the product/material to a required level.

- Ø Use of water added with salt: When salts (e.g. sodium chloride) are added in water, the temperature of the water falls and if this salt water is used for cooling , the process is called refrigeration. This is not practical as the drop in temperature is very small.

- Ø Use of naturally harvested ice: If naturally available ice is used for cooling, the process is called natural refrigeration. But, ice is not available every place and it can not meet the requirement of low temperature.

- Ø Use of ice and salt mixture: When salts (e.g. sodium chloride) are mixed in ice, it is possible to achieve temperature below 0 °C. In many towns, ice-cream is made using hand freezers employing ice and salt mixture for freezing of ice-cream.

- Ø Evaporative cooling with water: When ambient air passes through a spray of water or a wet grass pad, evaporation of part of water takes place by using sensible heat of the same air. As the sensible heat of the air decreases, the dry bulb temperature of the air decreases. The vapour produced due to evaporation of water goes with the air and hence total enthalpy of the air remains constant. This process is called adiabatic process. This method of air cooling is more effective when the relative humidity of the air is less. This principle of air cooling is employed in air coolers/dessert coolers. The lowest possible temperature of the air achieved in evaporative cooling is up to the wet bulb temperature of air. Another limitation of this method of cooling is higher humidity in the room area. The knowledge of psychrometry is necessary to understand the process of evaporative cooling which may not have much significance in air conditions, but the understanding of the process is important to understand the process of spray drying, cooling of water in cooling tower etc.

1.4.2. Artificial methods

The artificial method of refrigeration, which is known as vapour compression refrigeration system, is commonly used in dairy plants, food factories, air conditioning systems etc. The basic working principle is the same in all such systems. It may be house hold freeze, small capacity air conditioner or a big capacity refrigeration plant of any dairy/food factory. The basic cycle of the vapour compression refrigeration system is discussed in lesson 2.

1.5. UNIT OF REFRIGERATION

Capacity of refrigeration system is expressed as ton of refrigeration (TR). A ton of refrigeration is defined as the quantity of heat to be removed in order to form one ton (2000 lbs.) of ice at 0 °C in 24 hrs, from liquid water at 0 °C. This is equivalent to 12600 kJ/h or 210 kJ/min or 3.5 kJ/s (3.5 kW).

1 TR = 12600 kJ/h or 210 kJ/min or 3.5 kW.

The capacity of refrigeration plant required in any dairy/food plant can be estimated based on the cooling load requirement of the plant.

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Lesson 2. Elementary Vapour Compression Refrigeration Cycle

2.0. INTRODUCTION

The principle of refrigeration is based on second law of thermodynamics. It sates that heat does not flow from a low temperature body to a high temperature body without the help of an external work. In refrigeration process, since the heat has to be transferred from a low temperature body to a high temperature body some external work has to be done according to the second law of thermodynamics as shown in Fig.2.1. This external work is done by means of compressor.

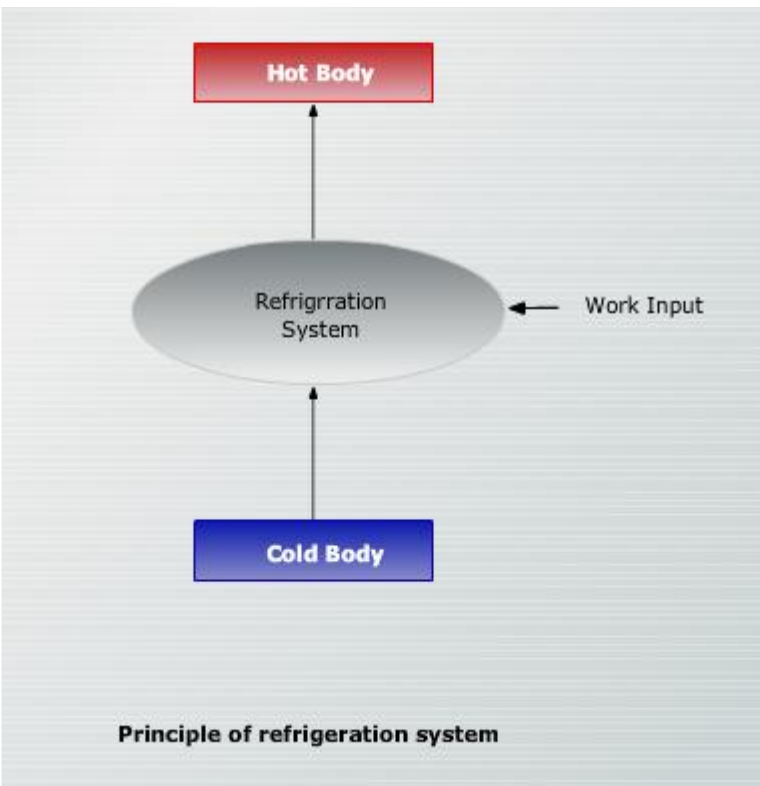

Fig. 2.1 Principle of refrigeration system

2.1. VAPOUR COMPRESSION REFRIGERATION SYSTEM

The vapour compression refrigeration system is widely used in commercial and domestic refrigeration and air conditioning plants. The working fluid called refrigerant completes its function upon evaporation during which it absorbs the heat in an amount equivalent to its refrigerating effect. Vapour compression refrigeration system using ammonia as refrigerant is widely used in India for industrial refrigeration, air conditioning and cold storages. Refrigeration is very essential requirement for low temperature storage of different food and dairy products. Refrigeration is also very essential for cold chain of handling many agricultural produce especially fruits and vegetables.

Vapour compression system mainly consists of compressor, condenser, receiver, expansion valve and evaporator. The refrigeration system is an enclosed gas tight system of tubes and equipment. It is so constructed that a control quality of refrigerant flows (due to expansion valve) from one necessary steps to another at definite and predetermined pressure. A block diagram of a vapour compression refrigeration system is shown in Fig. 2.2. The liquid refrigerant absorbs the heat from a zone of low pressure (evaporator) by means of its evaporation. The heat is dissipated in a zone of higher pressure (condenser) by means of condensation. The refrigerant like ammonia, R-22 etc. absorbs the heat at evaporator through evaporation. The compressor pumps the vapour refrigerant and the pressure of the refrigerant vapour is raised to a level so that it can be condensed at normal temperature of the cooling medium. Thus, four fundamental steps which are required to complete the mechanical cycles are as under.

1. Evaporation

2. Compression

3. Condensation

4. Pressure reduction or Expansion

The major components required in the system to accomplish the above four operations are (i) Evaporator (ii) Compressor (iii) Condenser (iv) Receiver and (v) Expansion valve.

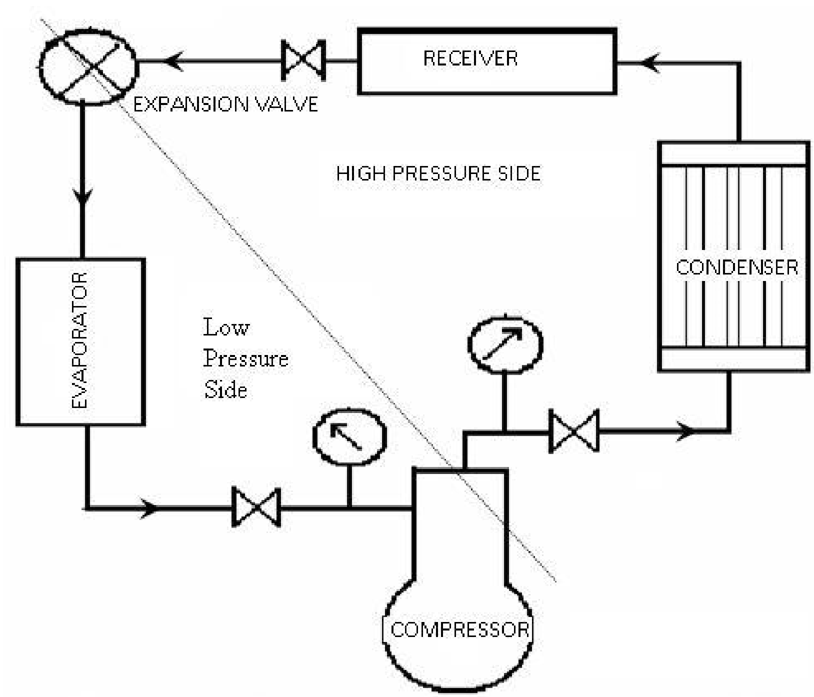

Fig. 2.2: Block diagram of vapour compression refrigeration system

2.1.1. Evaporation

The first step in the refrigeration cycle is the evaporation of liquid refrigerant. It is during this phase that the actual work of refrigeration is performed. The equipment in which the evaporation takes place is called the evaporator or cooling coil. The evaporator is the cooling element of the system.

The amount of liquid refrigerant that will evaporate in a given time depends upon the amount of heat transfer in the evaporator. The rate of heat transfer through a given heat exchanger depends on overall heat transfer co-efficient, area of heat transfer and the temperature difference. The refrigeration capacity of any system is based on the amount of cooling effect produced per unit time. The ton of refrigeration, which is common unit of the capacity, is the equivalent of 3000 kcal/h or 12600 kJ/h cooling effect. This is also equivalent to the amount of refrigeration done by melting of one ton (2000 lb.) of ice from and at °C in 24 hour.

2.1.2. Compression

The gas received from the evaporator is compressed by reciprocating compressor. It has two functions. (i) To draw the cold vapour from the evaporator to maintain a pressure in evaporator sufficiently low to achieve the desired evaporating temperature. (ii) To compress the vapour refrigerant and to deliver the compressed gas into the condenser where it can be liquefied at ordinary temperature by means of cooling water or air depending on the type of condenser.

Most of the power required to drive the compressor is absorbed by the vapour in the process of compression and appears as heat which must be taken out by the condenser. The compressor is located between the evaporator and the condenser. The condenser is connected to the discharge a side of the compressor. The compressor is a dividing component that joins the low pressure side to the high pressure side of the system.

2.1.3. Condensation

The purpose of the condensation is to provide a means by which the vapour rejects the heat absorbed at the evaporator as well as the heat of compression. The condenser is an arrangement of the pipe and tube possessing high heat condensing properties and so constructed as to allow for the entrance and exist of the refrigerant. These pipes or tubes are exposed to a cooling media (water or air) which absorbs the heat from refrigerant.

Connected to the outlet of the condenser and in some system forming an integral part of the condenser itself is a tank known as liquid receiver. The purpose of the receiver is to receive the liquid refrigerant upon condensing and to store until it is required by evaporator. The liquid receiver should be large enough to store the entire refrigerant charged in the system in liquid stage.

2.1.4. Pressure reduction or expansion

The refrigerant is in liquid stage under high pressure in the receiver and it is ready for re-use in the evaporator where much lower pressure is maintained. Some means of retaining of pressure difference is required. This is done by arranging a throttling device in the line between receiver and evaporator. This throttling maintains a higher pressure on one side and allows a lower pressure condition to exist on the other side when flow of refrigerant occurs.

The throttling device is often referred to as an expansion valve but its real purpose is to achieve pressure reduction so as to lower the boiling point of the liquid refrigerant.

2.1.5. Receiver

The receiver of the system receives the refrigerant from the condenser and delivers it to the evaporator through expansion valve depending on the requirement of evaporator. This component has important role to collect the entire charge of the system during maintenance work of the plant.

The basic cycle of vapour compression refrigeration is the same for small and large capacity vapour compression refrigeration systems. Refrigerant is a the heat transfer medium in vapour compression refrigeration cycle which absorbs the heat through evaporation at the evaporator and rejects the heat absorbed at evaporator plus the heat of work of compression at the condenser.

Thus, according to second law of thermodynamics,

Heat rejected at condenser, kJ/h = Refrigerating effect, kJ/h + Work of compression, kJ/h.

2.2. PRIMARY REFRIGERANT

The working fluid of vapour compression refrigeration system is known as primary refrigerant which absorbs the heat through evaporation at the evaporator and again becomes liquid on cooling in the condenser. Primary refrigerants include only those working fluids which pass through the cycle of evaporation, compression, condensation and expansion. These refrigerants have very low boiling point. e.g. boiling point of ammonia is -33.3 ° C at atmospheric pressure. There are many refrigerants which as discussed in another lesion.

2.3. CO-EFFICIENT OF PERFORMANCE (C.O.P.)

The performance of the refrigeration system is expressed as C.O.P. which is the ratio of refrigerating effect produced to the work of compression. The cooling effect (out put of the system) is produced at the evaporator and the refrigerant is compressed by the compressor using the electrical power (input to the system). The various aspects associated with performance of the system are discussed in some another lesson. Higher C.O.P. is always desirable in order to get more cooling effect with less energy in put.

C.O.P=Refrigeration effect, kJ/hWork of compression, kJ/h'

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Lesson 3.

Representation Of Vapour Compression Refrigeration Cycle On P-V, T- Φ And P-H Diagrams, Use Of Refrigerant Properties Tables And P-H (Mollior) Charts.

3.1. INTRODUCTION

The properties of refrigerant do not remain constant during the operation of the system. Refrigerant properties tables are used to know the thermodynamic properties of refrigerant at saturation points at various temperatures and pressures. The properties of refrigerant in any thermodynamic state can be represented on Pressure-Enthalpy (P-H) chart. These charts (Mollier Diagrams) are plotted for 1.0 kg mass of refrigerant. The values of enthalpy and entropy are based on an arbitrarily chosen datum. The refrigerant properties tables and charts are used to determine theoretical refrigerating effect, work of compression and co-efficient of performance (C.O.P.) of the refrigeration system operating under different conditions.

In order to understand the presentation of the working cycle on P-V, T- ф and P-H diagrams, the block diagram of the system which is shown in lesson 2 is again indicated Fig. 3.1. The various processes involved in the cycle are as under.

1-2, Compression

2-3, Condensation

3-4, Expansion

4-1, Evaporation

The above mentioned processes are indicated on P-V, T- ф and P-H in Fig. 3.2, 3.3 and 3.4 respectively.

In order to get the values of enthalpy, entropy, specific volume, temperature etc, it is necessary to locate the points on actual diagrams corresponding to the operating suction and discharge pressure of the system.

3.2. REFRIGERANT PROPERTIES TABLES

Refrigerant properties table are used to obtain values of pressure, volume, enthalpy and entropy of refrigerant at different temperatures under saturated condition. Properties tables of R-22 and R-717 are given bellow. The thermodynamic properties of refrigerant under superheated, sub cooled and two phase conditions can not be obtained from these tables but can be obtained from P-H (Mollier chart) diagram of the respective refrigerants.

3.3. REFRIGERANT P-H DIAGRAMS

P-H diagram are used to obtain the thermodynamic properties of refrigerants at any point during operation of refrigeration system. When the operating suction and discharge pressure are known we can get values like enthalpy, specific volume and corresponding temperature etc. from the diagram. The theoretical COP can be calculated by reading the enthalpy values at different stages. The P-H diagram of R-717 and R-22 are given in Figure 3.5 and 3.6 respectively.

3.4. HOW TO USE REFRIGERANT PROPERTY TABLES AND P-H CHARTS?

Suppose, an ammonia VCR plant operates at suction pressure of 2 bar (absolute) and condensing pressure of 12 bar (absolute) so the corresponding values of temperature, specific volume, specific enthalpy and specific entropy of both gas and liquid can be read in the front of the pressure value in table. (It is highlighted in the ammonia properties table.)

The operating cycle for above mentioned conditions for ammonia is marked in the chart.

From the chart

h_1 = 1420 kJ/ kg (enthalpy at the suction of compressor).

h_2 = 1650 kJ/ kg (enthalpy at the end of compression)

h_3 = 325 kJ/ kg (enthalpy at the end of condensation)

The values of h_1 and h_3 can be obtained from chart as well as from table but the value of h_2 can only be obtained from P-H chart.

Table 3.1 R-22 Properties table

Saturation temp. in °C	*Saturation pressure in bar*	*Specific volume in m^3/kg*		*Specific enthalpy in kJ / kg*			*Specific entropy in kJ / kg K*	
		Liquid	*Vapour*	*Liquid*	*Vapour*	*Latent*	*Liquid*	*Vapour*
(t)	(p)	(v_f)	(v_g)	(h_f)	(h_g)	(h_{fg})	(s_f)	(s_g)
−100	0.020 09	0.000 643	8.3412	−63.45	203.73	267.18	−0.3144	1.2293
−95	0.031 50	0.000 647	5.4344	−58.14	206.19	264.33	−0.2843	1.2004
−90	0.047 92	0.000 652	3.6381	−52.87	208.64	261.51	−0.2550	1.1736
−85	0.077 31	0.000 656	2.5204	−47.61	211.11	258.72	−0.2269	1.1489
−80	0.103 93	0.000 661	1.7816	−42.40	213.60	256.00	−0.1989	1.1267
−75	0.147 59	0.000 666	1.2842	−37.17	216.11	253.28	−0.1721	1.1066
−70	0.205 17	0.000 672	0.9420	−31.93	218.62	250.55	−0.1461	1.0874
−65	0.279 65	0.000 677	0.7037	−26.68	221.16	247.84	−0.1206	1.0702
−60	0.374 48	0.000 683	0.5351	−21.42	223.67	245.09	−0.0959	1.0543
−55	0.496 21	0.000 689	0.4131	−16.13	226.18	242.31	−0.0712	1.0396
−50	0.647 58	0.000 696	0.3229	−10.81	228.69	239.50	−0.0473	1.0262
−45	0.832 41	0.000 702	0.2556	−5.40	231.20	236.60	−0.0234	1.0137
−40	**1.055 86**	**0.000 709**	**0.2049**	**0.00**	**233.67**	**233.67**	**0.0000**	**1.0024**
−38	1.158 62	0.000 712	0.1882	2.20	234.65	232.45	0.0096	0.9982
−36	1.269 10	0.000 715	0.1728	4.40	235.65	231.25	0.0188	0.9940
−34	1.387 31	0.000 719	0.1590	6.58	236.60	230.02	0.0280	0.9902
−32	1.513 24	0.000 721	0.1465	8.83	237.60	228.77	0.0373	0.9860
−30	1.646 90	0.000 724	0.1353	11.05	238.55	227.50	0.0464	0.9283
−28	1.789 38	0.000 727	0.1253	13.29	239.52	226.23	0.0557	0.9785
−26	1.940 69	0.000 731	0.1161	15.58	240.48	224.90	0.0645	0.9747
−24	2.102 07	0.000 734	0.1077	17.77	241.41	223.64	0.0733	0.9710
−22	2.274 48	0.000 738	0.1000	19.99	242.33	222.34	0.0821	0.9676
−20	2.457 93	0.000 741	0.0930	22.21	243.25	221.04	0.0908	0.9638
−18	2.653 10	0.000 744	0.0865	24.47	244.17	219.70	0.0996	0.9605
−16	2.859 03	0.000 748	0.0806	26.72	245.08	218.36	0.1080	0.9576
−14	3.078 76	0.000 752	0.0752	28.94	245.96	217.02	0.1164	0.9538
−12	3.311 72	0.000 756	0.0701	31.16	246.84	215.68	0.1248	0.9508
−10	3.557 93	0.000 759	0.0655	33.40	247.72	214.32	0.1336	0.9479
−8	3.813 21	0.000 763	0.0612	35.66	248.60	212.94	0.1419	0.9454
−6	4.091 72	0.000 767	0.0573	37.92	249.46	211.54	0.1503	0.9425
−4	4.379 72	0.000 771	0.0536	40.19	250.30	210.11	0.1591	0.9396
−2	4.683 17	0.000 775	0.0503	42.52	251.14	208.62	0.1675	0.9370

Refrigeration Table For R-22

t	p	v_f	v_g	h_f	h_g	h_{fg}	s_f	s_g
0	5.002 07	0.000 779	0.0471	44.94	251.97	207.03	0.1763	0.9345
1	5.168 41	0.000 781	0.0457	46.16	252.37	206.21	0.1809	0.9333
2	5.339 17	0.000 783	0.0443	47.38	252.77	205.39	0.1855	0.9320
3	5.514 34	0.000 785	0.0429	48.64	253.17	204.53	0.1901	0.9303
4	5.693 52	0.000 787	0.0416	49.91	253.56	203.65	0.1943	0.9291
5	5.876 21	0.000790	0.0403	51.16	253.95	202.79	0.1989	0.9278
6	6.062 76	0.000 792	0.0391	52.41	254.33	201.92	0.2035	0.9266
7	6.253 93	0.000 794	0.0379	53.68	254.71	201.03	0.2077	0.9253
8	6.449 93	0.000 797	0.0368	54.94	255.08	200.14	0.2123	0.9241
9	6.650 34	0.000 799	0.0357	56.22	255.44	199.22	0.2169	0.9228
10	6.855 17	0.000 801	0.0346	57.52	255.81	198.29	0.2211	0.9216
11	7.066 21	0.000 803	0.0336	58.80	256.15	197.35	0.2257	0.9203
12	7.277 24	0.000 805	0.0326	60.07	256.48	196.41	0.2303	0.9190
13	7.497 93	0.000 808	0.0316	61.38	256.83	195.45	0.2345	0.9178
14	7.725 52	0.000 810	0.0307	62.72	257.17	194.45	0.2391	0.9165
15	7.955 17	0.000 813	0.0298	64.02	257.50	193.48	0.2437	0.9152
16	8.187 58	0.000 315	0.0289	65.32	257.81	192.48	0.2483	0.9140
17	8.425 52	0.000 817	0.0281	66.63	258.13	191.50	0.2529	0.9127
18	8.632 41	0.000 820	0.0273	67.95	258.44	190.49	0.2571	0.9115
19	8.917 93	0.000 822	0.0266	69.27	258.73	189.46	0.2617	0.9102
20	9.172 41	0.000 825	0.0258	70.59	259.00	188.41	0.2663	0.9089
21	9.433 10	0.000 827	0.0251	71.93	259.29	187.36	0.2709	0.9077
22	9.699 31	0.000 830	0.0244	73.31	259.58	186.27	0.2755	0.9065
23	9.971 03	0.000 833	0.0237	74.66	259.85	185.19	0.2801	0.9052
24	10.248 28	0.000 835	0.0230	76.04	260.11	184.07	0.2847	0.9039
25	10.531 03	0.000 838	0.0224	77.39	260.38	182.99	0.2889	0.9027
26	10.819 31	0.000 841	0.0218	78.79	260.64	181.85	0.2935	0.9014
27	11.113 10	0.000 844	0.0212	80.16	260.89	180.73	0.2981	0.9002
28	11.412 41	0.000 846	0.0206	81.54	261.12	179.58	0.3023	0.8989
29	11.690 34	0.000 849	0.0200	82.96	261.37	178.41	0.3069	0.8977
30	12.034 48	0.000 852	0.0194	84.38	261.60	177.22	0.3115	0.8964
31	12.351 03	0.000 855	0.0189	85.77	261.81	176.04	0.3161	0.8948
32	12.673 10	0.000 858	0.0184	87.17	262.02	174.85	0.3207	0.8935
33	13.000 70	0.000 861	0.0179	88.57	262.22	173.65	0.3249	0.8922
34	13.337 93	0.000 864	0.0174	90.00	262.40	172.40	0.3295	0.8909
35	13.682 76	0.000 867	0.0169	91.43	262.58	171.15	0.3337	0.8893
36	14.030 34	0.000 870	0.0165	92.85	262.74	169.89	0.3383	0.8880
37	14.382 07	0.000 874	0.0161	94.24	262.88	168.64	0.3429	0.8868
38	14.743 45	0.000 877	0.0156	95.63	263.00	167.37	0.3471	0.8851
39	15.111 03	0.000 881	0.0152	97.03	263.13	166.10	0.3517	0.8838

Table 3.2 Ammonia Properties table

Saturation temp. in °C	Saturation pressure in bar	Specific volume in m^3/kg		Specific enthalpy in kJ/kg			Specific entropy in kJ / kg K	
		Liquid	Vapour	Liquid	Vapour	Latent	Liquid	Vapour
(t)	(p)	(v_f)	(v_g)	(h_f)	(h_g)	(h_{fg})	(s_f)	(s_g)
−50	0.408 96	0.001 426	2.6281	−44.43	1373.27	1417.70	−0.1943	6.1603
−48	0.459 72	0.001 431	2.3565	−35.44	1376.80	1412.24	−0.1551	6.1192
−46	0.516 00	0.001 436	2.1177	−26.60	1380.20	1406.80	−0.1157	6.0789
−44	0.577 10	0.001 441	1.9062	−17.81	1383.31	1401.12	−0.0769	6.0394
−42	0.644 55	0.001 446	1.7196	−8.97	1386.67	1395.64	−0.0384	6.0008
−40	**0.717 93**	**0.001 451**	**1.5537**	**0.00**	**1390.02**	**1390.02**	**0.0000**	**5.9631**
−38	0.793 84	0.001 456	1.4077	8.97	1393.13	1384.16	0.0381	5.9262
−36	0.886 07	0.001 462	1.2775	17.81	1396.35	1378.54	0.0758	5.8900
−34	0.980 96	0.001 467	1.1614	26.84	1399.51	1372.67	0.1131	5.8545
−32	1.081 65	0.001 472	1.0574	35.68	1402.48	1366.80	0.1506	5.8198
−30	1.195 86	0.001 477	0.9644	44.66	1405.60	1360.94	0.1876	5.7856
−28	1.317 24	0.001 483	0.8820	53.68	1408.53	1354.85	0.2242	5.7521
−26	1.447 90	0.001 488	0.8069	62.61	1411.45	1348.84	0.2607	5.7195
−24	1.588 41	0.001 494	0.7397	71.73	1414.39	1342.66	0.2970	5.6872
−22	1.739 86	0.001 500	0.6793	80.76	1417.28	1336.52	0.3330	5.6556
−20	1.902 76	0.001 505	0.6244	89.78	1420.02	1330.24	0.3684	5.6244
−18	2.078 07	0.001 511	0.5750	98.76	1422.72	1323.96	0.4043	5.5939
−16	2.265 51	0.001 517	0.5303	107.83	1425.28	1317.45	0.4397	5.5639
−14	2.466 34	0.001 523	0.4896	116.95	1427.88	1310.93	0.4747	5.5356
−12	2.680 69	0.001 529	0.4526	126.16	1430.54	1304.38	0.5096	5.5055
−10	2.908 96	0.001 536	0.4189	135.37	1433.05	1297.68	0.5443	5.4770
−8	3.153 65	0.001 541	0.3884	144.35	1435.33	1290.98	0.5789	5.4487
−6	3.413 80	0.001 548	0.3604	153.56	1437.93	1284.37	0.6139	5.4210
−4	3.690 62	0.001 554	0.3348	162.77	1440.02	1277.25	0.6473	5.3940
−2	3.984 27	0.001 561	0.3113	171.98	1442.17	1270.19	0.6812	5.3670
0	4.295 86	0.001 567	0.2898	181.20	1444.45	1263.25	0.7151	5.3405
1	4.458 48	0.001 571	0.2798	185.80	1445.49	1259.69	0.7321	5.3277
2	4.626 62	0.001 574	0.2702	190.40	1446.54	1256.14	0.7487	5.3145

Refrigeration Table For R-717

t	p	v_f	v_g	h_f	h_g	h_{fg}	s_f	s_g
3	4.799 31	0.001 578	0.2610	195.17	1447.59	1252.42	0.7653	5.3017
4	4.976 82	0.001 582	0.2521	199.85	1448.63	1248.78	0.7818	5.2888
5	5.158 62	0.001 585	0.2436	204.46	1449.56	1245.10	0.7989	5.2765
6	5.347 45	0.001 589	0.2354	209.06	1450.49	1241.43	0.8154	5.2638
7	5.540 07	0.001 592	0.2276	213.73	1451.54	1237.81	0.8320	5.2513
8	5.738 20	0.001 595	0.2201	218.50	1452.54	1234.04	0.8487	5.2389
9	5.941 86	0.001 598	0.2128	223.11	1453.39	1230.28	0.8652	5.2266
10	6.151 03	0.001 603	0.2060	227.72	1454.22	1226.50	0.8814	5.2141
11	6.366 41	0.001 607	0.1992	232.53	1455.30	1222.77	0.8979	5.2020
12	6.587 31	0.001 610	0.1928	237.15	1456.11	1218.96	0.9142	5.1900
13	6.814 20	0.001 614	0.1866	241.92	1456.96	1215.04	0.9307	5.1780
14	7.047 17	0.001 617	0.1807	246.60	1457.80	1211.20	0.9470	5.1659
15	7.282 76	0.001 621	0.1751	251.44	1458.63	1207.19	0.9634	5.1542
16	7.531 04	0.001 624	0.1696	256.14	1459.47	1203.33	0.9794	5.1421
17	7.781 38	0.001 628	0.1643	259.88	1460.24	1200.36	0.9956	5.1302
18	8.038 62	0.001 633	0.1592	265.54	1460.92	1195.38	1.0118	5.1186
19	8.305 51	0.001 637	0.1543	270.35	1461.75	1191.40	1.0280	5.1073
20	8.572 41	0.001 641	0.1496	275.16	1462.60	1187.44	1.0442	5.0956
21	8.851 72	0.001 645	0.1450	279.77	1463.21	1183.44	1.0604	5.0843
22	9.136 55	0.001 648	0.1407	284.56	1463.84	1179.28	1.0763	5.0729
23	9.426 90	0.001 652	0.1365	289.37	1464.63	1175.26	1.0924	5.0616
24	9.726 90	0.001 656	0.1324	294.19	1465.33	1171.14	1.1083	5.0503
25	10.027 60	0.001 661	0.1284	298.90	1465.84	1166.94	1.1242	5.0391
26	10.340 69	0.001 665	0.1246	303.82	1466.59	1162.77	1.1402	5.0279
27	10.661 37	0.001 669	0.1210	308.63	1467.22	1158.59	1.1563	5.0170
28	10.991 72	0.001 673	0.1174	313.45	1467.85	1154.40	1.1721	5.0061
29	11.327 58	0.001 678	0.1140	318.26	1468.45	1150.19	1.1879	4.9951
30	11.668 96	0.001 682	0.1107	323.08	1468.87	1145.79	1.2037	4.9842
31	12.016 55	0.001 686	0.1075	327.89	1469.50	1141.61	1.2195	4.9733
32	12.375 17	0.001 691	0.1045	332.71	1469.94	1137.23	1.2350	4.9624
33	12.744 82	0.001 695	0.1015	337.52	1470.36	1132.84	1.2508	4.9517
34	13.121 37	0.001 700	0.0987	342.48	1470.92	1128.44	1.2664	4.9409
35	13.503 45	0.001 704	0.0960	347.50	1471.43	1123.93	1.2821	4.9302

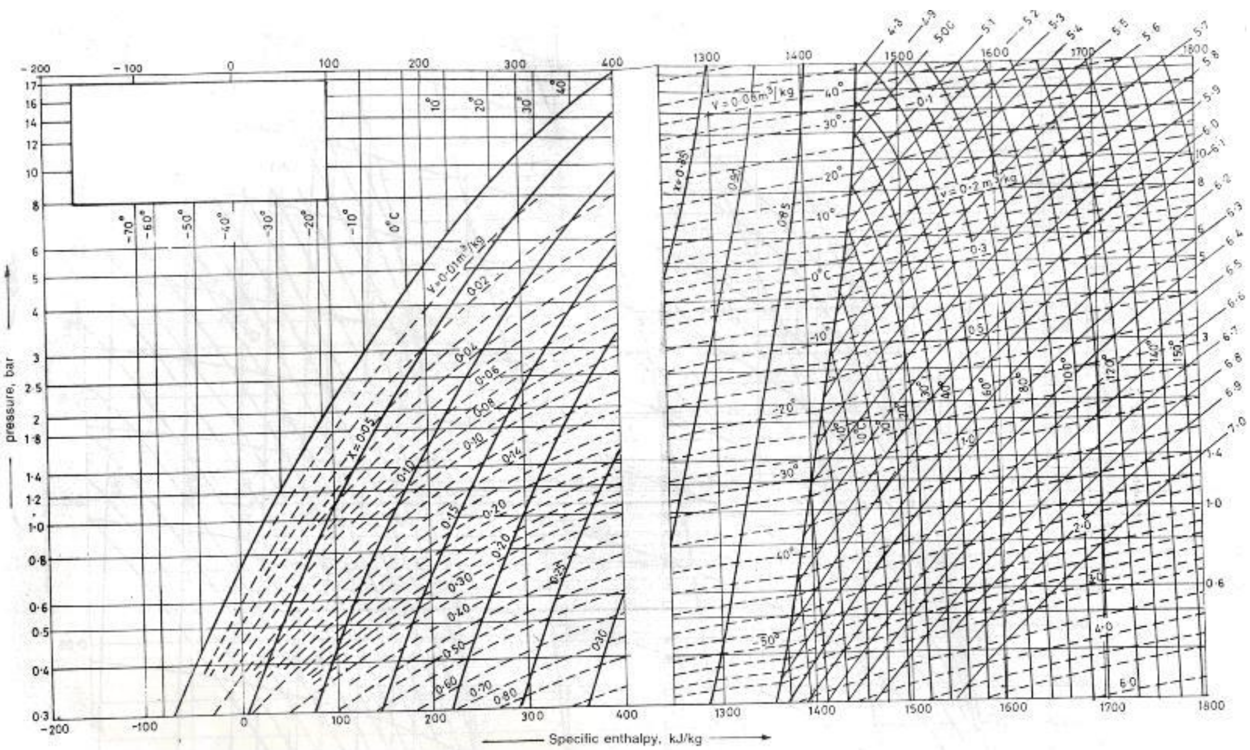

Fig 3.5 P-H diagram of ammonia refrigerant

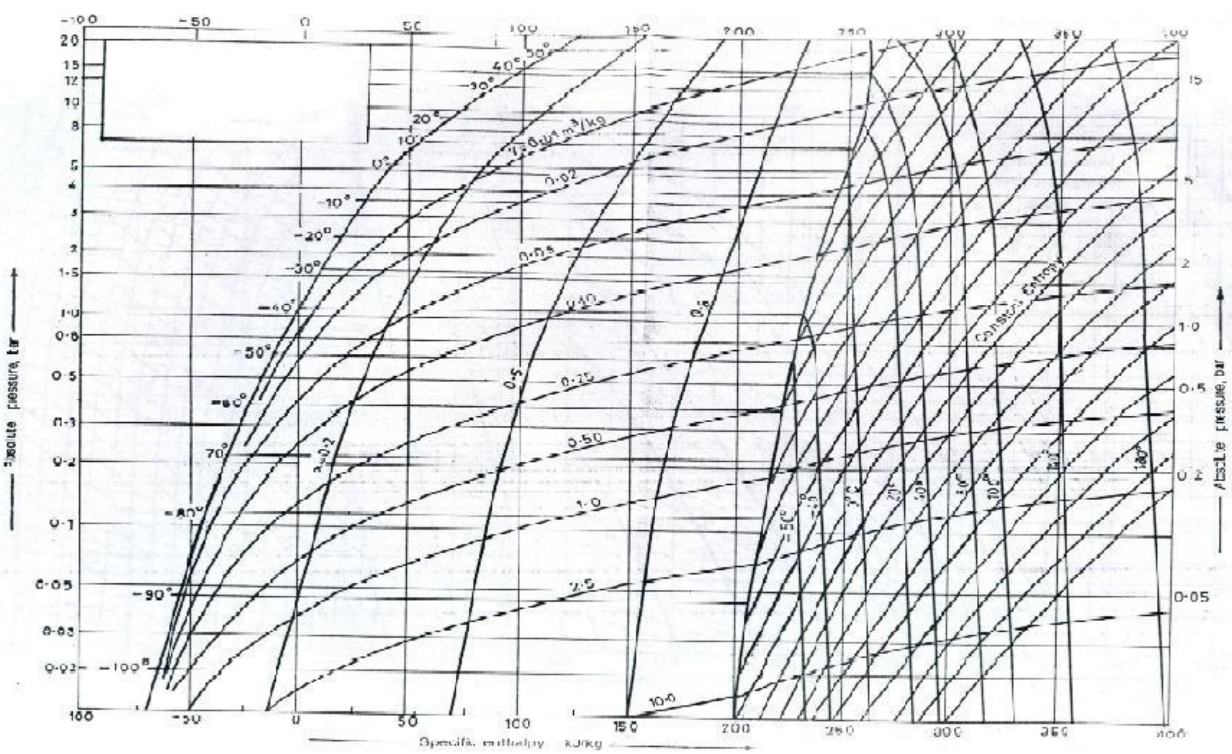

Fig 3.6 P-H diagram of R-22 refrigerant

Lesson 4.

Theoretical and Actual Cycle, Performance of Refrigeration Cycle

4.1. INTRODUCTION

A working cycle of VCR system is represented in P-H diagram in **Fig. 4.1** . The processes of evaporation (4-1) and condensation (2-3) are treated as isothermal processes. The expansion process (3-4) is a constant enthalpy process; while compression process (1-2) is an isentropic process. Based on the operating condition evaporating cycle can be presented on the P-H diagram to obtain values of refrigerating effect, work of compression to calculate the theoretical process of the plant. The operating condition of the plant varies depending upon the temperature requirement as well the temperature of cooling medium available at condenser. The cycle described bellow is a simple saturated cycle in which the liquid after condensation and vapour after evaporation are saturated and lie on the saturated liquid and vapour curves respectively in P-H chart.

4.2. ACTUAL CYCLE

The actual cycle deviates slightly due to pressure drop caused by friction in piping and valves. In addition to this there will be heat loss or gain depending on the temperature difference between the refrigerant and the surrounding. Further compression will be polytropic due to friction and heat transfer instead of isentropic. The actual VCR cycle is depicted in fig. 4.2. It is clear from the both actual and theoretical cycles that little pressure drop takes place when refrigerant pass through the evaporator (From 4-a). The processes a-b and b-c are depicting superheating of suction vapour inside the evaporator and outside the evaporator respectively; where as the processes c-d and d-1 are showing the pressure drop in line and wire drawing effect pressure drop inside the compressor valve respectively. The processes 2-e and e-f are the pressure drop in compressor discharge valve and delivery line respectively. Processes f-h is desuperheating of gas in condenser and h-3 is sub cooling of liquid in condenser.

The operating cycle deviates as discussed above however, the saturated cycle is used to determine the COP of the cycle for all practical purpose. It is well observed that the pressure drop in the evaporator due to frictional pressure drop and momentum pressure drop is larger than that of the condenser.

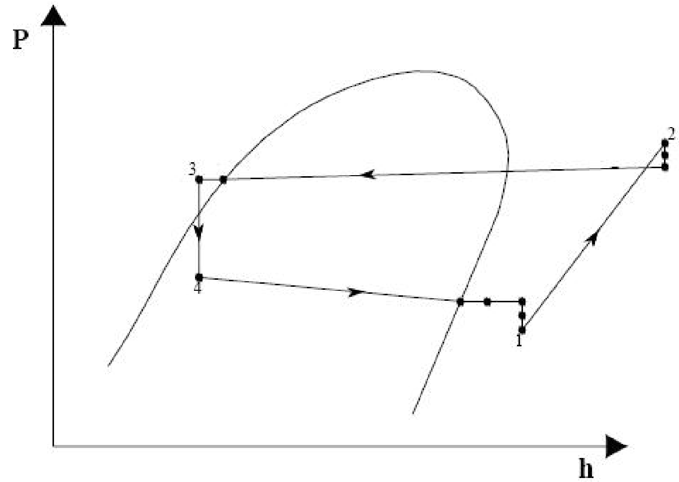

Fig. 4.2: Actual vapour compression refrigeration cycle on P-H diagram

4.3. PERFORMANCE OF REFRIGERATION SYSTEM

The performance of refrigeration plant is expressed as "Co-efficient of Performance (COP)" which is defined as the ratio of refrigerating effect produced to the work of compression. It is obvious that higher COP is desirable to reduce the operating cost of the system. The COP is greatly influenced by several factors like operating variables, part load performance, design of plant components, maintenance of the system etc. Many research workers have emphasized the importance of matching design of plant components and efficient heat transfer at evaporator and condenser in order to achieve optimum level of performance.

Refrigeration systems have been a subject of continuous modification and development to meet the specific demand of the industry. Systems for large-scale air conditioning for business complexes, cold storages, hospitals, library, etc. are very common in US and in many other developed countries. There are number of technical and global issues which include energy conservation, alternative refrigerant due to depletion caused by CFC refrigerants and technical problems of air quality. It is necessary to implement energy management strategy which involves total commitment at all levels, education/awareness campaign, training, energy audit, cost benefit analysis, maintenance programme etc.

4.4. THEORETICAL AND ACTUAL COP

The operating cycle of the plant can be plotted on P-H diagram corresponding to suction and the discharge pressure of the system to calculate theoretical COP of the plant. The measurement of actual COP, which is the ratio of actual cooling effect produced to the actual power consumption, is difficult under practical conditions. The work of compression can be easily obtained by installing energy meters but the rate of cooling effect produced is difficult to estimate, as the refrigerating effect is continuously in use. The measurement of actual COP is more important to know the performance of the plant.

The estimation of actual refrigerating effect produced is practically difficult; however under certain conditions, it is possible to calculate the rate of refrigerating effect produced by indirect method. For any vapour compression refrigeration plant, $Q = R + W$; where Q = heat removed at condenser, kJ/h; R = refrigerating effect produced, kJ/h; W = work of compression, kJ/h. If it is possible to estimate the value of heat removed at condenser by measuring the change in enthalpy of the cooling medium, the refrigerating effect can be estimated as the work of compression is measured by energy meters. The COP of vapour compression refrigeration plant varies from 2.5 to 4.5 depending on the operating conditions of the plant.

4.5. FACTORS AFFECTING THE PERFORMANCE OF REFRIGERATION PLANT

The COP of a vapour compression refrigeration plant is mainly affected by operating conditions of the plant as well as maintenance aspects of the plant. The operating conditions of the refrigeration plant play very important role on the performance of the refrigeration system. The important factors, which are affecting the performance of vapour compression refrigeration system, are listed below.

1. Evaporating temperature

2. Condensing temperature

3. Sub-cooling of liquid refrigerant

4. Super heating of suction gas

5. Heat transfer at evaporator condenser

6. Presence of non-condensable gases in the system

7. Volumetric efficiency of compressor

8. Multi- stage compression and throttling system

9. Design of plant components

10. Maintenance of the plant

4.5.1. Effect of evaporating temperature

The evaporating temperature of the plant is maintained depending on the temperature requirement for the given application. Lower evaporating temperature reduces the refrigerating effect per kg of refrigerant circulated in the system and increases the work of compression leading to reduction in COP of the plant. Therefore, it is desirable to operate a refrigeration plant with highest possible evaporating temperature and undue lower evaporating temperature should be avoided.

4.5.2. Effect of condensing temperature

The condensing temperature is fixed by the temperature of cooling medium available as well as the efficiency of heat transfer at the condenser. The rate of heat rejected at the condenser is function of overall heat transfer co-efficient, heat transfer area and the temperature difference between the refrigerant and the cooling medium. Lower condensing temperature is desirable to get higher COP of the system. The efficiency of cooling tower is very important to get lower temperature of water for water-cooled condenser of the plant. In case of evaporative condenser, dry bulb temperature, wet bulb temperature and velocity of air play important role in heat transfer at the condenser. It is possible to save energy by operating the refrigeration plant during colder hours of the day.

4.5.3. Sub-cooling of liquid refrigerant

Sub-cooling of liquid refrigerant is desirable as it increases the refrigerating effect without any change in work of compression. The sub-cooling achieved in the condenser using low temperature water is maximum advantageous as compared to sub- cooling using liquid refrigerant from the receiver.

4.5.4. Effect of super heating of suction gas

For trouble free operation of refrigeration system, a little super heating of suction gas is desirable to eliminate the chances of wet compression. However, excessive super heating of suction gas is not desirable as it increases the work of compression and piston displacement of the compressor. It also causes compressor head cooling difficulties in tropical countries like ours. The power consumption for compression of refrigerant under superheated condition will increase due to diverging nature of isentropic lines.

4.5.5. Heat Transfer at evaporator and condenser

The efficiency of heat transfer at condenser and evaporator is greatly influencing the performance of the plant. The design of these components of the plant to achieve maximum rate of heat transfer under the practical conditions is very important consideration. The rate of heat transfer is function of overall heat transfer coefficient (U-value), heat transfer area and the temperature difference. Efficient cleaning of condenser and defrosting of evaporator at regular interval is important to achieve higher U-value. It is necessary to examine the factors affecting the rate of heat transfer depending on the type of evaporator and condenser to optimize the rate of heat transfer

4.5.6. Multi-stage compression system

It is usual practice to manufacture single stage compressor with compression ratio around 7 to 8. However, for large capacity plants even in this pressure ratio, the multi stage compression is employed. The compressed refrigerant vapour after first stage compression is passed through flash chamber where it becomes saturated by cooling at intermediate pressure. Multi-compression system reduces the work of compression as well as improves the volumetric efficiency of the compressor and decreases the temperature of refrigerant gas leaving the compressor. The volumetric efficiency plays significant role on the capacity of the refrigeration plant. Optimum inter-stage pressure is necessary to maintain in order to achieve maximum advantage of two-stage compression system.

4.6. ESTIMATION OF REFRIGERATION PLANT CAPACITY

The total refrigeration requirement of a dairy plant can be estimated by calculating the cooling load of various operations and based on the hourly refrigeration load requirement. The capacity is expressed as ton of refrigeration which is equivalent to 12600 kJ/h (3000 kcal/h) heat removed by the plant from the evaporator. The refrigeration load of chillers, pasteurizers, cold storages etc. is calculated to arrive at the total cooling requirement of the plant. A simple way of estimating a capacity of refrigeration plant of a bulk cooler having 5000 liters milk storage capacity is given below. (Density of milk=1.032 g/cm^3; Specific heat of milk = 3.9 kJ/kg K; Initial temperature of milk=35°C; Final cooling temperature=2°C)

Heat to be removed from the milk = 5000 x 1.032 x 3.9 x (35-2)

= 664092 kJ

The bulk coolers are designed to cool the entire quantity of milk in about 3.5 hours. Therefore,

Heat to be removed from the milk per hour = 189741 kJ/h

Capacity of the refrigeration plant in ton = 442723/12600

= 15 ton

4.7. EFFICIENT USE OF REFRIGERATION

It is essential to make efficient use of refrigeration produced to achieve over all advantage of efficient operation of the plant. In dairy plants usually ice-bank refrigeration system (thermal energy storage system- TES) is widely used for chilling and processing of milk. TES system is widely recognized as a demand side management technology for shifting cooling electric demand from peak day-time period to off peak night time. In many countries TES has enabled users to significantly reduce their electricity cost by reducing peak demand and taking advantage of lower off peak usage rates often with large utility incentive payment.

The following aspects are important for conservation of refrigeration in dairy and food plants.

1. Efficient design of cold storage and ice-bank refrigeration system.

2. Selection of energy efficient refrigeration plant and its components.

3. Minimize air change load in cold storages

4. Proper insulation to chilled water pipelines.

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Lesson 5.

Effect of Change of Operation Conditions on the Working of Vapour Compression Refrigeration Plant.

5.1. INTRODUCTION

The operating conditions of the plant vary depending on the evaporating and condensing temperature. Operating condition changes due to variation in the condensing pressure, as it is governed by the temperature of the cooling medium available at the condenser. It is very essential to understand the effect of variations of operation conditions in order to operate the refrigeration plant under optimum operating conditions. The effect of change of operating conditions is discussed bellow.

5.2. EFFECT OF EVAPORATING PRESSURE

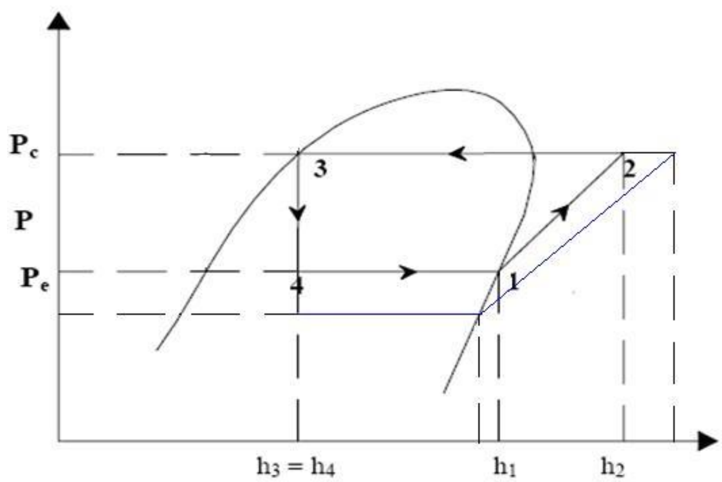

Fig. 5.1 Effect of evaporating temperature

Enthalpy (H)

Here as shown in figure 1-2-3-4 is cycle operated at P_e evaporating pressure and cycle 1’-2’-3-4’ is operated at evaporating pressure P_e’ which is lower than the first cycle.

(a) Effect on RE (Refrigerating Effect):

$RE_1 = (h_1 – h_3)$ kJ/kg (1st Cycle)

$RE_2 = (h_1' – h_4')$ kJ/kg (2ed Cycle)

So from P-H chart it can be said that $RE_1 > RE_2$

Thus it is clear that lower evaporating/suction pressure/temperature decreases RE per kg of refrigerant circulated in the system.

(b) Effect on work done (WD):

$WD_1 = (h_2 - h_1)$ kJ/kg (1st Cycle)

$WD_2 = (h_2' - h_1')$ kJ/kg (2ed Cycle)

So from P-H chart it can be said that $WD_2 > WD_1$

Thus, it is clear that lower evaporating/suction pressure/temperature increases the work of compression per kg of refrigerant circulated in the system.

(c) Heat removed at condenser (Q):

$Q_1 = (h_2 - h_3)$ kJ/kg (1st Cycle)

$Q_2 = (h_2' - h_3)$ kJ/kg (2ed Cycle)

It is clear from the P-H chart that $Q_2 > Q_1$.

Thus, lower evaporating/suction pressure/temperature increases the heat to be removed at the condenser.

(d) Co-efficient of Performance (COP):

It is obvious that if the work done is higher, then the COP is bound to decrease as it is numerator value in the calculation of COP. Lower evaporating temperature/pressure reduces COP of the system.

Therefore, it can be concluded that lower suction pressure/temperature is not desirable for the refrigeration system. It is recommended to operate vapour compression refrigeration system at highest possible evaporating pressure/temperature. However the evaporating pressure maintained is dependent on the requirement of temperature at the cold store.

5.3. EFFECT OF CONDENSING PRESSURE

The effect of condensing pressure is shown on P-H diagram in Fig. 5.2.

As shown in P-H diagram (Fig. 5.2), a vapour compression refrigeration system operates as 1-2-3-4 having evaporating pressure P_e while cycle 1-2'-3'-4' operates at higher condensing pressure p_e. In practical situation, such variations are taking place depending on the temperature of the cooling medium employed at the condenser.

(a) Effect on RE (Refrigerating Effect):

$RE_1 = (h_1 - h_4)$ kJ/kg (1st Cycle)

$RE_2 = (h_1 - h_4')$ kJ/kg (2ed Cycle)

From P-H chart it can be said that $RE_1 > RE_2$

Thus, it is clear that higher condensing temperature/pressure reduces refrigerating effect per unit weight of refrigerant circulated in the system.

(b) Effect on work done (WD):

$WD_1 = (h_2 - h_1)$ kJ/kg (1$^{st\ Cycle}$)

$WD_2 = (h_2' - h_1)$ kJ/kg (2$^{ed\ Cycle}$)

From P-H chart it clear that $WD_2 > WD_1$

Thus, higher condensing temperature/pressure increases the work of compression of the system.

(c) Heat removed at condenser (Q):

$Q_1 = (h_2 - h_3)$ kJ/kg (1$^{st\ Cycle}$)

$Q_2 = (h_2' - h_3')$ kJ/kg (2$^{ed\ Cycle}$)

Higher condensing pressure/temperature slightly increases the total heat to be removed at the condenser.

(d) Co-efficient of Performance (COP):

It is clear from the above analysis that higher condensing temperature/pressure adversely affects the COP of the system as refrigerating effect decreases and work of compression increases. Therefore, operation of the refrigeration plant at higher condensing temperature/pressure is not desirable. The power consumption of the plant which is operating at higher condensing temperature consumes more electrical power. The temperature of cooling medium used at the condenser as well as heat transfer rate in the condenser are very important factors affecting the condensing temperature/pressure of the system.

(1) **Effect of sub-cooling:**

The cooling of liquid refrigerant after condensation is called sub-cooling of refrigerant. Little sub-cooling always takes place in the condenser but for achieving higher degree of sub-cooing low temperature water or refrigerant is used. The effect of sub-cooling of liquid refrigerant is shown on P-H diagram in Fig. 5.3.

The sub-cooling of the refrigerant increases the refrigerating effect per kg of refrigerant circulated in the system without any change in the work of compression. Therefore, sub-cooling is always desirable to reduce the operating cost of the system.

(1) **Effect of super heating:**

The increase in temperature of refrigerant after formation of saturated vapour at given suction pressure is known super heating of suction vapour. The super heating may take place with in the useful area (cold store/ice-bank tank) or out side the useful area. The cycle 1-2-3-4 is a saturated cycle while cycle 1'-2'-3-4 operates with suction vapour super heated.

It appears from the P-H diagram (Fig. 5.4) that super heating useful cooling improves the COP of the system. However, excessive superheating is not desirable as it increases the volume of refrigerant to be pumped by the compressor and increases the temperature of gas leaving the compressor. Higher temperature of gas leaving the compressor creates problem of head cooling of compressor. This problem is more serious in ammonia plant as compared to R-22 refrigerant. Effective insulation of suction pipeline is very essential to reduce the level of superheating of suction gas. Little superheating is recommended for smooth running of compressor without liquid pumping but excessive superheating should be avoided in vapour compression refrigeration system.

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Lesson 6.

Mathematical Analysis of Vapour Compression Refrigeration System

6.1. INTRODUCTION

Refrigeration is a process of removing heat from a space to reduce the temperature below the immediate surrounding temperature. Thus, refrigeration refers the pumping of heat from lower to higher temperature. Heat can flow from higher to lower temperature but reverse heat flow requires input power according to second law of thermodynamics (Fig.6.1).

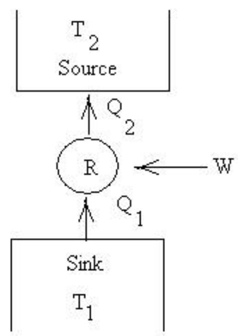

Fig. 6.1: Second law of thermodynamics applied to refrigeration system

In refrigeration system, it is essential to maintain temperature T_1 lower than T_2 (condensing temperature). The performance of refrigeration system as co-efficient of performance is given by

COP = <span style='font-family:"Cambria Math","serif";mso-ascii-font-family:"Cambria Math"; mso-hansi-font-family:"Cambria Math";font-style:italic;mso-bidi-font-style: normal'>Refrigeratiing effectWork of Compression"Refrigeratiing effect" /"Work of Compression"

And

Relative COP = Actual COP/Theoretical COP

Refrigeration cycle is reversed Carnot cycle used in engines.

Carnot COP= T_1 /(T_2- T_1)

Carnot cycle is the maximum possible for a refrigeration system operating between two temperatures.

Example: A reversed Carnot cycle is used to produce 1500 kJ/S to heat the space. The heat is taken from atmosphere at $10^{o}C$ and supplied to conditioned room at $25^{o}C$. Find the kW required to run the system.

Solution:

$$\text{Carnot COP} = \frac{T_1}{T_2 - T_1} = \frac{10+273}{25-10} = 18.9$$

$$\text{But COP} = \frac{Q_1}{W} = \frac{Q_2 - W}{W}$$

$$\therefore 18.9 = \frac{1500 - W}{W}$$

$$\mathbf{W = 75.38\ kW}$$

6.2. VAPOUR COMPRESSION REFRIGERATION SYSTEM

In vapour compression refrigeration system, heat is absorbed from the evaporator (Refrigeration effect, R) and work of compression (W) is also added in the system. Heat Q is rejected (Q) at the condenser considering no heat gain or loss any where in the system. Q=R+W.

The operating cycle of the system is represented on P-H, T-Q and P-V diagram in Lesson 4 and 5.

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Lesson 7.

Elementary Numerical on Refrigeration

1.0. Problems:

1.1. A 50 ton ammonia vapor compression refrigeration system operates an evaporating temperature of -10 °C and condensing temperature of 30 °C. There is no sub-cooling of liquid refrigerant and superheating of suction vapour. Determine (i) theoretical C.O.P. of the plant and (ii) operating cost of the plant, if the cost of electricity is Rs. 5.00 per kWh.

Solution:

Given data : Capacity of the plant : 50 ton
Refrigerant used : Ammonia (R717)
Evaporating Temperature : -10 °C
Condensing Temperature: 30 °C

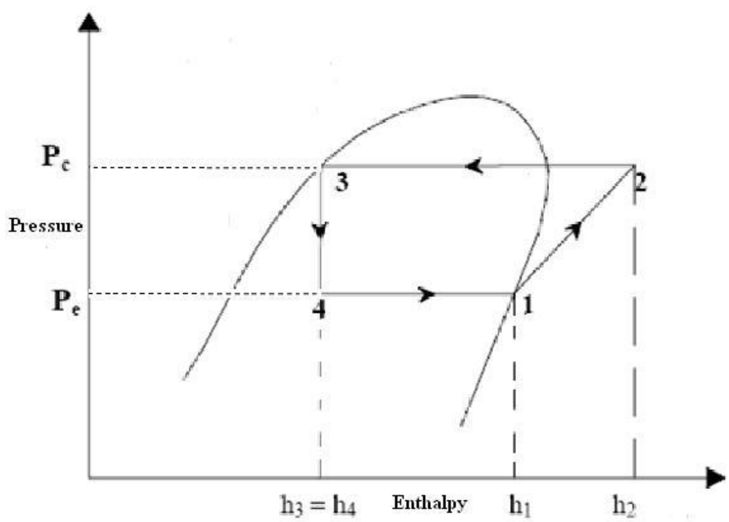

Fig.7.1: Chart

From the property table and chart:

h_1 = 1433.05 kJ/kg (From table)

h_2 = 1620.00 kJ/kg (From P-H chart)

$h_3 = h_4$ = 323.08 kJ/kg (From table)

$$\text{Theoretical COP} = \frac{\text{Refrigerat ing effect, kJ/kg}}{\text{Work of compressio n, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{1433.05 - 323.08}{1620 - 1433.05}$$

$$= \frac{1109.97}{186.95}$$

$$= \mathbf{5.937}$$

$$\text{Mass flow rate (MFR) of refrigerant in the evaporator} = \frac{210 \times 50}{1109.97}$$

$$= \mathbf{9.46\ kg/min}$$

Work of compression = MFR X Work done

= 9.46 X 186.95

=1768.547 kJ/min

$$= \frac{1768.547}{60}\ \text{kJ/s}$$

= 29.47 kW

Now the operating cost of the plant/ hour = 29.47 x 5

= Rs. 147.35

1.2. Find the capacity of vapor compression refrigeration system to be used for the bulk milk cooler from the following data.

i. Capacity of the bulk cooler: 1000 liters

ii. Initial temperature of supply milk: 37 °C

iii. Temperature to which milk is to be cooled: 2 °C

iv. Time for cooling: 3.5 hour

(Make necessary assumptions and indicate them clearly.)

Solution:

Given data : Capacity of the Bulk cooler (BC) = 1000 liters
Initial temperature of supply milk = 37 °C
Temperature to which milk is to be cooled = 2 °C
Time for cooling = 3.5 h

Assumptions made:

Specific gravity of milk = 1.032

Specific heat of milk = 3.89 kJ/kg °C

1 TR = 12600 kJ/h

Total heat to be removed from the milk = Q = m s ΔT

= (1000 ×1.032) × 3.89 × (37 - 2)

= 140506.8 k J

Heat removed / hour = 140506.8 / 3.5

= 40144.8 kJ/h

Hence, Capacity of refrigeration system required = 40144.8 / 1260

= 3.186 TR

1.3. A water cooler using R-22 as refrigerant works between -5^{0}C evaporating temperature and 45^{0}C condensing temperature. The output of cold water is 200 kg per hour cooled from 40^{0}C to 10^{0}C. Assuming 30 per cent loss of useful cooling and 80 per cent volumetric efficiency of the compressor, calculate compressor power in kW and volumetric displacement of the compressor.

Solution: Given data

Refrigerant used = R-22

Evaporating Temperature = - 5 °C

Condensing Temperature = 45 °C

Output of cold water = 200 kg/h

Initial temperature of water = 40 °C

Final temperature of the water = 10 °C

Cooling loss = 30 %

Volumetric efficiency of compressor = 80 %

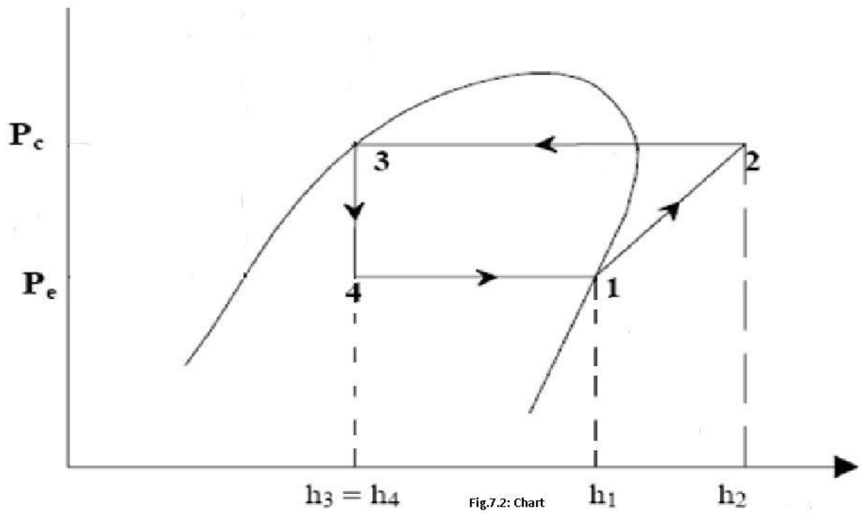

Fig.7.2: Chart

From the property table and chart:

h_1 = 249.88 kJ/kg (From table)

h_2 = 290 kJ/kg (From P-H chart)

$h_3 = h_4$ = 105.58 kJ/kg (From table)

$$\text{Theoretical COP} = \frac{\text{Refrigerating effect, kJ/kg}}{\text{Work of compression, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{249.88 - 105.58}{290 - 249.88}$$

$$= \frac{144.38}{40.12}$$

$$= 3.6$$

Total heat to be removed from the water, $Q = m\ s\ \Delta T$

$$= 200 \times 4.186 \times (40\text{-}10)$$

$$= 25116 \text{ kJ/h}$$

Now 30 % cooling is lost so the total refrigeration required $= 25116 + (0.3 \times 25116)$

$$= 25116 + 7534.8$$

$$= \mathbf{32650.8}$$

Hence, Capacity of refrigeration system required $= \dfrac{32650.8}{12600}$

$$= \mathbf{2.59\ TR}$$

Mass flow rate (MFR) of refrigerant in the evaporator $= \dfrac{210 \times 2.59}{144.38}$

$$= \mathbf{3.76\ kg/min}$$

Work of compression = MFR X Work done

= 3.76 X 40.12

= 150.85 kJ/min

Hence compressor power $= \dfrac{150.85}{60}$

$$= \mathbf{2.514\ kW}$$

Specific volume of R-22 at -5 °C is 0.05545 m^3/kg

Considering 80% volumetric efficiency,

The volumetric displacement = 3.76×0.05545× (100/80)

= 0.2606 m^3/min

1.4. An ammonia vapour compression plant is to be selected for a milk chilling centre for the following requirements. Estimate the capacity, theoretical C.O.P. and kW of the compressor for the refrigeration plant. (Make appropriate assumptions and specify them clearly.)

(i) Milk handling capacity of chilling centre = 60,000 litres/day.

(ii) Condensing pressure = 12 bar

(iii) Evaporating pressure = 2 bar

(iv) Temperature of supply milk = 35 ^{0}C

(v) Working hours of the plant = 18 hours/day

Solution:

Assuming the milk is cooled to 2^{0}C

TR required for the chilling centre = m s ΔT

= (60000×1.032)×3.89×(35-2)

=7948670.4 kJ/day

TR of the plant = 7948670.4 / 18 X 12600

=35.04 TR

Considering 20% refrigeration losses,

Capacity of the plant = 35.04+ (35.04×0.2)

= 42 TR

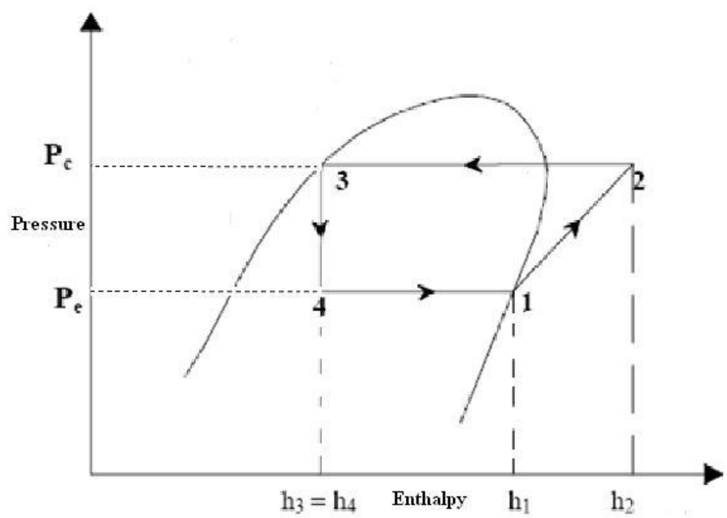

Fig.7.3: Chart

From the property table and chart: h_1 = 1422.72 kJ/kg (From table)

h_2 = 1660 kJ/kg (From P-H chart)

$h_3 = h_4$ = 327.89 kJ/kg (From table)

$$\text{Theoretical COP} = \frac{\text{Refrigerating effect, kJ/kg}}{\text{Work of compression, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{1422.72 - 327.89}{1660 - 1422.72}$$

$$= \frac{1094.83}{237.28}$$

$$\mathbf{= 4.61}$$

$$\text{Mass flow rate (MFR) of refrigerant in the evaporator} = \frac{210 \times 42}{1094.83}$$

$$\mathbf{= 13.81\ kg/min}$$

$$\text{Work of compression} = \text{MFR X Work done}$$

$$= 13.81 \text{ X } 237.28$$

$$\mathbf{= 3276.92\ kJ/min}$$

$$\text{Hence compressor power} = \frac{3276.92}{60}$$

$$\mathbf{= 54.62\ kW}$$

1.5. An ammonia refrigeration plant is working between the temperature limits of 30 ^{0}C and -15^{0}C. The load on the unit is 20 ton. Find the (i) Theoretical C.O.P. and (ii) kW of the compressor. If the temperature required in the evaporator is -30^{0}C, then find out the change in theoretical C.O.P. and kW of the compressor. There is no change in condensing temperature.

Solution:

Given data: Refrigerant used: Ammonia (R-717)

Evaporating Temperature: -15^{0}C and -30^{0}C

Condensing Temperature:30 ^{0}C

Capacity of the plant: 20 TR

At operating conditions of -15⁰C and 30 ⁰C:

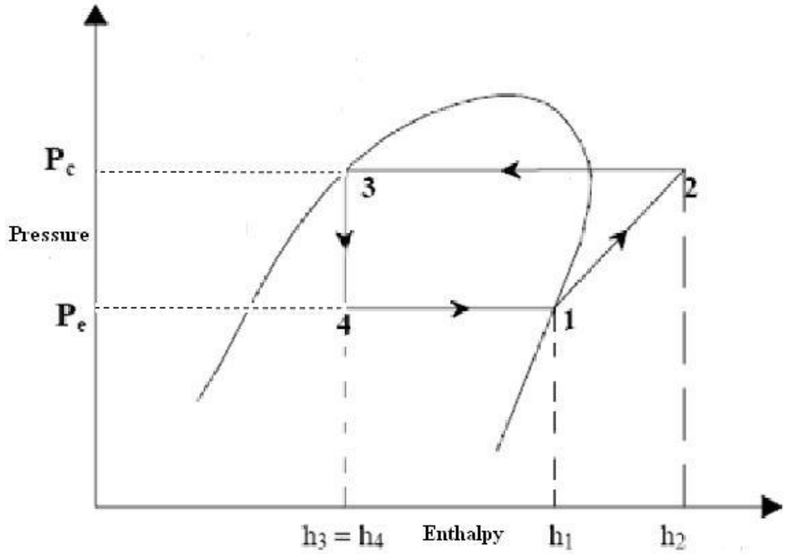

Fig.7.4: Chart

h_1 =1426.58 kJ/kg (From table)

h_2 =1640 kJ/kg (From P-H chart)

h_3 =323.08 kJ/kg (From table)

$$\text{Theoretical COP} = \frac{\text{Refrigerating effect, kJ/kg}}{\text{Work of compression, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{1426.58 - 323.08}{1640 - 1426.58}$$

$$= \frac{923.5}{213.42}$$

$$\mathbf{= 4.33}$$

$$\text{Mass flow rate (MFR) of refrigerant in the evaporator} = \frac{210 \times 20}{923.5}$$

= 4.55 kg/min

Work of compression = MFR X Work done

= 4.55 X 213.42

= 970.62 kJ/min

$$\text{Hence, compressor power} = \frac{970.62}{60}$$

= 16.18 kW

At operating conditions of -30⁰C and 30 ⁰C:

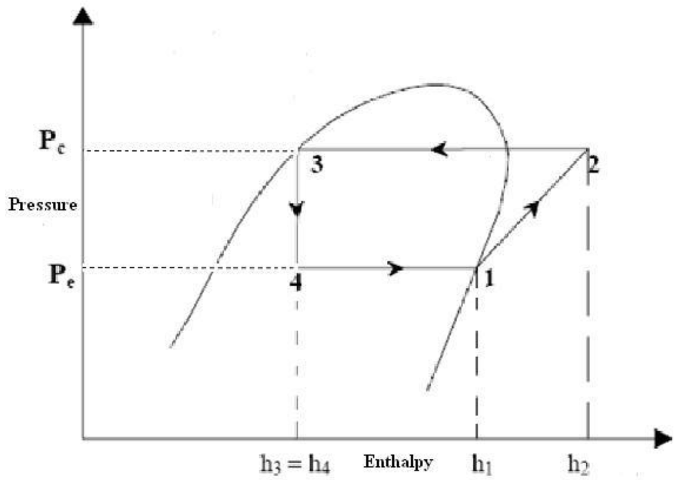

Fig.7.5: Chart

h_1 =1405.6 kJ/kg (From table)

h_2 =1742 kJ/kg (From P-H chart)

h_3 =323.08 kJ/kg (From table)

$$\text{Theoretical COP} = \frac{\text{Refrigerating effect, kJ/kg}}{\text{Work of compression, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{1405.6 - 323.08}{1742 - 1405.6}$$

$$= \frac{1082.52}{336.4}$$

$$= \mathbf{3.22}$$

$$\text{Mass flow rate (MFR) of refrigerant in the evaporator} = \frac{210 \times 20}{1082.52}$$

$$= \mathbf{3.88\ kg/min}$$

Work of compression = MFR X Work done

= 3.88 X 336.4

= 1305.23 kJ/min

$$\text{Hence, compressor power} = \frac{1305.23}{60}$$

$$= \mathbf{21.75\ kW}$$

$$\%\ \text{Change in COP} = \frac{4.33 - 3.22}{4.33} \times 100$$

$$= \mathbf{25.64\ \%}$$

$$\%\ \text{Change in kW} = \frac{21.75 - 16.18}{16.18} \times 100$$

$$= \mathbf{34.43\ \%}$$

1.6. A single cylinder single acting R-22 compressor works between -10 ^{0}C and 40 ^{0}C. The vapour leaves the evaporator dry and saturated. Find (i) capacity (ii) kW of motor from the following data.

(i) Diameter of the compressor cylinder = 15 cm.

(ii) Stroke of the compressor = 15 cm.

(iii)R.P.M. of the compressor =850.

(iv)Volumetric efficiency of the compressor =75 %

(v)Mechanical efficiency of the compressor = 96%

(vi)Motor efficiency = 97%

Solution:

$$\text{Vapour displacement of the compressor} = \frac{\pi}{4} \times d^2 \times l \times rpm \times no.\ of\ acting \times \eta_v$$

$$= \frac{3.14}{4} \times (0.15)^2 \times 850 \times 1 \times 0.75$$

$$= 11.26\ m^3/min$$

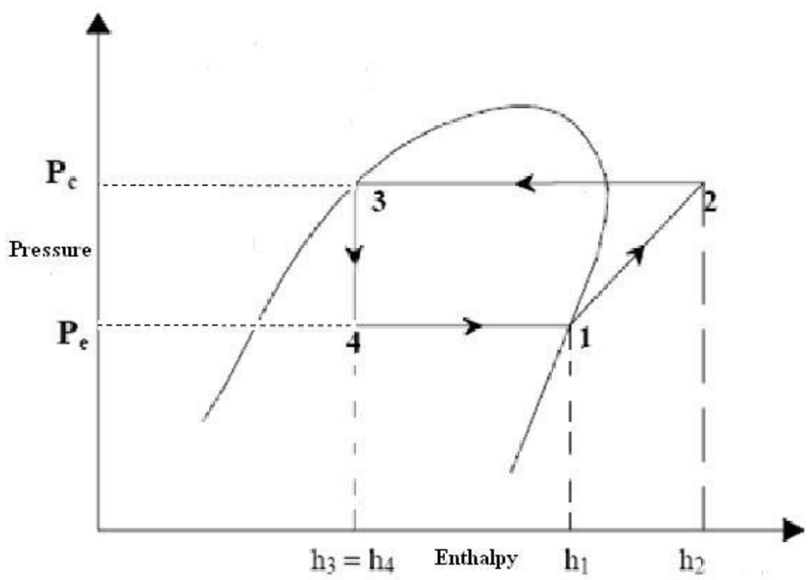

Fig.7.6: Chart

h_1 = 247.72 kJ/kg (From table)

h_2 = 285 kJ/kg (From P-H chart)

h_3 = 98.44 kJ/kg (From table)

$$\text{Theoretical COP} = \frac{\text{Refrigerating effect, kJ/kg}}{\text{Work of compression, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{247.72 - 98.44}{285 - 247.72}$$

$$= \frac{149.28}{37.28} \therefore$$

$$= \mathbf{4.004}$$

At -10 ^{0}C, the specific volume of saturated R-22 vapour is 0.0655 m^3/kg

$$\therefore \text{Mass Flow Rate} = \frac{11.26}{0.0655}$$

$$= 171.908 \text{ kg/min}$$

$$\text{Capacity of the system in TR} = \frac{171.908 \times 149.28}{210}$$

$$= \mathbf{122.2\ TR}$$

kW of compressor = MFR X Work done

= 171.908 X 37.28

= 6408.73 kJ/min

$$= \frac{6408.73}{60} \text{ kW}$$

$$= \mathbf{106.8\ kW}$$

$$\text{kW of input motor} = 106.8 \times \frac{100}{96} \times \frac{100}{97}$$

$$= \mathbf{114.69\ kW}$$

1.7. A 10 ton Ammonia vapour compression refrigeration system operates at suction temperature of -10 ^{0}C and condensing temperature of 35 ^{0}C. There is no sub cooling and superheating of suction vapour. Calculate the theoretical COP, kW and temperature of gas leaving the compressor (Uses only refrigerant property data).

Solution: Data given:

Refrigerant used: Ammonia(R-717)

Capacity of the plant: 10 TR

Evaporating temperature: -10 ^{0}C

Condensing temperature: 35 ^{0}C

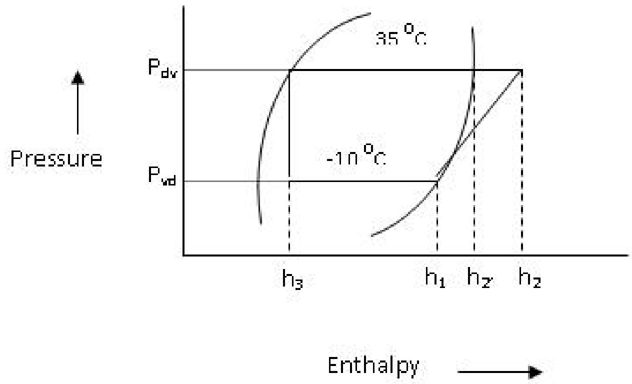

Fig.7.7: Chart

h_1 = 1433.05 kJ/kg (From table)

h_3 = 347.5 kJ/kg (From table)

$h_{2'}$ = 1471.43 kJ/kg (From table)

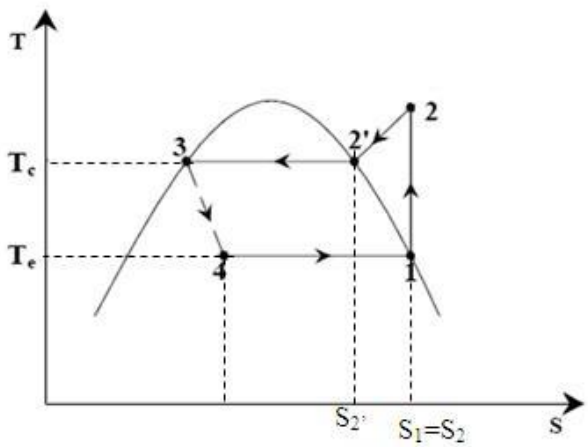

Fig.7.8: Chart

S_1 = 5.477 kJ/kg K (From table)

$S_{2'}$ = 4.93 kJ/kg K (From table)

Taking specific heat of ammonia as 2.8 kJ/kg K, the enthalpy h_2 can be obtained as under:

$S_1 = S_2 = S_{2'} + Cp \log_e\left(\frac{T_2}{T_2'}\right)$

$5.477 = 4.93 + 2.8 \log_e \frac{T_2}{35+273}$

$\frac{5.477-4.93}{2.8} = \log_e \frac{T_2}{308}$

$0.195 = \log_e \frac{T_2}{308}$

$\frac{T_2}{308} = 1.22$

$T_2 = 1.22 \text{ X } 308$

$= 374.45 \text{ K } (101.45^0\text{C})$

$h_2 = h_{2'} + Cp(T_2 - T_{2'})$

$= 1471.43 + 2.8(374.45 - 308)$

$= 1657.5 \text{kJ/kg}$

$$\text{Theoretical COP} = \frac{\text{Refrigerating effect, kJ/kg}}{\text{Work of compression, kJ/kg}} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$= \frac{1433.05 - 347.5}{1657.5 - 1433.05}$$

$$= \frac{1085.55}{244.44}$$

= 4.8

Mass flow rate (MFR) of refrigerant in the evaporator $= \frac{210 \times 10}{1085.55}$

= 1.93 kg/min

Work of compression = MFR X Work done

= 1.93 X 244.44

= 471.77 kJ/min

Hence, compressor power $= \frac{471.77}{60}$

= 7.85 kW

Temperature of gas leaving the compressor = **101.45^{0}C**

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MODULE 2 - REFRIGERANTS

Lesson 8. Definition and Classification of refrigerants

8.1. INTRODUCTION

- Any substance that absorbs heat through evaporation is known as refrigerant. It is a medium of heat transfer which absorbs heat by evaporation at low temperature and gives up heat by condensing at high temperature and pressure.
- Broader definition includes secondary cooling mediums like cold water, brine solutions
- Widest applications at present are ammonia or halocarbons.

8.2. DESIRABLE CHARACTERISTICS OF REFRIGERANT:

1. Non poisonous
2. Non corrosive
3. Non inflammable
4. Leaks easily detectable
5. Low boiling point
6. Stable gas
7. Suitable latent heat
8. Low specific volume
9. Minimum difference between evaporating & condensing pressure, one further classification of derivable properties; they can be categorized as thermodynamic, physical and chemical properties.

8.3. THERMODYNAMIC

1. Normal boiling point : should be low at atmospheric pressure

2. Freezing point : should be low

3. Evaporations & condensing pr : should be +ve and as near to atmospheric pressure

4. Critical temperature & pressure : should be high (ex.co_2p n it is just 30°c)

5. Latent heat: should be high

6. Specific heat : low for liquid, high for vapour

7. Specific volume: low for vapour

8.4. PHYSICAL

1. Dielectric strength : high for use in hermetic compression
2. Thermal conductivity : should be high
3. Viscosity : should be low
4. Leak tendency : detection of leak should be easy
5. Flammability: should not be flammable within range of conditions of operation. Nh_3 flammable between 16 to 27% in air

8.5. CHEMICAL

1. Toxicity : should not be toxic
2. Corrosion properties: should not react in the commonly used metals and materials.

8.6. CLASSIFICATION OF REFRIGERANTS (NRSC, USA)

Group-1: safest: R-11, R-12, R-22, R-744(co_2), R-500, R-502

Group-2: toxic& slightly inflammable: R-717, R-764(so_2), R-113

Group-3: inflammable: R-600(butane),R-170(ethane),R-290(propane)

(natural refrigerants: h_2o, air, nh_3, propane, butane)

8.7. COLOR CODE FOR THE CYLINDERS & COMPOSITION

R-717: (SHINY) Silver NH_3

R-11: ORANGE: (CCL_3F) Trichloromonofluoro Methane

R-12: WHITE: (CCL_2F_2) Dichlorodifluoromethane

R-22: GREEN : ($CHCL\ F_2$) Monochlorodi Fluoromethane

8.8. NUMBERING SYSTEM BY ASHRAE

(1) FOR SATURATED HYDRO-CARBON

R (M-1)(N+1)(P) WHERE,

M= No Of Carbon Atoms

N= No Of Hydrogen Atoms

P= No Of Fluorine Atoms

(2) FOR SATURATED COMPOUNDS

Digit 1 kept before (M-1)

EG: ETHYLENE R-1150

(3) INORGANIC REFRIGERANTS

Add molecular weight To 700

EG: NH_3 (17+700)

CO_2 (44+700)

(4) AZEOTROPES: They are mixtures of refrigerants arbitrary nos

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Lesson 9.

Properties of refrigerants and comparison

9.1. AMMONIA (REFRIGERANT)

- Boiling point at Atmospheric Condition is - 33^{0}C
- Possible to use for below 0 ^{0}C application
- It has very large latent heat
- Condenser is almost always of water cooled type.
- Some what inflammable with proper mixture of air.
- Very pungent smell
- Forms dense white fumes in presence of sulphour candle.
- It attacks copper and bronze in presence of moisture but does not corrode iron or steel.
- It is lighter than oil, and is easily separated from it
- It is extremely soluble in water.
- Used with large refrigeration systems, using reciprocating compressions.

9.2. R-22 ($CHCl\ F_2$)

v Mono chlorodifluoro methane

v Low evaporating temperature

v At atmospheric pressure B.P - 41^{0}C

v Stable, non-tonic, non- corrosive, non – irritating and non- inflammable

v Water is should be in it - there for driers and desiccants

v Leak detected by Halide Torch or electronic detector

v Phase out by 2020 in developed countries and 2030 in developing countries (Montreal 1987)

v Much faster phase out is being done (EU in 2004

Table. 9.1: Properties of refrigerants

Refrigerant	Boiling Point At Atm P,T)	Evaporator Pressure (at-15^{0}C) kgf/cm^{2}a	Condenser Pressure (30^{0}C) kgf/cm^{2}a	C. R.	Ref.Eff k.cal/kg	Mass flow Rate/TR kg/min	Sp.Vol. (-15^{0}C)	H.P/ TR	COP	Application	Relative Dielecticism comss
R-11 (CCl_3F)	+23.7	0.2055	1.2855	6.27	37.51	1.335	0.77	0.93	5.09	Large Centrifugal compressors	3.0
R-12 (CCl_2F_2)	-29.8	1.8622	7.5810	4.08	28.32	1.765	0.09	1.01	4.71	Wide range: All temp, All comp	2.4
R-22 ($CHClF_2$)	-40.8	3.03	12.26	4.04	38.46	1.3	0.08	1.02	4.66	Low Temp. & Small cap.	1.31
R-717	-33.3	2.41	11.895	4.94	264.28	0.1895	0.51	0.998	4.76		0.82
R-134 a $C_2H_2F_4$	-27										

9.3. OZONE LAYER & UV RAYS

· OZONE (O_3) is a bluish gas, harmful to breath.

· 90% of Earth's Ozone is in stratosphere (15 to 50 km) forming a layer of 2 to 5mm

thick under normal temperature & pressure.

· Its concentration varies with season, hour of day and location.

· Its concentration is greatest at 25 km from Equator & 16 km from poles.

· Ozone comes mostly from photo-disassociation of oxygen by UV Rays of very short wave length.

· Dobson is unit of level of ozone.

UV rays are again classified according their wave lengths.

UV-A 320-400 nm: Not absorbed by ozone

UV-B 280-320 nm ozone protects against most of UVB. Harmful because, it damages DNA, Melanonia and other skin cancer, damages some types of materials, crops and marine organisms.

UV-C <280 nm despite being extremely dangerous, it is completely absorbed by ozone and normal O_2.

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Module 3 - Refrigeration Plant Components

Lesson 10.

Reciprocation Compressor- Construction, Working and Maintenance

10.1. INTRODUCTION

The compressor is referred as the heart of the vapour compression refrigeration system. The function of compressor is to suck the refrigerant gas from low pressure side and to compress it to discharge pressure so that condensation of gas can be done either by water or air at ordinary room temperature. The compressor also keeps low pressure in the evaporator for efficient evaporation of the refrigerant. There are three main groups of compressors.

1. Reciprocating compressor
2. Rotary compressor
3. Centrifugal compressor

The reciprocating compressor consists of a piston moving back and forth in the cylinder with suction and discharge valve arranged to allow pumping to take place.

The rotary and centrifugal compressors have rotating members but the rotary compressor has a positive displacement where as a centrifugal compressors draws the vapour and discharges it at high pressure by centrifugal force.

10.2. RECIPROCATING COMPRESSOR

Reciprocating compressors consist of one or more cylinders with suitable valves for suction and discharge of the refrigerant gas. The compression of suction gas is achieved by reciprocating pistons. The design of the cylinders may vary depending upon the number of cylinders, arrangement of cylinders and acting (single acting or double acting). Reciprocating compressors are widely used in dairy plants and these compressors are driven by the electric motor. Reciprocating compressors are classified as under:

1. Open type

2. Semi hermetic compressors

3. Hermetic compressors

10.3. OPEN TYPE RECIPROCATING COMPRESSOR

An open type reciprocating compressor consists of cylinders in which piston moves back and forth for suction and discharge of the refrigerant vapour. The main parts are piston, cylinder, connecting rod, crankshaft, cylinder head, crank case, suction and discharge valve etc. In open type of reciprocating compressor, power is received from an external source with one end of the crank shaft extending through the crank case to either a direct motor drive or a V-belt, gear or chain drive. With such a design, it is necessary to seal the crank case against the leakage of refrigerant.

On the down stroke of the piston, the low pressure is created between the top of the piston cylinder head and the suction side of the evaporator. This causes the refrigerant vapour to run in to the cylinder. On the discharge stroke of the piston, the gas is compressed and discharged to the out let of the compressor. The valves in the cylinder head are so designed that, depending on the position of the stroke, one is open while the other is closed.

10.4. COMPRESSOR CYLINDERS

The number of cylinders varies from one to as many as sixteen. In multi cylinder compressors, the cylinders may be arranged in line, radically or at an angle to each other to form a V or W pattern. Compressor cylinders are usually constructed of close-grained cast iron which is easily machined. For small compressors, the cylinders and crank case housing are often cast in one piece while for large compressors the cylinders and crank case housing are usually cast separately which is flanged and bolted together. The cylinders of large compressors are usually equipped with replacement liners or sleeves.

Small compressors often have fins cast integral with the cylinders and cylinder head to increase cylinder cooling, whereas large compressors contain water jacket for this purpose.

10.5. PISTONS

Pistons employed in refrigeration compressors are of two types (a) automotive and (b) double trunk.

Automotive type pistons are used when the suction gas enters the cylinder through suction valves located in the cylinder head. Double trunk pistons are used when the suction gas enters through ports in the cylinder wall and in the side of the piston and passes into the cylinder through suction valves located in the top of the piston. These pistons are provided with piston rings. The pistons are manufactured from close-grained cast iron.

10.6. DESIGN OF VALVES

Generally two types of valves are used in compressors.

1. Non flexing ring plate type valves

2. Flexible or Reed type valves.

All valves operate upon pressure differential, and there are many modification of each class. These valves are designed to open easily and close quickly.

10.5.1. Ring plate type valves

The ring plate is a thin ring which is held closed in the top of the cylinder by spring. The valve is held closed by small springs. When the refrigerant vapour pressure inside the cylinder is greater than the spring tension, the valve opens on the up stroke of the piston to allow vapour to pass through the large discharge ports to the discharge outlet.

The suction valve opens on the down stroke of the piston because the cylinder pressure is less than the vapour pressure in the suction line. On the up stroke of the piston the suction valve closes and the pressure within the cylinder causes the discharge valve to open. Valves are manufacture from specially heat-treated alloys. The noise level during operation can be reduced by putting a plastic material installed in the valve.

10.5.2. Flexing type valves:

Small modern refrigeration compressors use high grade steel reed or disc valves. These valves are quiet, simple and long lasting. For these reasons, they are especially adopted to high speed compressors.

10.6. COOLING OF COMPRESSOR HEAD

Temperature rise of compressor head is controlled by cooling the upper part of the cylinder walls and cylinder head to prevent the over heating of cylinder head. Cooling of compressor head prevents the damage to head, piston and rings. The cooling is done in ammonia compressor by jacketing a cylinder wall and head through which water is circulated. In case of R-12 or R-22 refrigerant, the discharge temperature will be lower than ammonia compressor and hence generally fines are provided which facilitates the transfer of heat to the surrounding air.

10.7. SHAFT SEAL

Open compressors are made with the crankshaft extending through the crank case for connecting with motor, v-belt, gear or chain drive. A shaft seal is required to prevent refrigerant leakage. The leakage may take place under both static and moving conditions at the point at which the crank shaft passes through the housing.

10.8. CRANK SHAFT AND BEARING

Crank shaft employed in large compressors are usually constructed of forged steel. The eccentric type shaft which consists of a cast iron eccentric mounted on a straight steel shaft is often used in smaller compressors. It is fastened for the shaft by a key and lock screw arrangement. Bearing may be of sleeve type or antifriction type and the crank shaft is made up of drop forged steel.

10.9. LUBRICATION

Proper lubrication is essential for the compressors. It is essential to know the interval of lubrication, type of lubricant, quantity of lubricant etc. In some compressors force feed system is used for the circulation of oil. The oil pump takes suction from the crank case through filters and discharge oil vertically in the passage in the compressor. It is also necessary to know the manufactures instructions for the lubrication requirement.

Two systems, splash lubrication and forced lubrication are common at present. In splash lubrication with each rotation of the shaft the crank and connecting rod dip into the crank case oil reservoir and thereby splash the lubricant in to openings. As a general rule, small vertical compressors (up to 10 kW) are splash lubricated and above this size, most compressors employ some type of forced feed lubrication.

10.10. HERMETICALLY SEALED COMPRESSOR

The hermetic type of compressor may be either reciprocating or rotary and is a direct drive unit with both motor and the compressor hermetically sealed with in the housing. It is used in domestic refrigerator, air conditioner and other small capacity systems. This arrangement eliminates the necessity of many shaft seals. Some advantages of hermetically sealed compressors are listed below.

1. It prevents the leakage of refrigerant.
2. It also reduces the operating noise considerably.
3. It eliminates external drive
4. Lubrication is greatly simplified
5. Motor operates in an ideal atmosphere
6. Cooling of motor is done by suction gas

10.10 Volumetric efficiency of reciprocating compressor:

The volumetric efficiency of reciprocating compressor is defined as the ratio of actual volume of refrigerant gas delivered on each stokes to the piston displacement of the compressor. Higher volumetric efficiency of the compressor is desirable to get higher actual capacity of the refrigeration system. The actual volume of gas handled by the compressor is normally less than the piston displacement of the compressor due to several factors.

10.11 Factors affecting the volumetric efficiency of reciprocating compressor:

a) **Clearance volume:** The volume of the cylinder between the top of the piston and the delivery valve plate when the piston is at top dead center is known as clearance volume. Clearance volume is necessary, though it should be minimum, to prevent damage to the valves. The refrigerant gas present in the clearance volume will expands on the down ward movement of the piston. There will not be any intake of suction gas until the pressure of the cylinder gas drops below the suction pressure of the refrigeration system. The clearance volume is expressed as % of the total volume. It is about 2-4 % of the cylinder volume. Higher cylinder volume reduces the volumetric efficiency of the reciprocating compressor.

b) Compressor ratio: Higher compression ratio reduces the volumetric efficiency of the compressor. Higher compression ratio either due to higher discharge pressure or lower suction pressure. This requires more piston displacement in order to expand the refrigerant gas present in the clearance volume for the intake of suction gas. Therefore, lower compression ratio during operation of the refrigeration system is desirable to achieve higher volumetric efficiency of the compressor.

c) Leakage through Valves: Any leakage of gas through suction valve or delivery valve will reduce the volumetric efficiency of the compressor. It is obvious that the gas filled in the cylinder leaks back either on suction side or from delivery side to the cylinder will reduce the volumetric efficiency of the compressor.

d) Cylinder heating effect: Higher temperature of cylinder wall increases the volume of the gas which intern reduces the intake of suction gas. Therefore, effective head cooling of the compressor is important to avoid reduction in volumetric efficiency of the compressor due to cylinder heating effect.

e) Wire drawing effect: The restriction of area for the flow of refrigerant which causes pressure drop of the refrigerant is known as wire drawing effect. When refrigerant gas enters through suction valve of the compressor, it experiences wire drawing effect. The extent of pressure drop depends on the velocity of the refrigerant, type of refrigerant and valve design. Wire drawing effects adversely the volumetric efficiency of the compressor.

10.12. MAINTENANCE OF RECIPROCATING COMPRESSORS

It is very important to carry out maintenance (preventive and break down maintenance) of compressor by skilled personals. The following points must be considered for effective maintenance of the compressors.

1. Regular lubrication programme considering the recommendations of the manufacturers.

2. Replacement of piston rings, piston, liner, shaft seal, valves etc. if required.

3. Regular checking of bolts, nuts, foundation, etc.

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Lesson 11.

Rotary, Centrifugal Compressors and Screw Compressor

11.1. INTRODUCTION

The different types of compressors used in refrigeration plant includes rotary, centrifugal compressors and screw compressor

11.2. SCREW COMPRESSOR

It consists of two helical rotors which are installed in a common housing. The refrigerant vapour is drawn into the gaps between the teeth on rotation of the rotors. The gas is gradually compressed along the length of the rotors. These compressors belong to category of positive displacement compressor.

Screw compressors are used in large installations. One of the greatest advantages of the screw compressor is that the capacity can be varied down to 10 % of full load without excessive loss of efficiency. The working of screw compressor is indicated in Fig. 11.1. It is also reported that screw compressor consumes little less energy as compared to reciprocating compressor. It also requires less maintenance as compared to reciprocating compressor. These compressors are compact in design and require less maintenance. PLC based compressor control facilitates automation for higher efficiency.

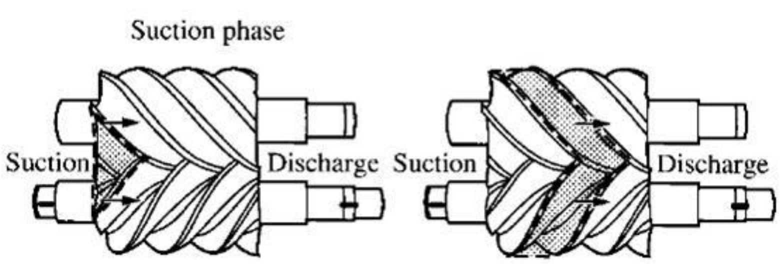

Fig. 11.1 Working of screw compressor

11.3. CENTRIFUGAL COMPRESSOR

The centrifugal compressor is generally used for refrigerants that require large vapour displacement and low condensing pressure. Single stage centrifugal compressor can develop compression ratio to about 4.5. For higher compression ratio, multi stage centrifugal compressors with inter cooling are employed.

The impeller draws low pressure vapour from evaporator. The high speed of impeller leaves the vapor refrigerant at high velocity at the vane tips of the impeller. The volute casing collects the refrigerant vapour from the diffuser and it further converts kinetic energy into pressure energy.

Centrifugal compressor requires less maintenance due to less number of moving parts.

11.4. ROTARY COMPRESSOR

These compressors are positive displacement type and the refrigerant is compressed due to the movement of blades. There are two types of rotary compressors. The whole assembly of both the types of rotary compressor is enclosed in a housing which is filled with oil.

11.4.1. Single stationary blade type rotary compressor

This type of compressor consists of a stationary cylinder, a roller and a shaft having eccentric. A blade (sealing blade) is set into the slot of the cylinder in such a manner that it maintains contact with the roller by means of a spring. When roller rotates, the vapour refrigerant ahead of the roller is being compressed and the new intake of gas takes place.

11.4.2. Rotating blade type rotary compressor

This consists of a cylinder and a slotted rotor containing a number of blades. The centre of the rotor is eccentric with the centre of the cylinder. The blades are forced against the cylinder wall by the centrifugal action during the rotation of the motor. As the rotor turns, the suction vapour entrapped between the two adjacent blades is compressed and is discharged through the discharge port.

Lesson 12.

Condenser: Types, Construction, Working and Maintenance

12.1. INTRODUCTION

Condenser is a heat exchanger in which heat transfer from refrigerant to a cooling medium takes place. The heat from the system is rejected either to atmosphere air or to the water used as cooling medium. The water used as cooling medium which in term rejects the heat to the atmosphere. For the steady state operation, heat rejected in the condenser is the sum of heat absorbed by the evaporator and heat equivalent of work supplied to the compressor. On account of heat transfer in the condenser refrigerant passing through it is first de superheated and then condensed and may be little sub-cooled. Thus, the function of the condenser is to convert superheated refrigerant vapour into liquid refrigerant. The cooling medium used may be air, water or combination of air and water depending on the type of condenser employed in the refrigeration system.

12.2. CONDENSER LOAD

The amount of heat rejected or transfer by the condenser is termed as condenser load or condenser capacity.

Condenser capacity (kJ/s)	=	Mass flow rate of refrigerant, kg/s	×	Enthalpy change of refrigerant while passing through the condenser, kJ/kg

Based on the amount of heat to be rejected at the condenser, the flow rate of cooling medium required can be estimated as under.

$$\text{Amount of cooling medium required, kg/s} = \frac{\text{Heat to be removed, kJ/s}}{\text{Difference between the enthalpy values of the leaving and entering cooling medium, kJ/kg}}$$

The condensing pressure of the refrigeration system mainly depends on the temperature of cooling used in the condenser. It is desirable to use low temperature cooling medium in the condenser to get lower condensing pressure. This is desirable to get better COP of the refrigeration plant. Where the entering temperature of the condensing medium is relatively high, larger surface area of the condensers and higher flow rates are required to provide lower condensing temperatures than where the entering temperature of the condensing medium is lower.

12.3. TYPES OF CONDENSER

There are three types of condensers commonly used in vapour compression refrigeration system.

1. Air cooled condenser

2. water cooled condenser

3. Evaporative condenser

12.3.1. Air cooled condensers

The atmospheric air is used as a medium of heat transfer in air cooled condenser. The heat rejected by the refrigerant is received by the air. The air circulation over an air cooled condenser may be either natural convection or by the action of blower or fan. Accordingly, they are classified as natural draft or mechanical draft condenser. The air cooled condenser consists of finned tubing of copper or other suitable metal in which the vapour of the refrigerant enters from the top and the liquid refrigerant leaves from the bottom of the condenser. The heat transfer area, temperature of the air, velocity of the air, overall heat transfer co-efficient etc. are important parameters affecting the performance of the condenser. This type of condenser is used for relatively small capacity system as heat rejection rate per unit area of the tube is less as compared to other type of condensers. The air velocities normally employed are 2 m/s to 6 m/s.

In case of natural convection air cooled condenser, the air quantity circulated over the condenser is low and hence condensing surface required is relatively larger as compared to mechanical draft air cooled condenser. Generally, forced convection type air cooled condenser requires 10-15 m^2 surface area per ton of refrigeration considering 2-5 m/s face velocity of air over the coil. The air is blown or drawn through the condenser by a propeller type fan. Natural air cooled condensates are either plate surface or finned tubing or copper or steel or other metal depending on the refrigerant used. The condenser should be located in a well ventilated and cool space where sufficient quantity of air is available. The sketch of mechanical draft air cooled condenser and the photograph of a mechanical draft air cooled condenser are shown in Fig. 3.2 and 3.2 respectively.

The design of the condenser is relatively simple, it's operating and maintenance cost is also low. It is necessary to remove dust, lint etc. settled on the condenser by using a portable blower to maintain its heat transfer performance

Air cooled condensers are used in small capacity refrigeration systems such as window air conditioners, water coolers, split air conditioners etc. as the overall heat transfer co-efficient is low as compared to other types of condensers. The capacity of an air cooled condenser depends on heat transfer area, temperature difference, air velocity and overall heat transfer co-efficient between refrigerant and cooling air.

· Air cooled condensers are generally designed for condensing temperatures of 15 ºC to 20 ºC above the atmospheric temperature and the air quantity of 20 to 30 m^3 /ton.

· The condenser is mounted at a higher level than the compressor and provided with oil loop at the compressor discharge.

· Air cooled condenser are rarely made in size over 5 TR due to higher discharge pressure, higher power consumption and excessive fan noise.

So far as maintenance is concerned, removal of dust, dirt, lint etc. settled on the surface of the condenser is to be removed periodically by using air blower in order to maintain better heat transfer rate. The bearing of the blower fan requires lubrication as per the recommendation of the blower.

12.3.2. Water cooled condensers

In water cooled condenser, heat is rejected in the water which in tern cooled in the cooling tower and the same water is circulated in the water cooled condenser. Based on the constriction, water cooled condenser are of three types viz. shell and tube type, double pipe and shell and coil type.

In double pipe arrangement, the refrigerant condenses in the outer pipe and the cooling water flows through the inner pipe in counter current direction. The shell and coil type condenser consists of shell in which water coil is placed for the circulation of water. Both of these types are not commonly used on account of difficulty of cleaning the water side surface of the pipe or coil.

Shell and tube type water cooled condensers are widely used in commercial refrigeration plant. It consists of a cylindrical shell in which a number of tubes are arranged in parallel and held in place at the ends by tube sheets. The condensing water is circulated through the tubes and the refrigerant is contained in the shell. The end plates being baffled to act as manifolds to guide the water flow through the tubes. The arrangement of the end plate determines the number of passes the water makes through the condenser before leaving the condenser. The shell diameter range from 100 mm to 1500 mm, whereas length varies from 1000 mm to 6500 mm. The number and the diameter of the tube depend on the diameter of the shell. The tube diameter varies from 16 mm to 50 mm. A schematic diagram of two pass shell and tube type water cooled condenser is shown in Fig. 12.3.

Single pass vertical shell and tube type condenser may be employed in large capacity plant. The condensing water flows in the tubes by gravity through distributor installed at the top of each tube which imparts a swirling motion to the water. The hot refrigerant vapour usually enters at the side of the shell near the middle of the condenser and the liquid refrigerant leaves the condenser at the side of the shell near the bottom.

Water cooled condensers are designed for condensing temperature of about 10-12 ºC above the entering water temperature. The heat transfer rate in the condenser depends on the temperature of water, velocity of water, scale deposit, properties of refrigerant, surface area of the condenser etc. It is very important to maintain high overall heat transfer co-efficient for efficient condensation of the refrigerant. The temperature rise of the condenser water can be estimated based on the total heat rejected at the condenser and the mass flow rate of the water in the condenser.

It is necessary to use soft water for the water cooled condenser. The regular cleaning of the condenser tubes is necessary to maintain optimum heat transfer rate.

12.3.2.1. Heat transfer in water cooled condenser

The first step is the heat transfer taking place from refrigerant vapour to the tube through the film of condensed refrigerant liquid on the outside of the tube. The second step is the heat transferred from outside surface to the inside surface of the tube. The third step is the heat transferred through the layer of scale formed inside the tube. Finally, the heat is transferred from the boundary layer film to the stream of water flowing in side the tubes. Heat transfer steps involved are shown in Fig.12.4.

Thus,

$Q = U \times A \times \Delta t$

Where, Q = Heat transfer rate, kJ/s

U = overall heat transfer co-efficient, kJ/kg K

A = Condensing area of heat transfer, m^2

Δt = Temperature difference between refrigerant and cooling water, °C

12.3.3. Evaporative condenser

The evaporative condenser uses both, water and air for the condensation of the refrigerant. The evaporative condenser consists of a coil in which the refrigerant is flowing and condensing inside and its outer surface is wetted with water and exposed to stream of air to which heat is rejected principally by evaporation of water. The water is spayed over the pipes carrying hot refrigerant vapour and movement of air over the wet tubes creates evaporation of water resulting into cooling of the refrigerant. The heat lost from the refrigerant is carried by the air-water mixture leaving the condenser. These types of condensers are mainly classified in two groups.

1. Natural draft evaporative condenser

2. Mechanical draft evaporative condenser

The natural draft evaporative condenser is installed on open space or on the terrace of the building so that sufficient air at maximum velocity is available. The water is pumped from the sump prepared below the condenser and it is spayed on the condenser tubes. The performance of natural draft evaporative condenser varies considerably due to variation of air velocity over the condenser. The space requirement is also more as compared to mechanical draft evaporative condenser. Therefore, mechanical draft evaporative condenser is a better choice to achieve consistent performance of the condenser. The mechanical draft evaporative condenser consists of fan or blower which creates air draft over the wetted tubes. The schematic diagram of an induced draft evaporative condenser is shown in Fig. 12.6. It is necessary to supply make up water in the sump to compensate the water loss due to evaporation .

The induced draft fan is more desirable than forced draught one as it has following advantages.

Ø It provides even air distribution over the coil.

Ø It eliminates the chance of re-circulation of the same air.

Most of the heat given by the refrigerant vapour is carried by the air in the form of sensible and latent heat, hence the effectiveness of this type of condenser depends upon the wet bulb temperature of incoming air. Lower wet bulb temperature of incoming air is desirable to achieve better performance of the condenser. The quantity of water circulated through the condenser should be just sufficient to keep the condenser coil thoroughly wetted.

The removal of scale deposited over the surface of the condenser coil and cleaning of water sump at regular interval are necessary to maintain the rate of heat transfer.

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Lesson 13.

Cooling Towers and Spray Ponds-Types, Construction, Working and Maintenance

13.1. INTRODUCTION

The purpose cooling tower is to cool the water used in the water cooled condenser of the refrigeration plant so that the same water can be reused to absorb the heat at the condenser. The basic principle of cooling of water is the evaporative cooling achieved by using atmospheric air. The condenser water is sprinkled through nozzles and air on passing over the water droplets creates evaporative cooling of water. The heat lost from the water is carried by the air which increases the enthalpy of air leaving the cooling tower. The temperature drop achieved depends on the following factors.

- Surface area of water exposed to air stream.
- Dry bulb and wet bulb temperature of the air.
- Velocity of air.
- Water inlet temperature
- Direction of air flow in relation to water
- Contact time period of air with water.

Theoretically, it is possible to cool the water up to wet bulb temperature (wbt) of the air entering the cooling tower. Generally, the temperature of water coming out from the cooling tower is 3 °C to 5°C above the wbt of the entering air. It is very important to operate the cooling tower at optimum efficiency to get water at lowest possible temperature for the condenser.

13.2. TERMINOLOGY USED IN COOLING TOWER

13.2.1. Range of cooling tower : The difference between the temperature of water entering the cooling tower and temperature of water leaving the cooling tower is known as range of cooling tower.

13.2.2. Cooling tower approach : The difference between the temperature of water leaving the cooling tower and wbt of entering the air is called cooling tower approach.

13.2.3. Wind ward and lee ward side of cooling tower : The side from which ambient air enters the cooling tower is called wind ward side and the opposite side of wind ward is called lee ward side of the cooling tower. It varies as per the direction of ambient air in different seasons of the year. While evaluating the performance of cooling tower, it is essential to know the wind ward and lee ward side to measure the psychrometric properties of air as per the requirement.

13.2.4. Cooling tower capacity (load) : The rate of heat rejected at cooling tower is the measure of cooling tower capacity.

Cooling tower load = Mass flow rate of water, kg/h × Specific heat of water, kJ/kg K × $(t_1 - t_2)$

(kJ/h)

Where,

t_1 = temperature of water entering the cooling tower, K

t_2 = temperature of water leaving the cooling tower, K

13.2.5. Efficiency of Cooling Tower: The efficiency of cooling tower is defined as the ratio of actual drop in temperature of water to the ideal drop in the temperature of water.

η (%) = $(t_1\text{-}t_2/t_1\text{-}wbt_1)$ X 100

Where, wbt of entering air (on wind ward side)

The efficiency of cooling depend on several factors such as surface area of water exposed to air stream, dbt and wbt of the air, velocity of air, flow rate of water, in coming water temperature etc.

Efficiency values obtained in commercial of cooling tower are as under.

Types of cooling tower	**Efficiency (%)**
Spray ponds	50-60
Natural draft cooling tower	50-75
Mechanical draft cooling tower	70-90

The temperature of water leaving the cooling tower can be approximated by the following empirical formula.

$t_2 = t_1 + dbt_1 + 2wbt_1 / 4$

Where, dbt_1 = dry bulb temperature on wind ward side

wbt_1 = wet bulb temperature on wind ward side

The quality of air change when it passes through the cooling tower is depicted in Fig. 13.1. The quality of air on wind ward side and leeward side is at A and B respectively as indicated on psychrometry chart. The enthalpy of air increases when air passes through the cooling tower. It is possible to determine the quantity of air required for the cooling tower based on the change of enthalpy of air when air passes through the cooling tower.

13.3. TYPES OF COOLING TOWER

There are two main categories of cooling towers.

1. Natural draft cooling tower

2. Mechanical draft cooling tower

13.3.1. Natural draft cooling tower

The natural draft atmospheric cooling tower consists of a sump and arrangement for the spray of water from the height of about 5 m to 8 m (Fig.13.2). It is necessary to provide side louvers to minimize carry over water loss. The warm water from the condenser is nuzzled and the cooled water is collected in the sump constructed at the bottom. The performance of this cooling tower varies due to variation of air velocity and it requires more space for the installation of the cooling tower.

13.3.2. Mechanical draft cooling tower

In case of mechanical draft, fan or blower is used to supply the air through the cooling tower. In recent design of the cooling tower, warm water is passed over the supporting medium in form of thin sheet and air is blown through the cooling tower. Induced draft cooling as shown in Fig. 13.3 is preferred over the forced draft cooling tower as air distribution is more uniform in induced draft cooling tower.

13.4. HEAT AND MASS BALANCE OF COOLING TOWER

The heat from the water is transferred to air in cooling tower. Thus,

Heat lost by water = Heat gained by air

$$m_w \cdot C_{pw} \cdot (t_1 - t_2) = m_a \cdot (h_2 - h_1)$$

Where, m_w = mass of water flow in the cooling tower, kg/h

m_a= mass of dry air flow in the moist air, kg/h

C_{pw} = Specific heat of water, kJ/kg K

h_1= Enthalpy of air entering the cooling tower, kJ/ kg dry air

h_2 = Enthalpy of air leaving the cooling tower, kJ/ kg dry air

The above equation is used to find the amount of air required for the cooling tower by considering the load of the tower and the enthalpy of air on wind ward and lee ward side of the cooling tower. Enthalpy of air can be obtained from psychrometric chart by measuring dbt and wbt of the respective air.

13.5. MAINTENANCE ASPECTS OF COOLING TOWER

· Use the soft water as make up water to compensate the quantity of water evaporated.

· Cleaning of nozzles, filling material, sump etc. is necessary at regular interval.

· Replacement of cooling tower water at regular interval as salt concentration of water increases due to evaporation of water.

· Follow general maintenance requirement of pumps, fans etc.

· Adopt corrosion control measures depending on the material of construction of the cooling tower.

13.5.1. Example:

Water is cooled in a natural draft cooling tower. The temperature of water supplied to cooling tower is 35 °C and ambient conditions are 30 °C dbt and 24 °C wbt. Find the efficiency of the cooling tower.

Solution:

Using the equation to find the temperature of water leaving the cooling tower, °C

$t_2 = t_1+dbt_1+2wbt_1 / 4$

$= (35+30+2) * 24 / 4$

$= 28.25$ °C

The cooling tower efficiency can be calculated using the following equation.

η (%) = (t_1-t_2/t_1-wbt_1) X 100

η (%) = **= 59.09 %**

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Lesson 14.

Receiver, Expansion Valves and Evaporators

14.1. INTRODUCTION

The total refrigerant charge required in a refrigeration system depends on operating loads, type of components and distance between the components etc. The quantity of refrigerant in the system must be adequate at all the times so that liquid refrigerant enters the expansion valve. Over charge in the refrigeration system may also cause reduction in the efficiency due to accumulation of liquid refrigerant in the condenser.

14.2. RECEIVERS

The receiver has to play very important to evacuate the part of the system for maintenance and repair. The size of the receiver should be such that it can hold the entire charge of the refrigerant with 1/4 volume available for expansion and safety. It means that receiver should never be more than 80% full during pumping down of refrigerant in the receiver.

A simple sketch of a receiver is shown in the Fig. 14.1. Receivers of large capacity plants are commonly made of steel with welded dished ends and are installed horizontally. Small receivers of small plant may be vertical as it is convenient for installation. The liquid pipe from the condenser to the receiver should be sufficiently sized and provided with slope to promote easy movement of liquid refrigerant. The outlet pipeline from the receiver is connected either from the bottom or by means of an internal standpipe. A liquid shut off valve is fitted at the outlet of the receiver. Ammonia receivers may have an oil drum pot, and it is mounted with little slope towards oil pot. Receiver is provided with safety pressure relief devices, level indicator etc. In practice, the receiver is one-sixth full during normal running of the system.

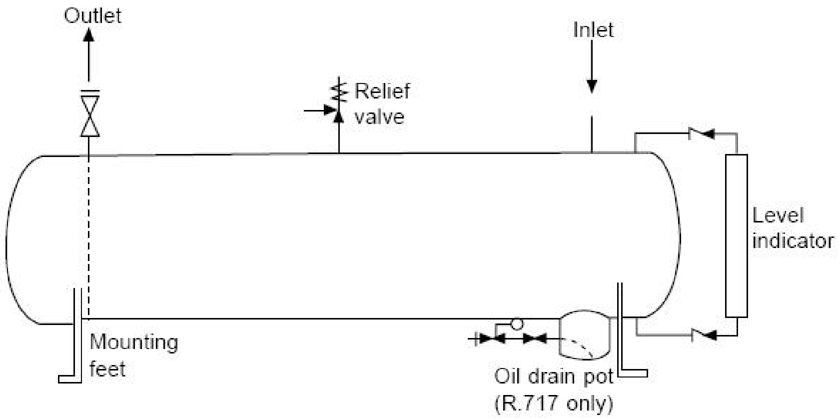

Fig. 14.1: Ammonia receiver

14.3. EXPANSION VALVES

This is one of the basic components of the vapour compression refrigeration system. The functions expansion devices are listed below.

1. It reduces the pressure of the refrigerant coming from the condenser as per the requirement of the system.

2. It regulates the flow of the refrigerant as per the load on the evaporator.

The different devices which are used to perform the above functions are listed below.

1. Hand expansion valve. 4. High side float valve

2. Automatic expansion valve 5. Low side float valve

3. Thermostatic expansion valve 6. Capillary tube

14.3.1. Hand Expansion Valve

The hand operated expansion valve is employed to manually regulate the flow of liquid refrigerant to the evaporator. However, its use is limited to systems operating under fairly constant loads for long periods of time. Where loads are subject to repeated and rapid fluctuations, a condition that prevails in many industrial and commercial systems, the need for greater flexibility in liquid flow control is required. Therefore, manually operated expansion valves are employed on by pass line. A sketch of a hand expansion valve is given in Fig. 14.2.

14.3.2. Automatic Expansion Valve

The automatic expansion valve functions in response to pressure changes occurring within the evaporator to allow more or less liquid refrigerant to flow in the evaporator. It is often referred to as a constant pressure expansion valve. Its use is limited to systems operated dry expansion under fairly constant loads and generally confined to small units.

If the pressure of evaporator falls due to decrease in load, the spring pressure causes the valve to open more. The rate of evaporation increases due to increased quantity of refrigerant and builds up the pressure until the equilibrium is reached with the spring tension. Reverse action takes place when the pressure in evaporator increases. A sketch of a hand expansion valve is given in Fig. 14.3.

14.3.3. Thermostatic Expansion Valve

The thermostatic expansion valve controls the flow of refrigerant through the evaporator in such a way that the quality of vapour leaving the evaporator will be always in superheated condition. If the quantity of liquid in the evaporator diminishes, more heat transfer surface is available for superheating the suction gas which raises the temperature of the feeler bulb and power fluid. The pressure of the power fluid thereby increases, which opens the valve wider and increases the flow of refrigerant into the evaporator. The increase in flow rate of refrigerant in the evaporator decreases the super heat of the suction gas which reduces valve opening. The degree of superheat of the vapor leaving the evaporator depends upon the initial setting of the spring tension. Once the valve is adjusted for a particular superheat, then that superheat will be maintained under all load conditions on the evaporator. The working principle of thermostatic expansion valve is explained in Fig. 14.4. This is most widely used refrigerant control device for medium size refrigeration systems.

14.3.4. High side Float Valve

This type of float valve allows only a minimum amount of liquid to remain in the high pressure side of the system so that nearly all the refrigerant charge is in the evaporator. As the compressed vapour condenses in the condenser, it flows into the float chamber from which it is forced under his pressure into the evaporator when a sufficient amount has collected to raise the float and open the valve. The float is set to open at a given level in the high side float chamber, it follows that the liquid level in the evaporator is nearly constant at all times. As entire charge of refrigerant is in the evaporator, it is necessary that the system contain only enough liquid refrigerants in order to prevent its carry over to the compressor. A system working with high pressure float valve is shown in Fig. 14.5.

The high side float control may be installed either above or below the evaporator as it is independent of the liquid level in the evaporator. The flow chamber must be very near to the evaporator.

The liquid level in the float chamber drops as the compressor is stopped and control valve is closed and remain closed until the compressor is restarted.

14.3.5. Low side Flat Valve

Low side float valve maintains constant level of refrigerant in the evaporator by supplying quantity of liquid refrigerant required to take the load in the evaporator. It maintains the evaporator always filled with the liquid refrigerant under all conditions irrespective of evaporator temperature and pressure. This method of control is used only on flooded evaporator. This is used in multiple or in parallel working evaporators used in commercial or industrial applications. A low pressure float valve system is shown in Fig. 14.6.

When the compressor is stopped, the liquid in the evaporator continues to vaporize until the pressure reaches a point corresponding to the temperature. At that point, evaporation ceases and the valve closes to isolate the evaporator.

14.3.6. Capillary tube

This device is only used for small capacity units like domestic refrigerators, water coolers and small commercial freezers. It is a small diameter tube connected between condenser and evaporator. The required pressure drop is caused due to heavy frictional resistance offered by a small diameter tube. The resistance is directly proportional to the length inversely proportional to the diameter. Different length and diameter combinations are recommended for the required pressure drop and flow quantity. The use of these expansion devices is limited to small units.

14.4. EVAPORATORS

The evaporator is a heat exchanger where the actual cooling effect is produced. The evaporator receives the low pressure refrigerant from the expansion valve and brings the material to be cooled in contact with the surface of the evaporator. The refrigerant absorbs the heat from the materials to be cooled (air/water/milk/any other material). The refrigerant takes up its latent heat from the load and becomes vapour. The refrigerant vapour produced in the evaporator is pumped by the compressor and low pressure is maintained to maintained low evaporating temperature. The evaporators are fabricated from different materials having various designs depending on the requirement of cooling the product or material. The following factors are to be considered in the design of the evaporators.

14.4.1. Heat Transfer

The heat transfer capacity of the evaporator is given by

Heat Flow rate (kJ/h), $Q = U \cdot A \cdot (T_f - T_s)$

Where, U = Overall Heat Transfer Co-efficient, kJ/($m^2 \cdot h \cdot K$)

A = Heat transfer surface area of evaporator, m^2

T_f = Temperature of the fluid to be cooled in the evaporator, K

T_s = Evaporating temperature of refrigerant, K

The requirement of surface area of the evaporator depends on value of U as well as temperature difference between the material to be cooled and the temperature of refrigerant.

14.4.2. Material of construction

The selection of material to be used for construction of evaporator is based on several factors such as type of refrigerant, thermal conductivity of the metal, cost, ease of fabrication, product to be cooled, etc. Copper is commonly used in small capacity plants due to its higher conductivity and ease of fabrication but it can not be used in ammonia plant as ammonia is corrosive to copper. Steel tubes/pipelines are commonly used in large capacity ammonia plant.

14.4.3. Velocity of refrigerant

Heat transfer co-efficient increases with increase in velocity of refrigerant in the evaporator. But increased velocity causes more pressure loss. It is very important point to be considered while selecting liquid over feed system.

14.4.4. Cooling requirement

The selection of metal for fabrication is mainly decided by the product to be cooled. The evaporator is fabricated from S. S. for ice-cream freezer, milk cooling equipment etc., while evaporator of ice-bank system is made from steel tubes. Air cooling evaporators for cold rooms, blast freezers, air-conditioning, etc. have finned pipe coils with fans to blow air over the coil.

14.5. TYPES OF EVAPORATORS

Ø Flooded Evaporators

Ø Dry type Evaporators

14.5.1. Flooded Evaporators

The evaporator which is always filled with the liquid refrigerant during operation of the system is called flooded evaporator (Fig. 14.7). This type of evaporator gives higher rate of heat transfer and entire surface area of the evaporator is utilized for the heat transfer. The refrigerant boils and the saturated vapour refrigerant leaves the evaporator from the top. This is widely used in large ammonia systems. The refrigerant enters a surge drum through a float type expansion valve. The vapour produced due to expansion of liquid refrigerant is directly pumped by the compressor of the system. This vapour does not take part in refrigeration hence its removal makes the evaporator more compact and reduces pressure drop. The liquid refrigerant enters the evaporator from the bottom of the surge drum. The mixture of liquid and vapour bubbles rises up along the evaporator tubes and is separated as it enters the surge drum. The unevaporated liquid refrigerant circulates again in the tubes along with the constant supply of liquid refrigerant from the expansion valve. The lubricating oil tends to accumulate in the flooded evaporator hence an effective oil separator must be employed after the compressor.

14.5.2. Dry type Evaporators

The liquid refrigerant is fed into the evaporator through the expansion valve and the quantity of refrigerant is so controlled that super heating of refrigerant vapour takes place at the end of evaporator (Fig. 14.8). Any increase or decrease in super heat at the end of evaporator due to change in evaporator load, alters the opening of expansion valve. The increase in load increases super heat at the end of evaporator which increases the flow rate of refrigerant to bring the super heat as it was earlier. The reverse action takes place when load decreases. These evaporators are provided with thermostatic expansion valve. In this type of evaporators, the oil keeps on, until it gets back to the compressor suction.

These evaporators are further classified as natural convection evaporator or forced convection evaporators. Forced convection evaporators used in cold storages of dairy and food plants. In ammonia plant, any oil present with liquid refrigerant will fall to the bottom of the evaporator which is drawn off from the oil drain connection provided in the evaporator.

Evaporators employed for air cooling have fined tube as where as liquid cooling evaporators may be shell and coil type or any form suitable for cooling of liquid. Plate type evaporator can be used for cooling of packaged product by conduction.

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Lesson 15. Temperature and Safety Controls: Thermostat, HP and LP Cut Out, Overload Protector

15.1. INTRODUCTION

It is very essential to incorporate various controls in the vapour compression refrigeration system in order to get required temperature and to control the plant operation under abnormal operating conditions. These controls help to operate the system safely and also operate the plant under optimum conditions to reduce the operating cost of the plant.

15.2. THERMOSTAT

Thermostat is used for controlling the temperature in cold rooms, brine tanks, freezing chambers, hardening chambers etc. It is a temperature operated electric switch making circuit when the temperature rises to a predetermined value and breaking circuit when the temperature falls to another predetermined value. It is necessary to stop the refrigeration plant at required temperature and again to start the plant at predetermined rise in the temperature. This is possible by using a thermostat which breaks and makes the electrical circuit of the compressor or solenoid valve. The different types of thermostats are available for different applications to cover wide temperature range. A schematic representation of a thermostat is depicted in Fig. 15.1.

Generally, thermostat should be fitted outside the cold room in which the temperature to be controlled. The phial is fixed inside a cold room by means of a clip. The capillary tube must not be passed through any room colder than the one in which the phial is fitted. The best way of passing the capillary tube through the wall of the cold room is by way of grouted-in sleeve having at least 10 mm bore. It must be seen that iron filings do not get into the instrument during installation, as such fillings will be attracted by the permanents magnet and thus lessen the proper clearance between armature and magnet.

The thermostat is adjusted by the knob to interrupt the circuit at the desired temperature, which can be read on the scale on the cover of the instrument. The temperature differential desired for the starting of the plant is set by the differential adjusting nut inside the case, which is likewise provided with an adjustment scale. It is necessary to check these settings during the operation of the plant.

For instance, the knob on the cover is set for a scale value of –29 °C, and the differential adjusting nut for 3 °C then thermostat will start the plant at –26 °C and stop it again at –29 °C.

Thus, Starting temperature = stopping temperature + differential

15.3. HIGH PRESSURE SWITCH (CUTOUT)

It is a safety control used in vapour compression refrigeration plant to stop the refrigeration system when discharge pressure rises above the pre-set value of cut out pressure. This is an electrically operated switch which breaks the electrical circuit of the compressor when condensing pressure increases excessively above the normal working pressure of the system.

Working of high pressure switch is indicated in Fig. 15.2. It is connected on high pressure side of the system.

The reasons of excessive high pressure are as under.

1. Inadequate or lack of cooling medium at the condenser.
2. Higher temperature of cooling medium used at the condenser.
3. Poor rate of heat transfer at the condenser
4. Presence of non-condensable gases in the refrigeration system.

This switch may be fully automatic (stars and stops plant) or semi-automatic (stops the plant automatically but re-starting manually) depending upon the wiring connection of the switch with the system.

15.4. LOW PRESSURE SWITCH (CUTOUT)

It is also a safety control used in vapour compression refrigeration plant to stop the refrigeration system when suction pressure drops below the pre-set value of cut out pressure. This is an electrically operated switch which breaks the electrical circuit of the compressor when suction pressure reduces below the normal working pressure of the system.

Working of low pressure switch is indicated in Fig. 15.3. It is connected on suction side of the compressor.

The reasons of excessive low pressure are as under.

1. Very low load at the evaporator.
2. Defective expansion valve/ blockage of expansion valve.

This switch may be fully automatic (stars and stops plant) or semi-automatic (stops the plant automatically but re-starting manually) depending upon the wiring connection of the switch with the system.

15.5. OVER LOAD PROTECTOR

The National Electric Code (NEC) defines Motor Overload Protection as that device which is intended to protect motors, motor-control apparatus, and motor branch-circuit conductors against excessive heating due to motor overloads and failure of the motor to start. Overload protection for large three-phase motors is provided by Thermal Overload Relays which are connected to Current Transformers (CT's). But in new installations utilize microprocessor-based motor protective relays which can be programmed to provide both overload and short-circuit protection. These protective relays often also accept inputs from Resistance

Temperature Devices (RTD's) imbedded in the motor windings (usually two per phase) and the relays are capable of displaying the winding and motor bearing temperatures, and provide both alarm and trip capability. The over Load protector is operated by a snap action of bimetal disc and is sensitive to both temperature and current. When properly connected shuts off the motor when temperature exceeds maximum safe level due to an overload condition.

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Lesson 16.

Defrosting, Refrigeration Piping and Balancing of Different Components of the System.

16.1. INTRODUCTION

The condensation of water vapour of the room/cold storage causes formation of frost over the evaporator. Formation of ice takes place in all the evaporators which are operating below the freeing point of water (0 °C). The accumulation of ice over the heat transfer surface reduces the heat transfer rate as the ice is poor conductor of heat. Therefore, it is necessary to remove the ice deposited over the evaporator at periodic time interval. The operation of removing frosted ice from the evaporator is known as defrosting of evaporator. The period of defrosting depends on type of evaporator, relative humidity of the cold room, evaporation temperature etc.

16.2. METHODS OF DEFROSTING

16.2.1. Manual Defrosting: The simplest of defrosting is to shut down the plant manually and restart it when the accumulated ice is melted away from the cooling coil. This method is not suitable for big capacity evaporators working at very low temperature as it takes long time and causes warming of the product stored in the cold storage.

16.2.2. Automatic Periodic Defrosting: The starting and stopping of refrigeration plant is automatic as per the change of evaporating pressure/temperature and defrosting is completed naturally. The frosting of evaporator coil causes reduction of suction pressure due to reduced heat transfer between the coil and the air. When the suction pressure falls below the predetermined value due to reduction of heat transfer on account of excessive frosting, pressure operated control stops the plant.

16.2.3. Water Defrosting: In this method, Water is sprayed in ample quantity over the ice accumulated on the evaporator to washout the ice from the coil. This method is used in many commercial cold storages. During defrosting cycle, the supply of refrigerant is stopped and water is poured over the cooling coil. The water together with the melted ice is removed through pipelines. The time required for defrosting varies from 10-20 minutes depending on the amount of ice deposited and the temperature of water used for defrosting.

16.2.4. Hot gas defrosting: The defrosting carried out by using hot refrigerant gas from the compressor is called hot gas defrosting of evaporator. Hot gas defrosting is shown in the Fig. 16.2. The process of defrosting is performed at regular interval (6-10 hours) by the action of solenoid valve which supplies hot refrigerant gas after compression to the evaporator. Hot gas supply for few minutes melts the ice accumulated on the evaporator. The condensed refrigerant is re-evaporated in the re-evaporator before it goes to compressor.

16.2.5. Defrosting by reversing the cycle: When hot refrigerant gas is from the compressor is passed to the evaporator, it melts the frost accumulated on the evaporator coil. The normal operating conditions of the cycle and defrosting cycle are shown in Fig. 16.3. This method is not used in commercial systems.

16.2.6. Electric Defrosting: This method of defrosting is employed for fined coil evaporator. A bank of electric heaters is located near the coil. During defrosting, the system remains closed and heater starts to melt the frost accumulated on the evaporator. The time required for defrosting varies from few minutes to 30 minutes depending on the size of the evaporated and level of frost deposited on the evaporator. This method is now widely used in household refrigerator.

In dairy and food cold storages, water defrosting and hot gas defrosting are commonly employed for defrosting of evaporators.

16.3. REFRIGERATION PIPING

It is necessary to connect all the component of the refrigeration plant using well designed pipelines. The quality of refrigerant and the flow rate of refrigerant are different in the system and accordingly size of pipelines are required in the system. For example, after the compressor the refrigerant is hot vapour to be supplied to condenser while liquid refrigerant flows in the pipeline between receiver and expansion valve. The distance of various components of the refrigeration system is one of the important considerations in selection of pipelines in order to minimize the pressure drop. The material of pipeline depends on the type of refrigerant used in the system. Small capacity systems using R-22 or R-134a requires copper pipelines to connect various components with flare fittings and brazing work. Copper pipelines can not be used in ammonia plant as ammonia is corrosive to copper and its alloys. Mild steel pipelines are commonly used in ammonia plant.

16.4. BALANCING OF DIFFERENT COMPONENTS OF THE SYSTEM

Balancing of various components of the refrigeration plant is very important design aspect of the system. Misbalancing of any component may greatly affect the performance of the system. Under size expansion valve may result in to lower evaporating pressure and over size expansion valve supplies higher flow rate of refrigerant leading liquid pumping of refrigerant. The rate of the heat transfer plays very important role in the design of evaporator and condenser. It is necessary to pump the vapour from the evaporator at the rate it is produced in the evaporator. If compressor is not pumping the vapour produced in the evaporator, then pressure of the evaporator increases and it will not be possible to achieve desirable temperature of cold storage. It is also obvious that evaporation of refrigerant in the evaporator takes place depending on the load and therefore capacity control of compressor is one of the essential requirements in economical working of refrigeration plant.

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Module 5- Multi-Evaporator And Compressor Systems

Lesson 17. Importance of Multiple Evaporator and Compressor Systems, Multi Evaporator and One Compressor Systems.

17.1. INTRODUCTION

There are many applications in dairy and food plants where refrigeration is required at different temperatures depending on the type of products to be stored. As instance, milk is normally stored at 3-4 ^{o}C while ice-cream storage requires –30 ^{o}C temperature. One simple alternative is to use different refrigeration systems to meet different requirement of temperatures for different cold storages. But, this may not be economically choice due to the high total initial cost of different systems. Another alternative is to use a single refrigeration system with one compressor and multi- evaporators operating at different temperatures as per the requirement of the products. It is also necessary to use multi-evaporator and two stage compression system if the compression ratio is more (>7).

17.2. MULTI- EVAPORATOR SYSTEMS WITH SINGLE COMPRESSOR

17.2.1. Multi- evaporators operating at the same temperature

A vapour compression system having two evaporators operating at the same temperature with single compressor is shown in the Fig.17.1 and the corresponding cycle is presented on P-H diagram in Fig. 17.2. The refrigerant from receiver is supplied to evaporators through individual expansion valves and the vapour produced in evaporators is pumped by the compressor. The saturated suction vapour having h_1 enthalpy is compressed to condensing pressure and the enthalpy the discharged vapour is h_2.

C.O.P. of the system = 210 X (T_1+T_2),kJ/min / work of compression,kJ/min

Where, T_1 = Capacity of E_1 in ton

T_2 = Capacity of E_2 in ton

Mass flow rate of refrigerant (m_1) in E_1= 210 x T_1 /h_1- h_4 kg/min

Mass flow rate of refrigerant (m_2) in E_2= 210 x T_2 /h_1- h_4 kg/min

Thus, C.O.P. = 210 x (T_1 + T_2) / (m_1 + m_2) (h_2 - h_1)

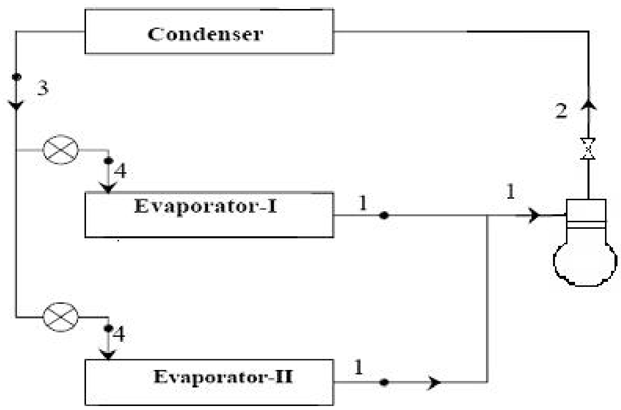

Fig. 17.1: Two evaporator with single compressor system

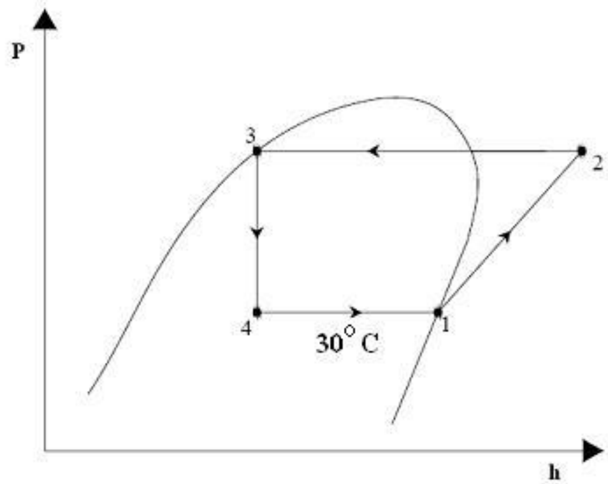

Fig. 17.2: Two evaporator with single compressor system on P-H diagram

17.2.2. Multi- evaporators operating at different temperatures

This is a practical requirement to have different temperatures for storage of product in dairy and food factories. Ice bank evaporators operate at about -8 °C to -10 °C, while evaporator of ice-cream hardening room may operate at -35 °C. Therefore, it is very common under field conditions to use multi-evaporators maintained at different temperatures.

A block diagram of vapour compression refrigeration consisting of two evaporators operating at different temperature and one compressor system is shown in Fig. 17.3. The working cycle of the system is represented on P-H diagram in Fig. 17.4.

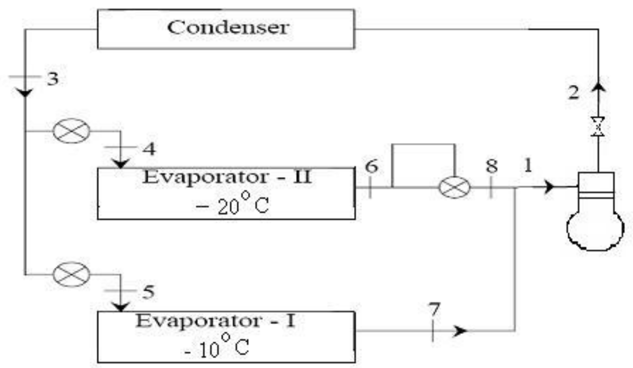

Fig.17.3: Two evaporators operating at different temperature and one compressor system

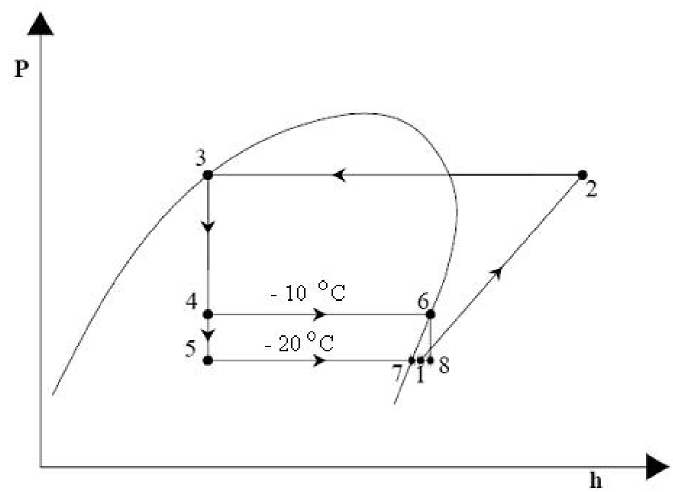

Fig.17.4: Working cycle of two evaporators operating at two different temperature and one

The refrigerant is supplied to two evaporators operating at different temperature through individual expansion valves. The saturated vapour leaving the lower temperature evaporator at point 7 is combined with the vapour coming from higher temperature evaporator. A pressure reducing valve is installed to reduce the pressure of the vapour corresponding to the pressure of lower temperature evaporator. The enthalpy of suction vapour can be obtained by taking heat and mass balance of the refrigerant vapour leaving the evaporators.

C.O.P. of the system = 210 x (T_1+T_2),kJ/min / work of compression,kJ/min

Where, T_1 = Capacity of E_1 in ton

T_2 = Capacity of E_2 in ton

Mass flow rate of refrigerant (m_1) in E_1= 210 x T_1 /h_6- h_4 kg/min

Mass flow rate of refrigerant (m_2) in E_2= 210 x T_2 /h_7- h_5 kg/min

Thus, C.O.P. = 210 x (T_1 + T_2) / (m_1 + m_2) (h_2 - h_1)

The enthalpy of the suction gas is determined as under.

$m_1 \cdot h_6 + m_2 \cdot h_7 = (m_1 + m_2)\, h_1$

$h_1 = (m_1 + m_2) \;/\; m_1 \cdot h_6 + m_2 \cdot h_7$

For calculating theoretical C.O.P. of such systems, it is necessary to read the values of enthalpy from refrigerant chart and P-H diagram. By calculating the enthalpy at point 1and locating that point on P-H diagram, the value of enthalpy at point 2 can be obtained from the refrigerant chart. The above analysis may deviate, if there is any superheating of vapour or sub-cooling of liquid refrigerant.

17.2.3. Multi-evaporator with a single compressor and multiple expansion valves:

Fig. 17.5 and Fig. 17.6 show system schematic diagram and P-H diagram of a multi-evaporator with a single compressor and multiple expansion valves. It can be seen from the P-H diagram that the advantage of this system compared to the system with individual expansion valves is that the refrigeration effect of the low temperature evaporator increases as saturated liquid enters the low stage expansion valve. This is possible as the flash gas is removed at state 4.

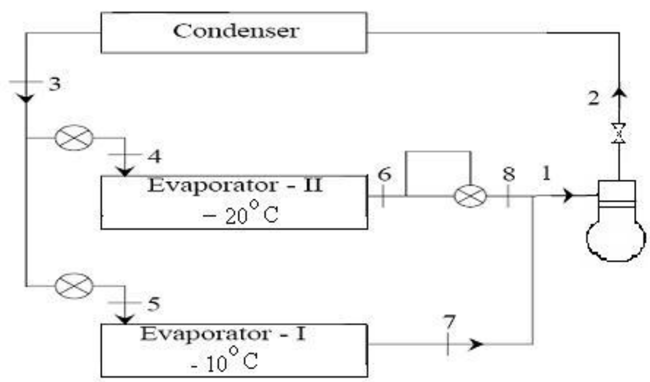

Fig. 17.5: System schematic diagram of a multi-evaporator with a single compressor and multiple expansion valves

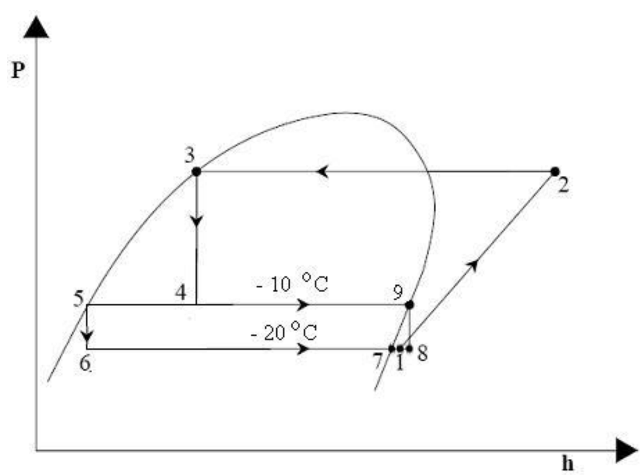

Fig. 17.6: System P-H diagram of a multi-evaporator with a single compressor and multiple expansion valves

C.O.P. of the system = 210 x (T_1+T_2),kJ/min / work of compression,kJ/min

Where, T_1 = Capacity of E_1 in ton

T_2 = Capacity of E_2 in ton

Mass flow rate of refrigerant (m_1) in E_1= 210 x T_1 /h_7- h_4 kg/min

Mass flow rate of refrigerant (m_2) in E_2= 210 x T_2 /h_8- h_6 kg/min

Thus, C.O.P. = 210 x (T_1 + T_2) / (m_1 + m_2) (h_2 - h_1)

The enthalpy of the suction gas is determined as under.

$m_1 \cdot h_6 + m_2 \cdot h_7 = (m_1 + m_2)\ h_1$

$h_1 = (m_1 + m_2)\ /\ m_1 \cdot h_6 + m_2 \cdot h_7$

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Lesson 18.

Dual compression and Individual compressors systems, compound compression

18.1. INTRODUCTION

It is a compressor having a low pressure inlet valve, medium pressure inlet ports and high pressure discharge valve. Thus, the purpose of such compressor is to take vapour from two different evaporators which are at different pressures. The discharge of the compressed refrigerant is at some common discharge pressure. The dual compressor is used to replace two compressors which are required for two sources of refrigerant vapour. A simple form of the dual compressor. The compressor consists of low pressure suction valve, medium pressure suction ports and high pressure delivery valve.

Expansion of liquid refrigerant takes place from P_3 to P_2 (3-4) and vapour produced in the expansion process is directly passed to dual compressor. The corresponding P-H diagram.

A dual compressor with multi loads at different temperatures. Evaporators are provided with individual expansion valves. Dual compressor pumps the vapour from E_1 and E_2 evaporators and delivers to a condensing pressure. The corresponding P-H diagram. The analysis of such system can be done by calculating the mass flow rate in E_1 and E_2 based on the capacity of these evaporators. The vapour is compressed from point $\mathbf{h_m}$ to h_2 and rise in enthalpy during compression gives the work of compression.

18.2. INDIVIDUAL COMPRESSOR SYSTEM

Dual compressor pumps vapour refrigerant from two separate evaporators without affecting the performance of the system while individual compressor requires two compressors. A vapour compression refrigeration system consisting of multi-evaporators with multiple expansion valves having individual compressor.

The corresponding P-H diagram is shown in **Fig. 18.7**. Assuming, T_1 and T_2 are the refrigeration load on E_1 and E_2.

$$\therefore \text{Mass flow rate in } E_1, m_1 = \frac{3.5\,T_1}{h_2 - h_1}\ kg/s$$

$$\therefore \text{Mass flow rate in } E_2, m_2 = \frac{3.5\,T_2}{h_8 - h_6}\ kg/s$$

Power required for $C_1 = m_1\,(h_3 - h_2)$ kW

Mass of refrigerant handled by C_2 compressor is given as under.

$mC_2 = m_1 + m_2 + m_v$

Where, m_v = mass of refrigerant vapour carried with $(m_1 + m_2)$

Power required for C_2 = mC_2 ($h_4 - h_8$) kW

Total power required for the system = (C_1 + C_2) kW

COP of the system= $(3.5\ (T_1+T_2))/(m_1\ (h_3-h_2\ )+mC_2\ (h_4-h_8\ )\ )$

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Lesson 19.

Comparison of Compound Compression with Single Compressor System

19.1. INTRODUCTION

It becomes necessary to adopt compound compression when compression ratio is high in order to reduce input power for the refrigeration system. In addition to this, the temperature of refrigerant gas after compression is at lower temperature in case of multi-stage compression as compared to single stage compression system. Normally, single stage compressor is used for compression ratios of around 7.0 to 9.0 depending upon the capacity of the plant.

19.2. REMOVAL OF FLASH GAS

The saturated liquid refrigerant is throttled through expansion valve from condensing pressure to evaporating pressure. In the process, 5-10 percent of liquid refrigerant becomes vapour depending on the extent of expansion of the refrigerant. This flash gas is removed by installing a tank which is known as flash chamber after the expansion valve as shown in Fig. 19.1. The separated flash gas is directly by-passed to the compressor which reduces the size of evaporator required in the system as pure liquid refrigerant enters in the evaporator. The arrangement has no effect on the thermodynamic cycle.

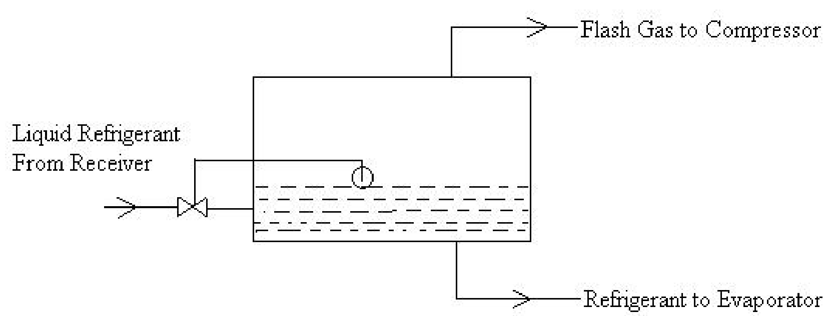

Fig. 19.1: Flash chamber

19.3. INTER- COOLING

It is necessary to cool the refrigerant between two stages of compression to get full advantage of multi-compression system. This process is called inter-cooling which is done by using refrigerant or combination of water and refrigerant. A two stage compressor with inter cooling using refrigerant is shown in Fig. 19.2. The corresponding cycle is indicated on P-H diagram in Fig. 19.3. The saving in work done by adopting two-stage compression with inter-cooling is shown in Fig. 19.4.

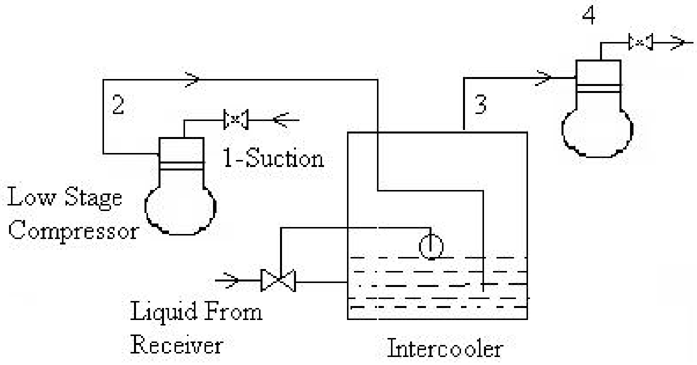

Fig. 19.2: Inter cooler in two-stage compression system

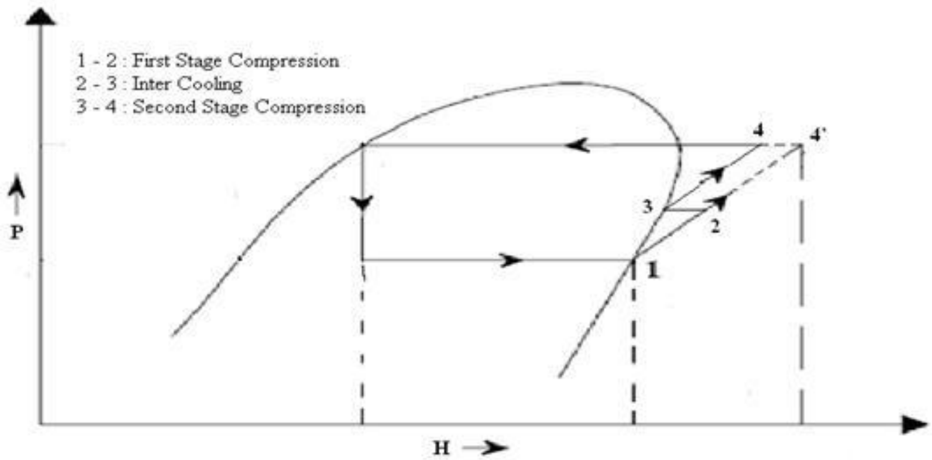

Fig.19.3: Inter cooling on P-H diagram

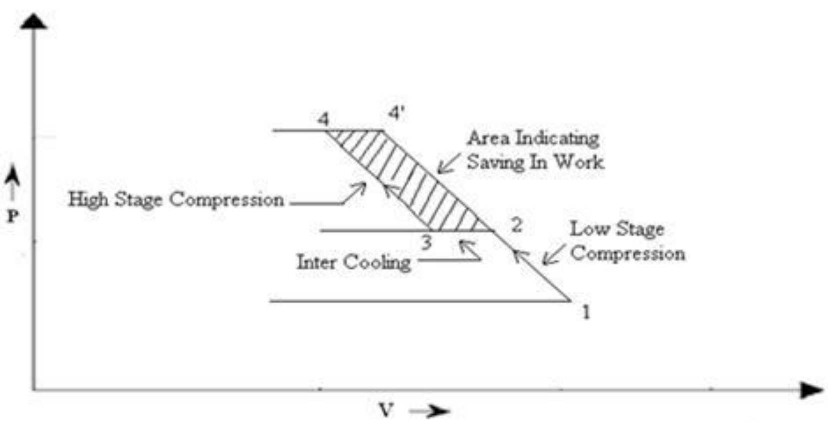

Fig.19.4: P-V diagram of two stage compression with inter cooling

Multi evaporator and single compressor system:

A refrigeration system having three evaporators operating at different temperatures and single compressor is shown in Fig. 19.5

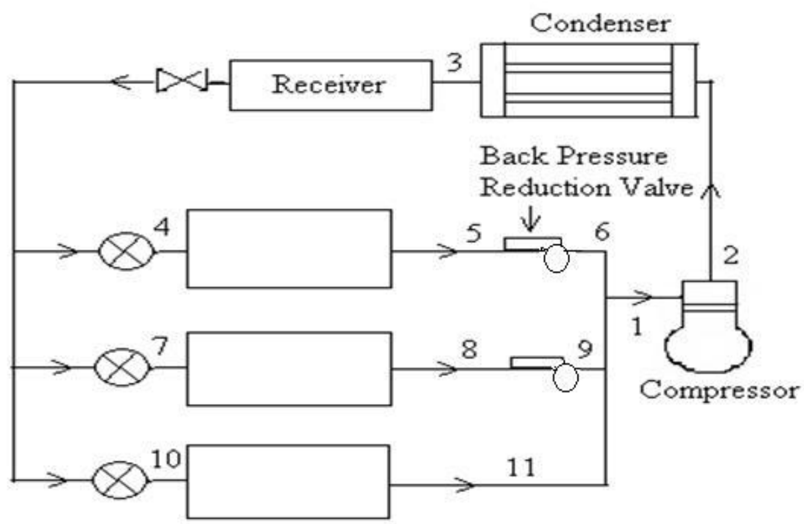

Fig. 19.5: Multievaporator and single compressor system

Three evaporators, E_1, E_2 and E_3 having T_1, T_2 and T_3 ton cooling capacity respectively are provided with individual expansion valve as operating temperatures are different to achieve storage conditions in cold storages. E_2 and E_3 are provided with back pressure reducing valve in order to reducing the pressure of the gas to the operating pressure of E_1 as it is operating at lower pressure. The corresponding P-H diagram is shown in Fig. 19.6.

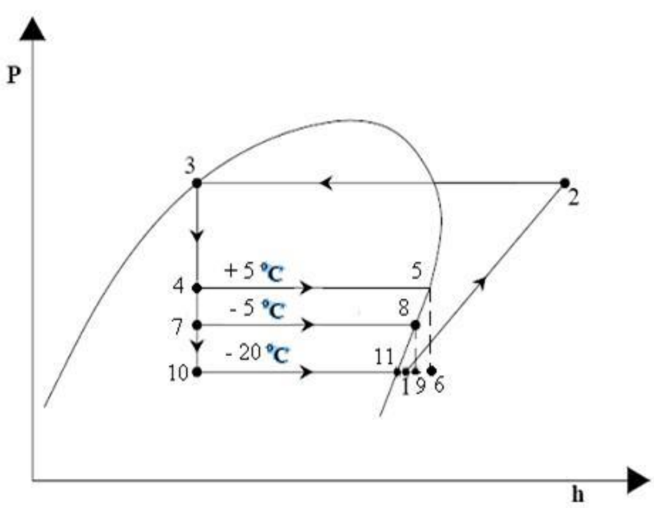

Fig. 19.6: Multievaporator and single compressor system on P-H diagram

Mass flow rate in E_1, E_2 and E_3 is calculated as under.

Mass Flow rate in E_1,m_1 = $(3.5T_1)/(h_{11}- h_{10})$ kg/s

Mass Flow rate in E_2,m_2 = $(3.5T_2)/(h_8- h_7)$ kg/s

Mass Flow rate in E_3,m_3 = $(3.5T_3)/(h_5- h_4)$ kg/s

By taking the heat and mass balance of the refrigerant vapour leaving each evaporator, the value of h_1 can be calculated as under.

$m_1 h_{11} + m_2 h_9 + m_3 h_6 = (m_1 + m_2 + m_3) h_1$

$\therefore h_1 = m_1 h_{11} + m_2 h_9 + m_3 h_6)/ m_1 + m_2 + m_3$

Work done of compressor and C.O.P. of the plant can be calculated as under.

$kW = (m_1+m_2+m_3) (h_2 - h_1)$

And

$C.O.P.= 3.5 (T_1+ T_2+ T_3))/ (m_1 + m_2 + m_3) (h_2- h_1))$

It is very essential to find enthalpy values from P-H diagram or refrigerant properties tables corresponding to the operating conditions of the plant to carry out energy analysis of the system. If each evaporator is provided with individual compressor, then determination of kW and C.O.P. is in the line with simple vapour compression refrigeration system. Power requirement of E_1, E_2 and E_3 can be calculated separately taking corresponding work of compression per kg of refrigerant and individual mass flow rate of refrigerant of each evaporator.

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Lesson 20.

Working and Mathematical Analysis of Above Systems

20.1. INTRODUCTION

Multi-evaporator vapour compression refrigeration systems are used where it is necessary to maintain different temperatures for the storage of products. For example, ice-bank evaporator operates at about -10 °C while evaporator of ice-cream cold storage is operated at -35 °C. Therefore, it is a basic requirement to maintain different temperature in dairy plants. Multi-compression (mostly two-stage compression) is usually required when compression ratio is above 7. It is obvious that multi-evaporator and multi-compressor system is employed to achieve different temperature requirement and higher compression ratio. In addition to this, sub cooling and inter-cooling of refrigerant become essential to improve the performance of the refrigeration plant. The actual mathematical analysis of the entire plant is based on the actual refrigeration cycle involved in the plant.

20.2. SYSTEM WITH MULTIPLE EXPANSION AND INDIVIDUAL COMPRESSOR

The power requirement can be reduced by the use of an individual compressor for each evaporator and by the multiple expansion valves. A system consisting of two evaporators with individual compressor is shown in Fig. 20.1. Discharge gas from these compressors is combined and is condensed in the condenser of the plant. The liquid refrigerant is throttled to a pressure equivalent to evaporating temperature of E_2 evaporator. The quantity of refrigerant throttled by this expansion valve is sum of the refrigerant flow rate needed for E_2 and E_1. This is known as multiple throttling of refrigerant. The corresponding P-H diagram is shown in Fig. 20.2.

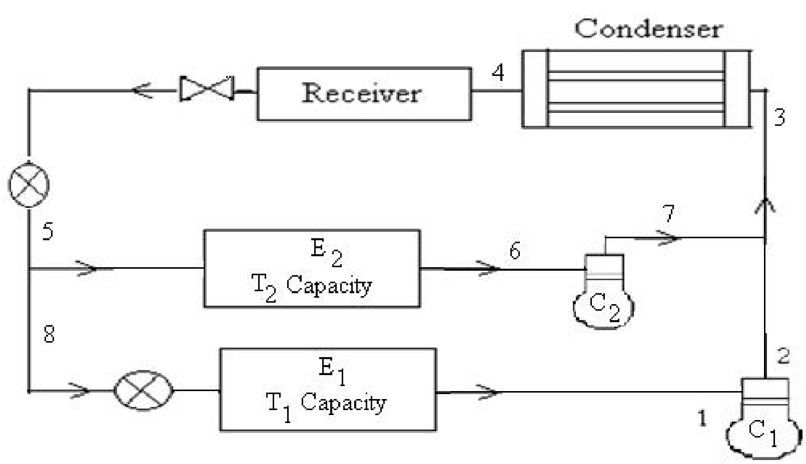

Fig.20.1: Multievaporator and individual compressor system

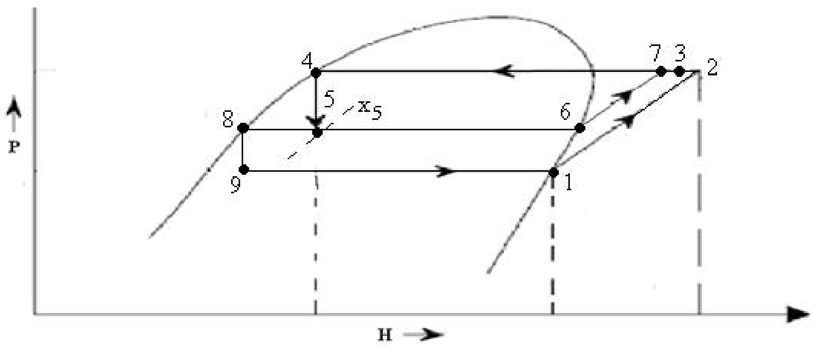

Fig.20.2: Multievaporator and individual compressor system on P-H diagram

$$\text{Mass flow rate in } E_1,\ m_1 = \frac{3.5T_1}{h_1 - h_9} \text{ kg/s}$$

$$\text{Mass Flow rate in } E_2,\ m_2 = m_1 + m_1 \left(\frac{X_5}{1 - X_5}\right) \text{ kg/s}$$

$$\text{kW of } C_1 = m_1 (h_2 - h_1) \text{ kW}$$

$$\text{kW of } C_2 = m_2 (h_7 - h_6) \text{ kW}$$

Enthalpy of compressed refrigerant can be calculated by taking heat & mass balance of the refrigerant after compression.

$$m_1 h_2 + m_2 h_7 = (m_1 + m_2) h_3$$

$$COP = \frac{3.5\ (T_1 + T_2)}{m_1\ (h_2 - h_1) + m_2\ (h_7 - h_6)}$$

System with multi-evaporator and multi compression:

A system consisting of two evaporators and multi-compressor are shown in Fig. 20.3. The refrigerant from the receiver is supplied to E_2 which is operating at higher temperature as compared to E_1 (say -5 °C). At the same operating pressure, flash chamber is installed to inter-cool the vapour refrigerant after first stage of compression. The corresponding P-H diagram is shown in Fig. 20.4.

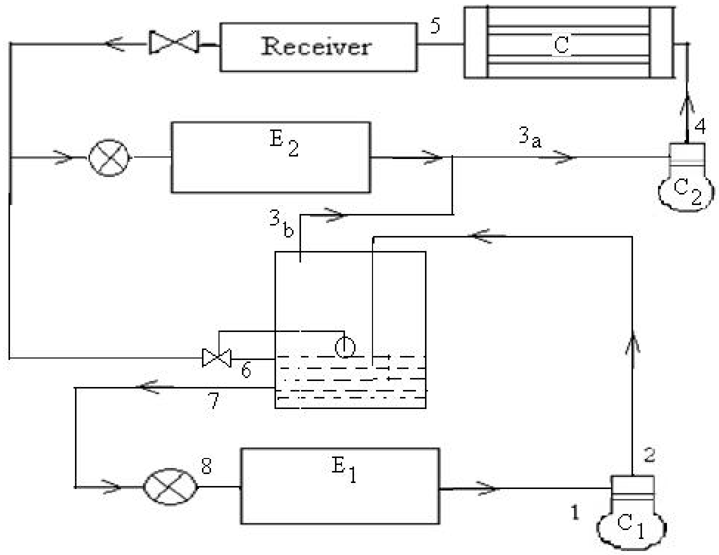

Fig.20.3: Multievaporator and individual compressor system on P-H diagram

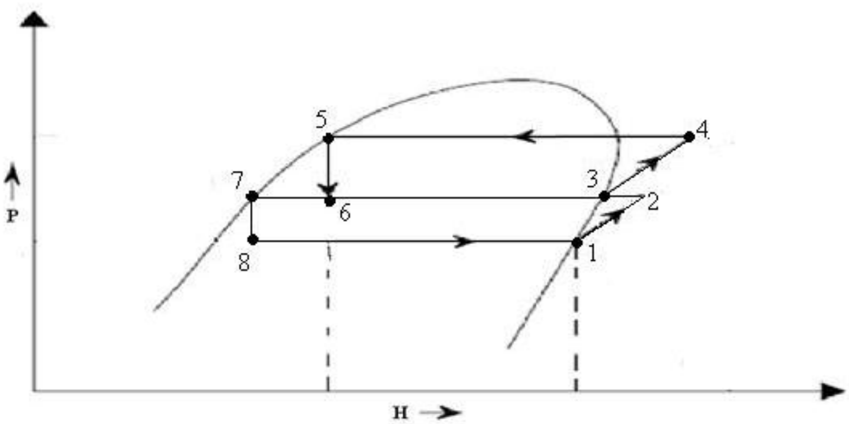

Fig.20.4: Multievaporator and compressor system on P-H diagram

The mathematical analysis is just similar to previous case. The condition of refrigerant leaving E_2 and the vapour coming from inter cooler is the same.

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Lesson 21.

Numerical On Multi-Evaporation And Compression Systems

1.1. Numerticals

a). A 10 ton ammonia vapour compression refrigeration system consists of one evaporator and two stage compression. The suction temperature is -30 ^{o}C and condensing temperature of 35 ^{o}C. Flash inter-cooling is done between two stages of compression. Find theoretic kW of each compressor and COP of the plant.

Solution: The working cycle is represented on P-H diagram.

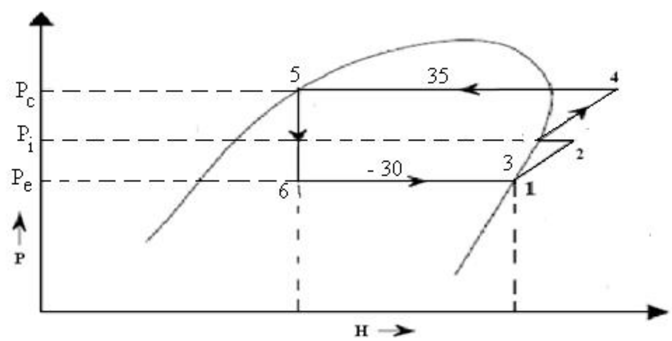

Fig. 21.1: Cycle on P-H diagram

From R-717 (ammonia) Table data,

And P_c = 13.5 bar at +35 ^{0}C

P_e = 1.2 bar at -30 ^{o}C

Therefore, $P_i = \sqrt{P_e \times P_c}$

$= \sqrt{1.2 \times 13.5}$

= 4.0 bar (The intermediate corresponding temperature = -2 °C)

h_1 = 1405.6 kJ/kg (From table)

h_2 = 1557.0 kJ/kg (from chart)

h_3 = 1442.5 kJ/kg (From table)

h_4 = 1617.0 kJ/kg (From chart)

h_5 = 347.5 kJ/kg (From chart)

Mass flow rate of refrigerant, $m = \dfrac{3.5 \times 10}{h_1 - h_5}$

$= \dfrac{3.5 \times 10}{1405.6 - 347.5}$

= 0.0331 kg/s

kW of Low Stage Compressor = 0.0331 × (h_2 - h_1)

= 0.0331 × (1557.0-1405.6)

=5.01 kW

The flash refrigerant required (m_i) at intermediate pressure (4 bar at -2 °C) for inter-cooling can be calculated as under. This refrigerant is also compressed by higher stage compressor.

$m (h_2 - h_3) = m_i (h_3 - h_5)$

∴ 0.0331(1557-1442.5) = mi (1442.5-347.5)

∴ mi = 3.46 × 10-3 kg/s

The total amount of refrigerant compressed by higher stage compressor is as under.

= 0.0331 kg/s + 0.00346 kg/s

=0.03656 kg/s

"kW of higher stage compressor = 0.03656 × (h_4- h_3)

= 0.03656 (1617.0 - 1442.5)

= 6.38 kW

COP= (3.5 ×10)/(5.01+6.38)

∴ COP = 3.07

b) Multi-evaporation and multi-compression ammonia vapour compression refrigeration system is used in commercial plant (Figure given below). The condensing temperature of the plant is 30 °C. The intermediate pressure between two stages of compression is 3.8 bar (absolute). Find the theoretical C.O.P. and work of compression of each compressor.

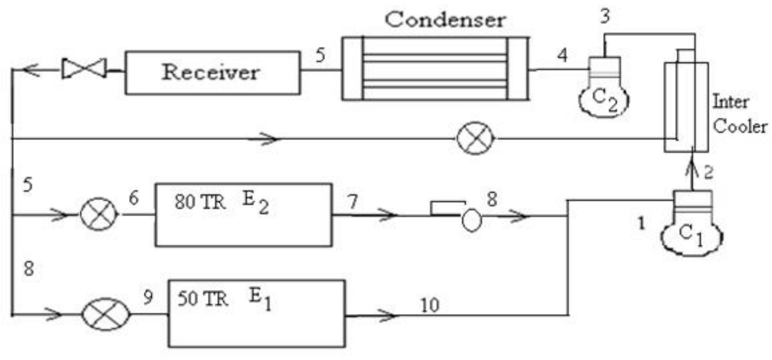

Fig.21.2: Multievaporator refrigeration system

Solution: The working cycle of the above system is indicated on P-H diagram

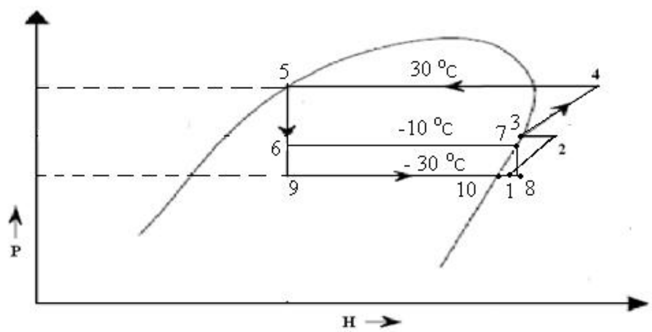

Fig.21.3: Multievaporator refrigeration system on P-H diagram

From the properties tables & chart

h_{10} = 1405.6 kJ/kg

h_8 = 1433.1 kJ/kg

h_3 = 1441.0 kJ/kg

h_2 = 1450.0 kJ/kg

h_4 = 1595.0 kJ/kg

h_5 = 323.1 kJ/kg

$$\text{Mass Flow Rate of Refrigerant In } E_1,\ m_1 = \frac{50 \times 3.5}{h_{10} - h_9}$$

$$= \frac{50 \times 3.5}{1405.6 - 323.1}$$

$$= \mathbf{0.1617\ kg/s}$$

$$\text{Mass Flow Rate of Refrigerant In } E_2,\ m_2 = \frac{80 \times 3.5}{h_7 - h_6}$$

$$= \frac{80 \times 3.5}{1433.1 - 323.1}$$

$$= \mathbf{0.2523\ kg/s}$$

Enthalpy h_1 can be calculated by heat and mass balance of refrigerants of E_1 and E_2 as under.

$m_1\ h_{10} + m_2\ h_7 = (m_1 + m_2)\ h_1$

$(0.1617 \times 1405.6) + (0.2523 \times 1433.1) = (0.1617 + 0.2523)\ h_1$

$227.29 + 361.57 = 0.414\ h_1$

$h_1 = 1422.37$ kJ/kg

The h_1 point can be located on P-H diagram corresponding to enthalpy 1422.37 kJ/kg.

kW of first stage compressor = $(m_1 + m_2) \times (h_2 - h_1)$

$= (0.1617 + 0.2523) \times (1450.0 - 1422.37)$

= 11.44 kW

The refrigerant used for inter-cooling between two stages of compression can be estimated as under.

$(m_1 + m_2) \times (h_2 - h_3) = m_i\ (h_3 - h_5)$

$(0.1617 + 0.2523) \times (1450.0 - 1441.0) = m_i\ (1441.0 - 323.1)$

$0.414 \times 9 = m_i\ (1117.9)$

$m_i = 0.00333$ kg/s

Second stage compressor compresses $m_1 + m_2 + m_i$ quantity of refrigerant.

kW for second compressor = (m_1 + m_2 + m_i) × (h_4 - h_3)

= (0.1617 + 0.2523 + 0.00333) (1595.0 - 1441.0)

= 0.4173 × 154

= 64.26 kW

"COP = " "3.5 (50 +80)" /"(11.44 +64.26)"

= 6.0

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Lesson 22. Simple Absorption Refrigeration System

22.1. INTRODUCTION

The Refrigeration by mechanical vapour compression system is an efficient method. But, the energy input is the shaft work, which is a high – grade energy (One that can be easily converted to other forms)and therefore very expensive. And the work required is relatively large because of compression of the vapours which undergo large changes in specific volumes. If same gas is available in liquid form, to pump that to higher pressure, the energy required is less. Hence in order to achieve this, the system designed is called Absorption Refrigeration system, in which the refrigerant vapour is dissolved in an inert liquid at the same pressure as the evaporator and the solution so formed is pumped to a container at condenser pressure. This liquid which is practically incompressible and undergoes very little change, in specific volume, requires very little work in raising its pressure. After raising the pressure, the refrigerant is separated from the solution by heating, and increasing the temperature.

The shaft work involved in both the cases, if you campare for compressing and just pumping for a particular case was 1.225% to that of V.C system.

Combination

Absorbent Refrigerant

Water Ammonia

Lithium bromide Water

22.2. SIMPLE ABSORPTION REFRIGERATION SYSTEM

The essential components are: an Evaporator, an absorber, a generator, a condenser, an expansion value, a pump and a reducing value.

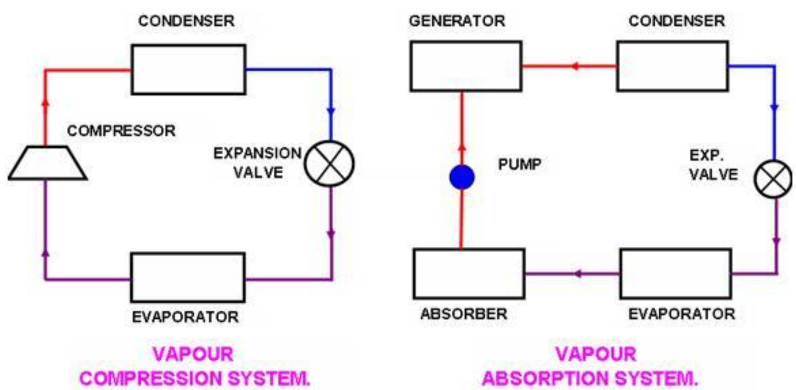

Fig. 22.1: Vapour compression and vapour absorption system

Table 22.1. Comparison of vapour absorption and compression

Absorption	**Compression**
1. Moving part is only a pump, which in some cases is also eliminated	The compressor is more noisy than pump and consumes more power.
2. Any low grade energy can be used	Only shaft work can be used in compressor, which is high grade energy.
3. Reduced evaporator pressure does not reduce the capacity. Just increase the steam supply in situations where lower temperature than designed is required.	Capacity reduced drastically when at low evaporator pressure.
4. The efficiency does not change much with load. Quantity of aqua circulated is controlled.	The efficiency goes down
5. Liquid refrigerant leaving evaporator has little impact in the system (due to sudden load fluctuation.	Detrimental to compression, especially when load fluctuations are frequent.
6. Absorption systems, except for domestic refrigerators occupy more space, per T n of refrigeration effect.	Occupation less space per T on of refrigeration.
7. Cheaper, especially, when electricity cost is higher and at lower approximate. temperature.	Costly operation, especially at lower evaporation temperature

Thus it is seen that the compressor in the VC system is replaced by an absorber, generator, a reducing value and a pump.

The choice of the absorbent depends on its -vc deviation from, Raoult's law (At a given temperature the ratio of the partial pressure of a volatile component in a solution to the vapour pressure of the pure component at the same temp is equal to its mole fraction in the solution. The mole fraction is equal to the number of moles of the component divided by the total no. of moles present. Mol = wt. of substance/ mol.wt . A negative deviation is said to be there when observed vapour pressure is less than the calculated pressure.)

In case of Ammonia and water, the total volume is approximately equal to the volume of water above plus 85% of volume normally occupied by the Ammonia. The combination of water and ammonia is called Aqua Ammonia. Is Absorption of ammonia to water is a chemical reaction?

22.2.1. Absorber: It may be in a shell, in which weak solution containing considerably less ammonia is sprayed or imposed and which absorbs ammonia from evaporator. Absorption of ammonia lowers the pressure in the absorber and as a result more ammonia vapour in drawn from the evaporator, creating lower pressure in evaporations.

There is some heat evolved due to vapours going into liquid and the heat due to condensation. To take this heat away, there will be a cooling water coil inside. The stronger solution is pumped by a pump into the generator, which is at a higher pressure than absorber.

22.2.2. Generator: In generator heat is added to the system by some heat source, may be a gas burner, steam or electric heating. Connections are provided for weak solution from generator and return to absorber through a reducing value.

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LESSON 23.

PRACTICAL VAPOUR ABSORPTION SYSTEM

23.1. INTRODUCTION

To improve the above simple system, especially to prevent water going along with ammonia vapours into condensor, to get more economy of heat, certain changes are made.

Analyser and Rectifier are added to reduce the passage of water into condenser.

23.2. ANALYSER

Analyser is usually an arrangement of trays and may be an integral part of the generator itself. Here, both the strong solutions from absorber and aqua from rectifier are dropped and allowed to flow on the trays and then get exposed and cooled. This partial cooling condenses water vapour and only ammonia vapour leaves at the top

23.3. RECTIFIER

Rectifier is a double pipe, shell and coil or shell and tube type of vapour cooler. It is usually water cooled. Cooling is just sufficient to remove all the water and leave only ammonia vapours in the condenser. Temp is almost 100 – 120^0 C.

23.4. HEAT EXCHANGER

This is located between absorber and generator, and serves to cool the weak aqua by heating the strong aqua. Double pipe system is used in small system, while shell and tube is used in large system. This will reduce the amount of heat added in the generator and decrease the amount of heat rejected in the absorber. The sizes of the generator and absorber will also be reduced.

23.5. DESIRABLE CHARACTERISTICS OF REFRIGERANT - ABSORBENT PAIR

(1) Low viscosity to minimize pump work

(2) Low freezing point

(3) Good chemical and thermal stability. Also, the two main thermodynamic requirements.

23.5.1. Solubility Requirement : - The refrigerant should have more than Raoult's law solubility in the absorbent so that a strong solution, highly rich in the refrigerant, is formed in the absorber by the absorption of the refrigerent vapour.

23.5.2. B.P.Requirement: - There should be a large difference in the normal binding points of the two substances; at least 200^{0}C, so that the absorbent exerts negligible vapour pressure at the generator temperature. Thus, almost absorbent free refrigerent is expelled off from the generator and the absorbent alone return to the absorber.

Ammonia forms a highly non ideal solution in water. So it satisfies solubility requirement. But B.P temperature. Difference is only 138^{0}C. So vapour leaving generator contains some amount of water, which creates problem.

In Lithium Bromide – water system, water is the refrigerant. Hence, it is only used in air conditioning application. Since Lithium Bromide is of selt, it exerts no pressure. But it is corrosive and plant works in high degree of vacuum.

23.6. Maximum COP of Vapour Absorption System

VA system is heat operated refrigerating machine. It may be considered as a combination of a heat engine and refrigerating machine.

The energy supplied to the system is in the form of heat Q_g at T_g. The Thermodynamic cycle is considered to comprise heat engine cycle, operating between the heat source temperature Tg and the temperature of heat rejection Tc, and a refrigeration cycle operating between the refrigeration temperature Tr and temperature of heat rejection Tc. The work done in the heat engine part of the cycle is equal to the work requirement of the refrigeration part of the cycle. Thus one may write for the COP of the cycle.

COP = $$\frac{Q_e}{Q_g} = \frac{W}{Q_g} \times \frac{Q_e}{W} = n_{th} * Cop_R$$

The COP of a heat operated refrigerating machine should, therefore, be minimum when each of the two terms has a maximum value which unloads so when both are equal to their respective Carnot values.

$$Cop_{max} = n_{th_{carnot}} \times Cop_{R_{carnot}} = \left(\frac{T_g - T_c}{T_g}\right)\left(\frac{T_e}{T_c - T_e}\right)$$

= $$= \left(1 - \frac{T_c}{T_g}\right)\left(\frac{1}{\frac{T_c}{T_e} - 1}\right)$$

In order that this should be high,

1) Temperature T_g of the heat source should be as <u>high</u> as possible.

2) Temperature T_c of the heat sink is as <u>low</u> as possible.

3) Temperature T_e of the refrigeration is as <u>high</u> as possible.

It may be noted that in case the condenser and absorber temperature are not the same and are equal to T_c and T_A respectively, then the minimum possible COP is given by

However, because of the presence of the H, and the partial pressure it exerts on the low pressure side of the system (Absorber and evaporator) the partial pressure exerted by the NH_3 vapour in these parts will be lower than that exerted by the ammonia vapour in the generator and condenser, when hydrogen is not present.

23.7. ABSORBENT SYSTEMS

23.7.1. Desirable characteristics:

1. Solubility requirement: The refrigerant should have more than Raoult's law solubility in the absorbent so that a strong solution, highly rich in the refrigerant, is formed in the absorber by the absorption of the refrigerant vapour.

2. Boiling points requirement: There should be a large difference in the normal boiling points of the two substances, at least 200^0C, so that the absorbent exerts negligible vapour pressure at the generator temperature. Thus almost absorbent – free refrigerant is boiled off from the generator and the absorbent alone returns to the absorber.

In addition to these, some of the other less important ones are

1. Low viscosity – to minimize pump work
2. Low freezing point
3. Good chemical & thermal stability – decomposition, polymerization, corrosion etc.

In the ammonia – water systems, ammonia is the refrigerant and water is the absorbent. Ammonia forms a highly non – ideal solution in water. Hence it is satisfactory from the point of view of the solubility requirement. But the difference in their Boiling point is only 133^0C. Hence the vapour leaving the generator contains some amount of water which results in many problems. Thus the ammonia – water system is not suitable from the point of view of the boiling point requirement.

In the lithium bromide – water system, water is the refrigerent and lithium boroxide the absorbent. Hence the mixture is used only in air – unlimited applications. The mixture is again non – ideal and is satisfactory from the point of view of the solubility requirement. Since lithium bromide is a salt, it exerts no vapour pressure. So the vapour leaving the generator is a pure refrigerant. The mixture satisfies the boiling point requirements also. However, it is corrosive and the plant work under high vacuum.

23.8 . USING ANALYZER – THE EXHAUSTING, COLUMN, AND DEPHLEGMOTOR – THE RECTIFYING COLUMN

In a system like that of ammonia – water, the vapours distilled from the generator contain a considerable amount of absorbent vapour which subsequently reaches the evaporator after condensation. As a result, evaporation would not be isothermal and the required low temperature would not be reached.

Not only is the evaporator pressure reduced due to concentration being less, it can be seen that the rich solution concentration at the given absorber temperature is also reduced, because of lower absorber pressure. Thus will ultimately result in a smaller difference in the rich and poor solution concentrations and hence a lower COP.

Hence, to return the absorbent to the generator and to allow, as for as possible only the refrigerant vapour to enter the condenser, two elements are added,

(1) Analyzer or the exhausting column.

(2) Dephlegmator and rectifier on the rectifying column.

The analyzer on the exhausting column is installed on top of the generator. The vapour learning the generator with certain ref. concentration in equilibrium with the boiling poor solution having certain concentration, enter this analyzer. As it travels upwards, counter flow to the entering rich solution, the vapours encounters heat and mass exchange with then falling rich solution ultimately learning their analyzer enriched in the refrigerant.

This method has the additional advantage of returning some heat from the vapour to the generator in the form of preheating of the rich solution with simultaneous cooling of the vapour.

The enriched vapour now enters the rectifying column where in heat is removed from the vapour by the circulation of the cooling medium. The leaving state of the vapour is determined by the temperature of the cooling medium. A part of the vapour is condensed and is returned as drip to the analyzer.

The latter method of increasing the refrigerant concentration of the vapour has a drawback in that it involves a loss of useful heat added in the generator which is rejected to the cooling medium in the rectifier. The drip returns to the generator and has to be evaporated again. The condition under which the use of rectifier would improve the COP would depend on the working pair being used and the operating conditions.

It may be noted that the use of an analyzer and rectifier is not necessary in the case of system such as lithium bromide – water in which case the absorbent or adsorbent doesn't it exist any significant vapour pressure, if at all.

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LESSON 24.

PROPERTIES OF REFRIGERANT – ABSORBENT COMBINATION

24.1. INTRODUCTION

1. Refrigerant should be much more volatile than absorbent.

2. Refrigerant properties must provide moderate +ve pressures same as VC

3. Both should be chemically stable at all operating conditions same as VC

4. They should not form solid phase in the operating conditions same as VC

5. Absorbent should have strong affinity for refrigerant.

6. Both should not cause corrosion in the range of conditions same as VC

7. Should not be toxic and inflammable.

8. Low viscosity to promote heat and mass transfer.

9. Refrigerant should have high latent heat to reduce mass flow same as VC rate.

10. Both must be completely miscible in liquid and vapour phases and no range of concentration values where a heterogeneous mixture would exist.

24.2. SIMPLE VAPOUR & IMPROVED VAPOUR

1. Lot of heat need to bring strong liquor to Generator temperature.
2. Hot weak liquor had lesser capacity to absorb NH_3 & cooling water will be needed more.
3. Substantial carry – over of water with NH3 from generator to condenser. This water raises the boiling point higher in Evaporator .

Table 24.1.

NH_3 – H_2O	**H_2O – Li Br**
1. Operates even below 0^0 C	1. Always above 0^0 C
2. Operates at +Ve pressure	2. Operates at –Ve pressure (Requires leak detector. Mass spectrometer)
3. Completely in liquid condition	3. Care to be taken to operate only in liquid range and avoid crystallization
4. Absorbent is volatile	4. Absorbent is non volatile in the operating range.
5. Rectifier and Analyser required.	5. Not required.

6. No problem of crystallization	6. Anti crystallization and anti corrosion agents needed.
7. COP is lower comparatively because low Evaporator .Temperature and Reflux could. reduces mass of refrigerant circulated	7. COP is higher
8. Solution of R – A is low in viscosity.	8.Solution R- A is high in viscosity
9. Can be easily Solar Driven at T_g 80^0C.	9. Use of solar Energy is difficult as T_g required is about 90 – 100^0C
10. Can be air cooled.	10. Usually water cooled to avoid crystallization.
11. System work under high pressure, hence sp.vol. is very low for NH_3. Therefore for same working capacity, this system smaller size	11. Work and vacuum conditions and sp .Volume of H_2O in very high. Hence, for same cooling capacity, the system is large in size.

VAPOUR ABSORPTION	**VAPOUR ADSORPTION**
1. The absorbate and absorber form a true solution	1. The adsorbate and adsorbent do not form a true solution
2. The extent of absorption depends on concentration pressure and temp.	2. The extent of adsorption depends on surface area, access of inner areas.
3. Mostly liquid.	3. Mostly Adsorbent is solid, though liquid and gas are also used
4. There is no surface specificity	4. Only at surfaces and some times at specific sites of the surfaces
5. Separation occurs due to affinity.	5. Separation occurs in addition to affinity, due to difference in mol. Wt., shape or polarity
6. Large molecular absorbate may also be absorbed	6. Mostly small molecular adsorbate are attached.
7. Attained some popularity in domestic refrigeration	7. Because of bulky requirement and valves, did not attain popularity in domestic refrigeration.
8. Application: Refrigeration, heat pump.	8. Applications: Paints,
9. Examples: NH_3/H_2O, H_2O – Li Br_2	9. Carbon, Silica gel, Alumina, Zeolite
10. There is not much increase in the combined volume.	10. It swells to about 10times its original volume.
11. Not much degeneration of absorption capacity on repeated use.	11. The adsorbent tends to lose its crystalline structure, and breakup into a powder which makes subsequent absorption more difficult. The powdered adsorbent is packed into regions less accessible to NH_3

12. COP	12. COP
13. VARS requires values, rectifiers etc.	13. Adsorption refrigeration does not require valves, rectifiers etc.

NH_3 – H_2O	**H_2O – Li Br**
1. Considerable flexibility of design for any configuration of specific application.	1. Prepackaged, standard size unit
2. Internal flow distribution & pressure drop restriction are less severe.	2. Internal flow distribution and pressure drop restriction are closely controlled.
3. Material used are usually steel.	3. Materials used are copper or copper nickel,
4. Design working pressure in high at 1200 kpa and 1720 or low and high side respectively.	4. Design working pressure in low at 103 and 516 kpa for low and high side.
5. Can be transported in diff. component, to site	5. Usually transported in one piece to site

VAPOUR COMPRESSION	**VAPOUR ABSORPTION**
1. Compact	1. Occupies more space
2. Moving parts need maintenance	2. Very little moving parts and hence maintenance is easy, & the operation is noiseless, vibration less also
3.High grade energy is required energy required is comparatively more	3. Heat, a low grade energy can also be used. Only 5% energy is needed compared to Vc system, as the liquid is incompressible needs less energy to transport & build in pressure.
4.Electricity rates are likely to increase. Only lower electric rates favour this.	4.System can be made electricity independent
5. Any other form of energy cannot be used	5. Waste heat, from exhaust gases, burning of agriculture and other wastes can be used
6.COP gets reduceed at reduced loads	6.COP reduction is negligible at reduced loads
7. Capacity falls as Evaporator temp. decreases (Evp. Temp may differ as per season)	7. Capacity fall is negligible even at low Evap. Temp. (by increasing Generator temp. it can be balanced)
8. System is electricity dependent	8. Higher electricity rates and lower fuel rates favour this.
9. Cannot use Solar Energy efficiently (Type of Evaporator is the same, in either of the system)	9. Can use Solar energy efficiently.
10. Energy consuption is high if electric generation is considered	10. Energy consumption is comparable lesser of at generation of electricity is considered
11. Initial investment is low	11. Initial investment is high. (Pay now save later)
12. Used for cooling mostly	12. It can be used as Heat pump, both is summer and winter,
13. COP =	13. COP of heat operated ref. machine is equal to the product of the thermal efficiency of heat engine

1 hp – 0.75 kw	part of cycle and COP of for cooling of the this refrigerating part of the cycle. COP = = 0.4 – 0.5 NH_3 – H_2O 0.47 LiBr – H_2O =
14. 100 TR – 15^{0}C plant needs 187.5 kw hp motor for compressor. It the same 100 TR to -20^{0}C, the capacity falls to 16 TR	14. 100 TR – 15^{0}C plant needs 11.25 kw hp motor for pump. Is the same 100 TR to – 20^{0}C, the capacity will be 100 TR, with steam pr. Requirement slightly higher

Analysis of Aqua-Ammonia Refrigeration System

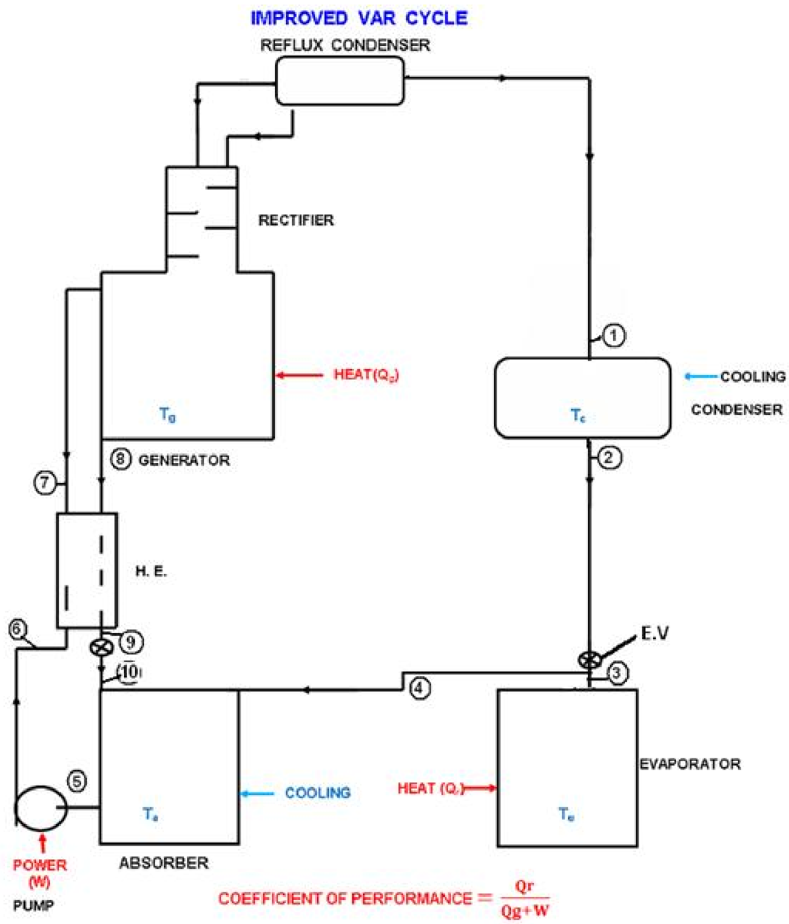
The analysis and design of aqua-ammonia refrigeration system, can be done by representing the processes and condition of aqua on X-h (concentration-ethalpy) chart at various components of the system. The input and output of the system being in the form of heat (enthalpy) only, the system performance can be calculated once the enthalpies at different points are known.

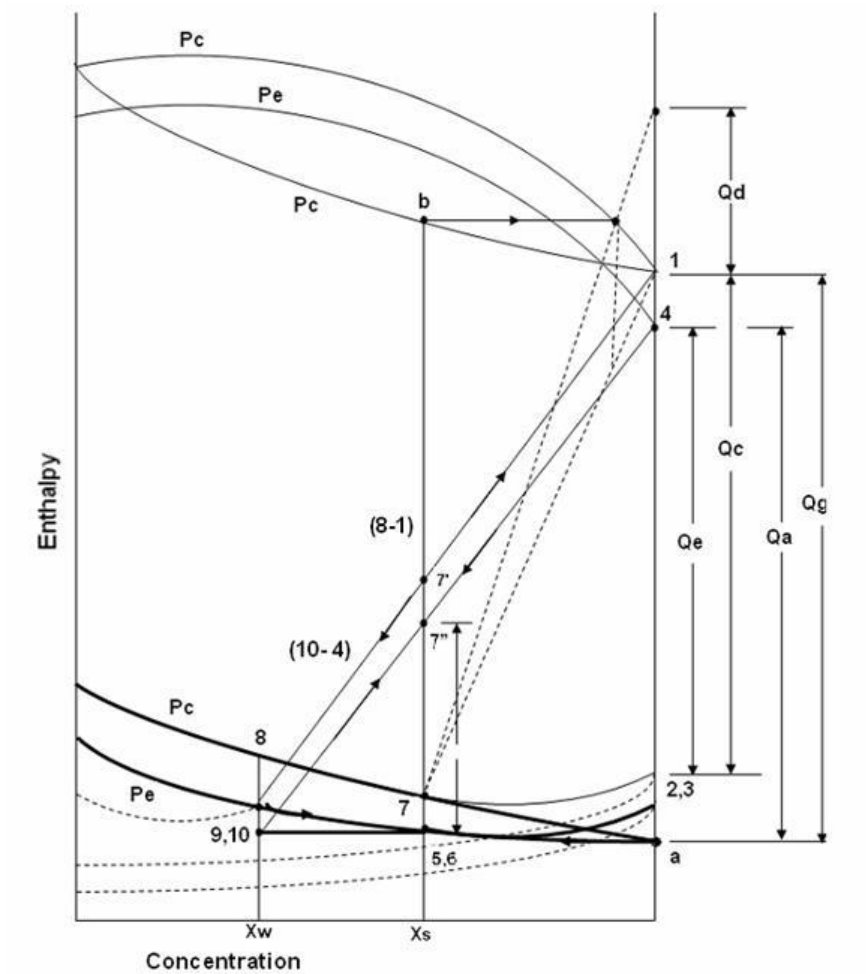
Once the temperatures at condenser & evaporator are known and the load to be taken by the system is known, then we can decide the circulation of aqua as well as pure NH_3 to be used in the system. Design of the sizes and arrangements of the different components of the system will follow.

Representation of Absorption system on Concentration-Enthalpy chart of Ammonia - Water

The working components of the system and concentration chart are shown in Fig. (a) and Fig: (b). The X-axis represents concentration (X) and Y-axis represents enthalpies in kJ/kg. The concentration is defined as amount of NH_3 in kg per kg of mixture. The lower part of the diagram is for the mixture of NlH_3 & water (liquid form) whereas the upper part of the diagram corresponds to NH_3 & water (vapour form) mixture. Both the parts of the diagram are for the same pressure lines (full lines). On the lower part of the diagram, constant temperature lines (dotted) are also drawn. In addition to this, auxiliary lines are also drawn as shown in Figure which are only pressure lines. They do not help to get any properties of solution of gaseous mixture but their use is to locate the points in gaseous mixture as per given conditions.

1) The concentration X = 0 means, there is no NH_3 and mixture is pure water.


IMPROVED VAR CYCLE
REFLUX CONDENSER
RECTIFIER
HEAT(Qg)
Tg
GENERATOR
COOLING
CONDENSER
Tc
H. E.
E.V
EVAPORATOR
ABSORBER
COOLING
Ta
HEAT (Qr)
Te
POWER (W)
PUMP
COEFFICIENT OF PERFORMANCE = Qr / (Qg+W)



X -h Chart for aqua - Ammonia Refrigeration System.

(2) The concentration X = 1 means, there is no water and mixture is pure NH_3

(3) If the concentration X = 0.4 (say) & pressure is 12 bar and then the temperature of the mixture will be 90°C, (from chart) if the mixture is saturated as it will have only one & one temperature at saturated condition. But if the temperature of the aqua is less than 90°C, then it indicates that the aqua NH_3 is at sub-cooled condition.

The basic components of NH_3 - absorption system and the conditions of aqua at different components of the aqua NH_3 system.

Point -1 This represents pure NH_3 - saturated vapour at condenser pressure (P_c) having a concentration X = 1. This point can be marked on X-h chart in vapour region as X & P_c are known.

Point - 2 This represents pure NH_3 - saturated liquid at P_c & X = 1. This point is marked in liquid region.

Point - 3 This represents the condition of pure NH_3 (wet) but at pressure P_c (evaporator pressure) but X = 1 and h remains constant as 2-3 is throtting process. The point - 2 also represents point 3.

Point - 4 This represents the condition of pure NH_3 at pressure P_e but saturated vapour which absorbs heat in the evaporator and converts from wet vapour to saturated vapour. This point is marked in vapour region.

Point - 5 This represents the strong aqua-solution coming out of absorber after absorbing vapour coming in from evaporator. Say X_s (concentration of strong solution = 0.6) and pressure is P_e. This point can be marked as X_s & P_e at point 5 are known. It is always assumed as saturated aqua at point '5' if not mentioned.

Point - 6 This is the condition of the aqua solution whose $X_s = 0.6$ (say) but the pressure is increased from P_e to P_c as it passes through the pump. when the aqua pressure increases passing through the pump.

Point - 7 As the strong low temperature aqua solution passes through heat exchanger, it gains the heat and its enthalpy increases but its concentration C_s remains same as well as pressure Pc also remains same.

Point - 8 The point '8' represents hot weak liquid (X_w) and this point can be located under the following two conditions.

(1) If degasifying factor is known. The degasifying factor is the amount of NH_3-gas removed from the strong liquid entering in the generator maintaining the pressure constant by supplying the heat Q_g (per kg of pure NH_3) in the generator. This gasification factor lies between 0.05 to 0.1 only even high gasification factor is desirable. This is because, higher gasification factor can also evaporate more water vapour & it creates troubles in evaporator & it is necessary to be removed completely before entering into the condenser.

If say the gasification factor is known and it is 0.1, then the concentration of aqua at point 8 = 0.6 - 0.1 = 0.5 as X_s (concentration at point 7) = 0.6 is known. The point '8' can be marked as $X_s = 0.5$ and pressure at the point 8 is P_c. The condition at point '8' can be considered as saturated liquid if not mentioned.

(2) If T_g (temperature in the generator) is known. The point 8 can be marked as it is the cross section of the pressure line P_c and temperature line T_g.

Point - 9 This shows the condition of weak liquid coming out of heat exchanger after giving heat to the strong solution so, the enthalpy is reduced. Deducting the heat lost by the weak solution in heat exchanger, the point 9 can be marked as concentration does not change.

Point -10 The point 10 represents the same enthalpy as at point 9 but at reduced pressure (p_e) as it passes through the pressure reducing valve.

Now join the points 8 and 7 and extend till it cuts to Y-axis l(b). Then join point-a and 5 and extend till it cuts the vertical line passing through '8' as shown. This also decides the position of the point 9 or 10.

Absorber

In absorber, the pure NH_3 gas enters at condition 4 and weak aqua solution enters at condition 10 and after mixing, strong aqua comes out at condition 5. The mixing occurring inside is undefined but aqua condition coming out is definitely known, Join the points 10 and 4 and extend the vertical line passing through point 7 till it cuts at point 7". Now we can say that mixing is taking place along the line 4-10 and at pressure P_c & resulting aqua is coming out at 5 after losing heat in the absorber. Joining the points 4 & 10 and marking point 7" is not necessary for solving the problems or designing the system components.

Generator

In generator, strong aqua is heated by supplying heat Q_g. The strong aqua enters into the generator at condition 7 and pure NH_3 vapour comes out at condition 1 and weak aqua at condition 8. Now join the points 8 and 1 and extend the vertical line through point 7 to mark the point 7' which cuts the line 1-8. Now, we can say that the separation is taking place along the line 1-8 and at pressure P_c. Joining the points 1 and 8 and marking the point 7' is not necessary for solving the problems or designing the system components.

Determination of Enthalpy change at Different Components of the System

The enthalpy change Q at different components and specific enthalpy 'q' are calculated in the following steps.

(1) Heat Removed in Condenser

The amount of heat removed in the condenser is given by

$Q_c = (h_2 – h_1)$ kJ/kg of NH_3.

As NH_3 saturated vapour enters in and NH_3 saturated liquid comes out.

This can be directly red from the X-h chart

(2) Heat Absorbed in the Evaporator

The amount of heat removed in the evaporator is given by.

$Q_c = (h_4 - h_3)$ kJ/kg of NH_3.

Where h_4 is the heat of saturated vapour at P_c and h_3 is the heat of mixture of NH_3 liquid & vapour at P_e or heat of NH_3 liquid at point '2' as 2 - 3 is constant enthalpy throttling process.

(3) Heat Removed from the Absorber

When NH_3 vapour at 4 and aqua at 10 are mixed, the resulting condition of the mixture in the absorber is represented by 7" and after losing the heat in the absorber (as it is cooled), the aqua comes out at condition 5. Therefore, the heat removed in the absorber is given by.

$q_a = (h"_7 - h_5)$ kJ/kg of aqua.

By extending the triangle Δ 10 - 7"- 5 towards right till 10 - 7" cuts at 4 and 10-5 cuts at point - a on y - axis. Therefore heat removed per kg of NH_3 is given by

$Q_a = (h_4 – h_a)$ kJ/kg of NH_3.

(4) Heat Given in the Generator

Q_g is the heat supplied in the generator and Q_d is the beat removed from the water vapour, then the net heat removed per kg of aqua is given by

$(q_g - q_d) = (h_{7'} - h_7)$ kJ/kg of aqua.

as the aqua goes in at 7 and comes out at condition 8 and 1 which can be considered a combined condition at 7'.

By extending the triangle Δ 8 - 7 - 7' towards right till 8 - 7' cuts at 1 and 8 - 7 cuts at 'a' on Y- axis then, the heat removed per kg of NH_3 is given by

$(Q_g - Q_d) = (h_1 - h_a)$ kJ/kg of NH_3.

Now for finding out Q_d separately, extend the verticle line 7 - 7' till it cuts the auxilary P_c line and mark the point 'b'. Then draw a horizontal line through 'b' which cuts the P_c line (in vapour region) at point 11. Then join the points '7 and 11 and extend that line till it cuts Y-axis at 12. Then the Q_d is given by

$(Q_d = (h_{12} - h_1)$ kJ/kg of Nh_3.

From the above two equations, Q_g can also be calculated.

Now Q_c Q_e Q_a Q_g and Q_d per kg of NH_3 flowing in the system will be known from the X-h chart.

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Lesson 25. Definition, Properties of Air Vapour Mixtures

25.1. INTRODUCTION

Air - conditioning signifies the control of an atmospheric environment either for human or to carry out industrial or scientific process efficiently. The purpose of air-conditioning is to supply sufficient volume of clean air having a specific amount of water vapour and at a required temperature within a selected space. The space may be a small compartment such as a research laboratory, computer laboratory or big area like cinema hall, shopping centre, air port etc. Thus, air conditioning refers the control of temperature, relative humidity, quality of air and distribution of air depending upon the application of air conditioning. The environmental air conditioned in terms of temperature, humidity, purity of air results in greater comfort to occupants when applied to public places , offices and factories . Air conditioning designed for industrial purpose has several benefits including better control of product quality and efficiency.

25.2. CLASSIFICATION

The air-conditioning systems are broadly classified into two groups.

1. Comfort air-conditioning

2. Industrial air-conditioning .

Air-conditioning system for human comfort provides an environment which is co m f ortable to the majority of the occupants . The comfort air-conditioning systems are subdivided into three g r ou p s . . .

25.2.1. Summer ai r -conditioning:

The initial quality of air decides the way of treatment of the air to condition the air for human comfort. The processes involved in treatment of air for air conditioning varies from place to place depending on the environmental quality of air. In many places, summer air-conditioning requires to reduce the s e ns ible heat and water vapour of the air by dehumidifying.

25.2.2. Winter air-conditioning

In winter, it is necessary to increase the se n si ble heat and water vapour content of the air by heating and humidif i cation .

25.2.3. Year-round air-conditioning

This system provides the control of temperature and humid i ty o f air throughout the year when the atmospheric conditions are changing a s per the season .

Indu s trial air-conditioning provides air at required temperature and relative humidity to perform a s pecific i n du s t rial p r oce ss s uccessfully . The de s ign condition s for industrial air conditioning are not based on th e comfort f eeling of th e human beings but on the requirement of the industrial process .

To understand the fundamentals of air conditioning, the basic knowledge of psychrometry is very essential.

25.3. PSYCHROMETRY

The science dealing with properties of air and vapour mixture is known as psychrometry. The knowledge of fundamental laws of gaseous mixture is necessary for understand i ng of psychrometry. The content of the water vapour in air is one of the factors responsible comfort air - conditioning. The subject of psychrometry is also important in many unit operations where air is used in the process. A few unit operations, where the knowledge of psychrometry is very essential for better understanding of the process, are listed below.

Ø Air heating system for spray drying plant.

Ø Performance evaluation of cooling tower.

Ø Performance evaluation of evaporative condenser.

Ø Evaporative cooling using water.

Ø Energy analysis casein or other types of dryers.

The properties of mixtures of air and water vapour are discussed in this lesson.

25.4. PSYCHROMETRIC PROPERTIES:

The definitions of different psychrometric properties of air are given below :

a) Dry air

The dry a ir is considered as a mixture of nitrogen and oxygen and small percentage s of other gases. The volumetric composition of air is 79% nitrogen and 21 % oxygen and the molecular weight of dry air is taken as 29 approximately .

b) Moist air

It is a mixture of dry air and water · vapour . The quantity of water vapour present in the a i r depend s upon the temperature of the air and its quantity may change from zero to maximum (the maximum amount depends on saturation condition).

c) Water vapour

The moisture present in the form of vapour is known as water vapour. The relative humidity of air is an important factor in all air - conditioning systems . .

The environmental (mixture of air and water vapour) is said to be saturated when it contain s m axi mum am o unt of water vapour that it can hold at the prevailing temperature. If the temperature of mixture of air and water vapour i s above the saturation temperature of the water vapour, the vapour is called superheated vapour .

d) Dry bulb temperature

The temper a ture of air measured by ordinary thermometer i s known as dry bulb temperature (dbt). : . .

e) Wet-bulb temperature

The temperature me asu red by the thermometer when its bulb i s c overed with wet c loth and is exposed to a current of moving air is known as wet bulb temperature (wbt) . The di f ference between the dry bulb and wet bulb temperature is known as wet bulb depression (wbd) . Wet bulb depression b e comes zero when the air is fully saturated.

f) Dew Point temperature

The temperature of the air is reduced by continuous cooling than the water vapour in the air will start condensing at a particular temperature. The temperature at which the · condensing starts is known as Dew , -point temperature. Dew point temperature is equal to the steam table saturation temperature corresponding to the actual partial pressure of the water vapour in the air . The difference between dry bulb temperature and dew point temperature is known as dew point depression (dpd).

g) Specific humidity (Humidity ratio)

It is defined as the mass of water yapour present per kg of dry air . It is expressed as g/kg dry air or kg/ kg dry air.

h) Absolute humidity

The weight of water vapour present in unit volume of air is known as absolute humidity.

i) Degree of Saturation

The degree of saturation is defined as the r atio of mass of water vapour associated with unit mass of dry air to mass of water vapour associated with unit mass of dry air saturated at the same temperature.

j) Relative Humidity

The relative humidity is defined as the ratio of actual mass of water vapour in a given . volume to the mass of water vapour if the air is fully saturated at the same temperature .

k) Sensible Heat of air

The quantity of heat which can be measured by measuring the dry bulb temperature of the air is known as sensible heat of the air.

i) Total heat of air

The total heat of the humid air is the sum of the sensible heat of the dry air and sensible and latent heat of water vapour associated with dry air.

m) Humid specific volume:

The volume of the air per kg of dry air in the mixture is known as humid specific volume of the air. It is expressed as m^3/kg dry air.

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Lesson 26. Psychrometric Chart and Its Application

26.1. INTRODUCTION
It is necessary to study the various psychrometric relations in order to understand the psychrometric chart.

26.2. DALTON'S LAW OF PARTIAL PRESSURE
According to Daltons Law of partial pressure the total pressure of a mixture of gases is equal to the sum of the partial pressures exerted by each gas when it occupies the mixture volume at the mixture temperature.

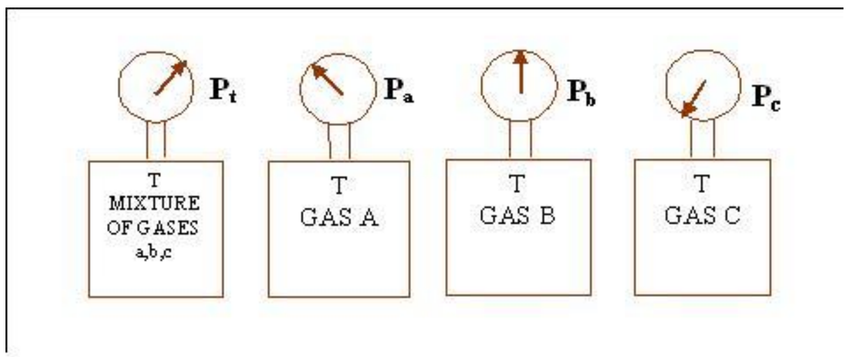

Fig. 26.1: Diagram explaining the principle of Daltons law

Gas A, B and C are filled in three different containers at temperature T and the partial pressures exerted by these gases are Pa, Pb and Pc respectively as shown in the Fig. 26.1. The total pressure (Pt) of the mixture of the gases will be equal to the sum of Pa, Pb and Pc. Mathematically,

$Pt = Pa + Pb + Pc$

If this law is applied to the moist air which contains dry air and water vapour, then
$Pt = Pa + Pv$
Where, Pt = Total pressure of moist air
Pa = Partial pressure of dry air

Pv = Partial pressure of water vapour of the air

Specific Humidity (ω):
As per the definition of specific humidity,
ω = Mass of water vapor in mixture / Mass of dry air in mixture = m_v/m_a
The mass of mixture is (ma + mv). The definition is given in terms of mass of dry air instead of total mass of mixture because the mass of mixture for a given volume of air changes with humidity.

The mass m_a is given as under.

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

Where, P_a is partial pressure of dry air and V is the volume of mixture and R_a is gas constant for dry air.

The mass m_v is given by the expressions as

$$m_v = \frac{P_v V}{R_v T}$$

Where, P_v is the partial pressure of water vapour of the air and R_v is gas constant of water vapour.

$$\omega = \frac{P_v V}{R_v T} \times \frac{R_a T}{P_a V} = \frac{R_a}{R_v} \times \frac{P_v}{P_a}$$

$$\text{But } R_a = \frac{\Re}{m_a} \text{ and } R_v = \frac{\Re}{m_v}$$

Where $\Re$ is universal gas constant and m_a and m_v are the molecular weights of air and water vapour.

$$\omega = \frac{M_v}{M_a} \times \frac{P_v}{P_a} = \frac{18}{29} \times \frac{P_v}{P_a} = 0.622 \frac{P_v}{P_a} = 0.622 \frac{P_v}{P_t - P_v}$$

26.3. DEGREE OF SATURATION (μ)

Where, ωs is the specific humidity of air when air is fully saturated.

$$\mu = \frac{\text{mass of water vapour in unit mass of dry air}}{\text{mass of water vapour in saturated unit mass of dry air}} = \frac{\omega}{\omega_s}$$

Where, ω_s is the specific humidity of air when air is fully saturated.

$$\therefore \mu = \frac{0.622\left(\dfrac{P_v}{P_t - P_v}\right)}{0.622\left(\dfrac{P_{vs}}{P_t - P_{vs}}\right)} = \frac{P_v}{P_{vs}}\left(\frac{P_t - P_{vs}}{P_t - P_v}\right) = \frac{P_v}{P_{vs}}\left[\frac{1 - \dfrac{P_{vs}}{P_t}}{1 - \dfrac{P_v}{P_t}}\right]$$

Where, Pvs is the partial pressure of water vapour when the air is fully saturated at the same temperature of the air. This can be calculated from steam table corresponding to the dry-bulb temperature of the air.

26.4. RELATIVE HUMIDITY (ø)

As per the definition of relative humidity,

As per the definition of relative humidity,

$$\phi = \frac{\text{mass of water vapour in a given volume}}{\text{mass of water vapour in same volume if saturated at the same temperature}}$$

$$= \frac{m_v}{m_{vs}} = \frac{\dfrac{P_v V}{R_v T}}{\dfrac{P_{vs} V}{R_v T}} = \frac{P_v}{P_{vs}}$$

Relative humidity plays very important role in comfort air conditioning and deciding the performance of many unit operations such as drying, evaporative cooling etc.

26.5. ENTHALPY OF MOIST AIR

The enthalpy of moist air is the sum of the enthalpy of one dry air and the enthalpy of water vapour associated with one kg dry air.

∴ Enthalpy of moist air = Enthalpy of one kg of dry air + Enthalpy of water vapour associated with one kg of dry air.

Thus,

$$h = h_a + \omega h_v$$

Where, h_a is the enthalpy of dry air and ωh_v is the enthalpy of water vapour associated with one kg of dry air.

$$h = C_{pa}(T_{db} - 0) + \omega\left[C_{pw}T_{dp} + (h_{fg})dp + C_{pv}(T_{db} - T_{dp})\right]$$

Where, C_{pa} = specific heat of dry air, kJ/kg K

C_{pw} = specific heat of water

T_{db} = Dry bulb temperature of air

W_{db} = Dew point temperature of air

$(h_{fg})dp$ = Latent heat of vaporization at dpt

C_{pv} = Specific heat of super heated vapour

The values of C_{pw} and C_{pv} used in deriving the equation are 4.186 kJ/kg K and 1.88 kJ/kg K respectively. The value of C_{pa} is taken as 1.005 kJ/kg K.

$$\therefore h = 1(T_{db} - 0) + \omega\left[4.18T_{dp} + (h_{fg})dp + 1.88(T_{db} - T_{dp})\right]$$

$$= T_{db} + \omega\left[(h_{fg})dp + 1.88T_{db} + 2.3T_{dp}\right]$$

$$= T_{db} + \omega\left[(h_{fg})dp + 1.88T_{db}\right] \text{ kJ/ kg dry air}$$

$$= 1.005T_{db} + \omega\left[2501 + 1.88T_{db}\right] \text{kJ/ kg dry air}$$

26.6. PSYCHROMETRIC CHART

Psychrometric chart shows the inter-relation of all the important properties of air. Using the chart, it is easy to obtained psychrometric properties of air and it helps to decide the various processes to be followed to achieve required quality of air. It saves lot of time and labour required for calculation of different properties of air and such values can be directly read from the chart.

Standard psychrometric charts are bounded by the dry-bulb temperature line (abscissa) and the vapour pressure or humidity ratio (ordinate). The left hand side of the psychrometric chart is bounded by the saturation line. Fig.26.4 shows the schematic of a psychrometric chart.

Psychrometric charts are readily available for standard barometric pressure of 101.325 kPa at sea level and for normal temperatures (0-50 oC). ASHRAE has also developed psychrometric charts for other temperatures and barometric pressures.

The constructional procedure is of psychrometric chart is based on the calculations for relative humidity, enthalpy, specific humidity etc. under different conditions using various

psychrometric relations.
The actual psychrometric chart is depicted in Fig. 26.2. It is necessary to know to parameters to locate the quality of air on psychrometric chart. After locating the point of quality, many other properties can be easily obtained from the chart. In majority of situations, dbt and wbt of the air are measured to find the location of the point on the psychrometric chart. If dbt and % Relative Humidity (RH) are known, then also it is possible to locate the point on the chart to read other properties of air.

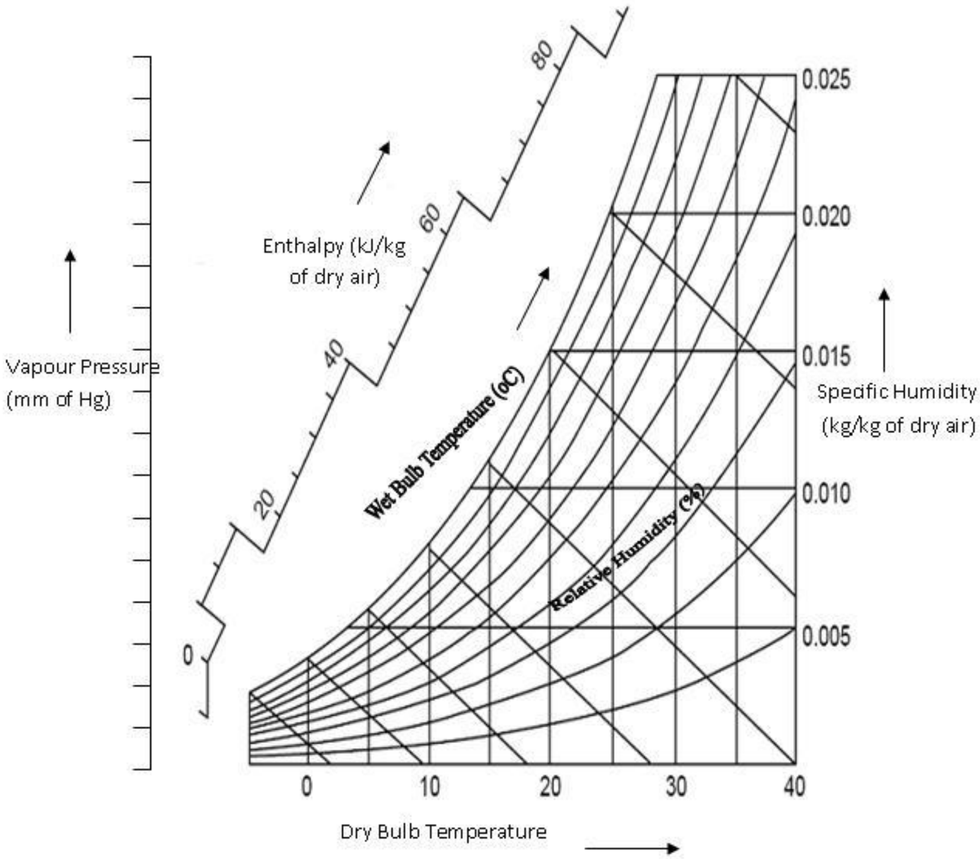

Fig. 26.2 Schematic psychrometric chart

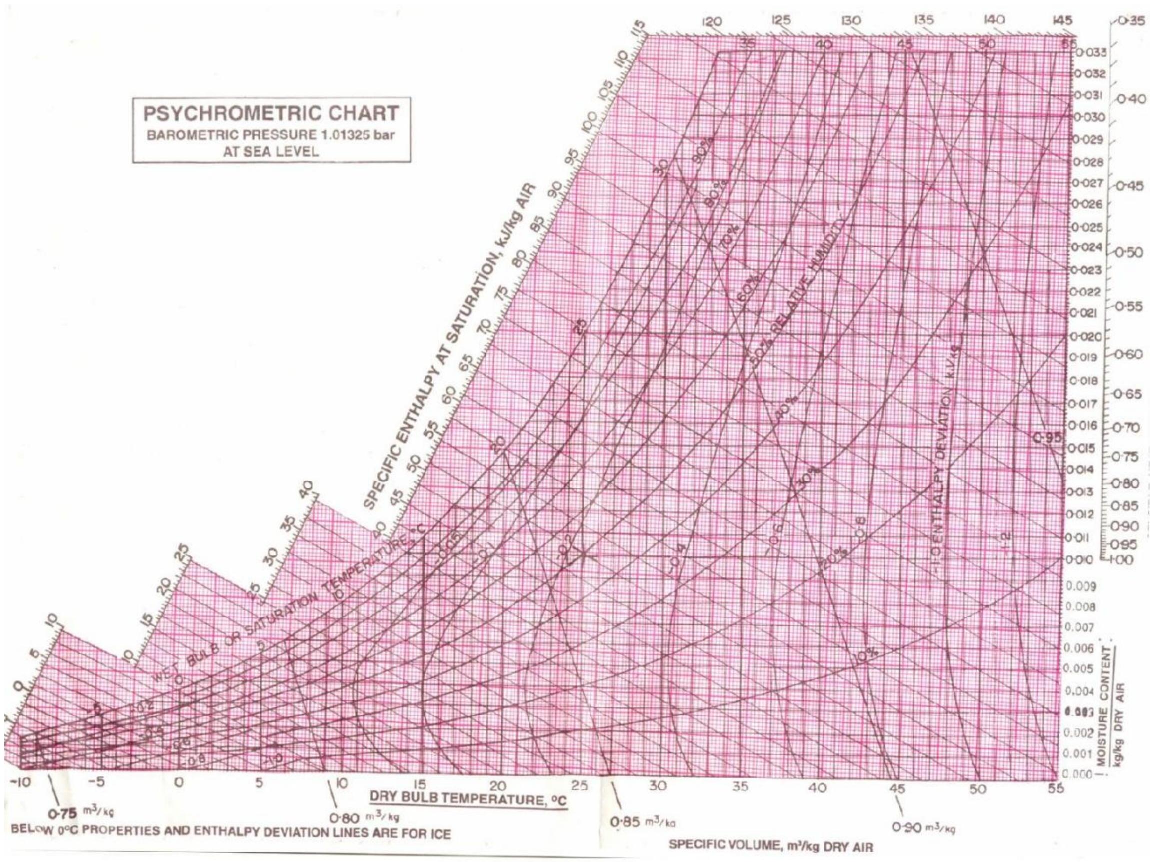

Fig. 26.3 Psychrometric chart

26.7. HOW TO USE THE PSYCHROMETRIC CHART?

Suppose, when the quality of air is measured by using dry bulb thermometer and wet bulb thermometer, the values obtained are: dbt = 30 oC and wbt = 20 oC

This point is located on psychrometric chart as indicated below and the values of specific humidity, relative humidity, enthalpy and dew point temperature are as under.

Specific humidity = 10.75 g/kg dry air

Relative humidity = 40 %

Enthalpy = 57.5 kJ/kg dry air

Dew point temperature = 15 oC

Partial pressure of water vapour of the air = 12.75 mm of Hg

The above values obtained from psychrometric chart can be also calculated by using various pychrometric relations. By knowing any two of the parameters the quality of the air can be located and all other parameters found out from the Psychrometric chart in the similar manner.

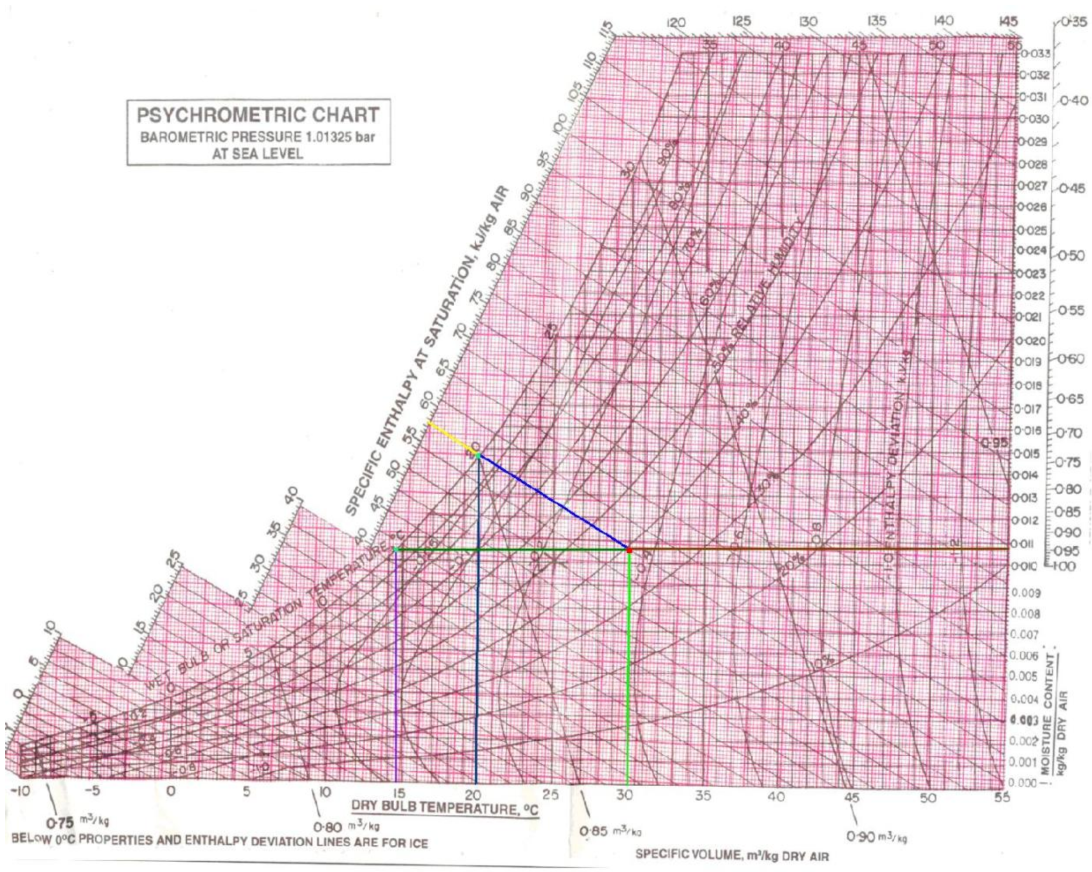

Fig. 26.4 Psychrometric chart Reading

Fig. 26.4 Psychrometric chart Reading

Many investigators have developed different expressions which relate Pv with WBT for given

DBT. The Apjohn equation which is commonly used for this purpose are given below.

$$P_v = (P_{vs})_{wbt} - \frac{1.8 p_t (T_{db} - T_{wb})}{2700}$$

Where,
- P_v = partial pressure of water vapor
- P_{vs} = partial pressure of water vapor at saturation (obtained from steam table corresponding to WBT of air)
- P_t = atmospheric pressure
- T_{db} = Dry bulb temperature
- T_{wb} = Wet bulb temperature

a) A laboratory psychrometer reads 40°C DBT and 28°C WBT. Calculate the following:

i) Specific humidity

ii) Relative humidity

iii)Enthalpy of the air

Solution: Using Apjohn Equation,

$$P_v = (P_{vs})_{wbt} - \frac{1.8 p_t (T_{db} - T_{wb})}{2700}$$

$$= 2.834 - \frac{1.8 \times 76 (40 - 28)}{2700}$$

$$= 2.834 - 0.608$$

$$= 2.226 \text{ cm of Hg}$$

$$\mathbf{= 0.0296 \ bar}$$

Specific humidity,

$$\omega = 0.622 \frac{P_v}{P_t - P_v}$$

$$= 0.622 \frac{0.0296}{1.03 - 0.0296}$$

$$= 0.0183 \text{ kg/ kg dry air}$$

$$\mathbf{= 18.3 \ g/ \ kg \ dry \ air}$$

Relative humidity,

$$\%RH = \frac{P_v}{(P_{vs})_{dbt}} \times 100$$

$$= \frac{2.226}{5.531} \times 100$$

$$\mathbf{= 40.24589 \ \%}$$

Enthalpy of air,

$$\mathbf{H = 1.005 \times dbt + \omega (2501 + 1.8 \times dbt)}$$

$$= 1.005 \times 40 + 0.0183(2501 + 1.8 \times 40)$$

$$\mathbf{= 87.2859 \ kJ/ \ kg \ dry \ air}$$

Lesson 27.

Psychrometric Processes - Sensible Heating, Sensible Cooling, Humidification, Dehumidification and Mixing of Air.

Psychrometric Processes:

It is necessary to carry out various processes on air in order to get the required quality of air. The quality of ambient air varies through out the year and by employing one or combination of processes, it is possible to get required conditions of the air.

1. Sensible Heating of air:

Heating of air without addition or subtraction of water vapour of the air is termed as sensible heating of the air. The sensible heating can be achieved by passing the air over heating coil like electric resistance heating coils or steam coils as shown in Fig. 27.1. The process is represented on the psychrometric chart in Fig. 27.2

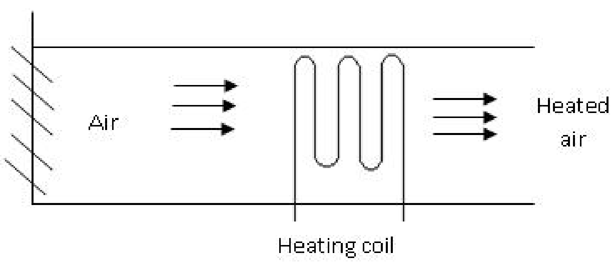

Fig. 27.1: Sensible heating of air

$$\text{Capacity of heating coil in kW} = \frac{V_a}{V_{sp}} \times (h_2 - h_1)$$

Where, V_a = flow rate of the air, m^3/s

V_{sp} = specific volume of air, m^3/kg dry air

h_1 = initial enthalpy of the air (kJ/kg of dry air)

h_2 = enthalpy of the heated air (kJ/kg of dry air)

The efficiency of heating coil is expressed as by-pass factor (B) of the heating coil.

$$B = \frac{t_3 - t_2}{t_3 - t_1}$$

The efficiency of the heating coil is better with lower bypass factor. Lower by-pass factor of heating coil is desirable to achieve higher efficiency of the coil.

Example:

An ambient air at 30 °C dbt and 22 °C wbt is heated to 180 °C dbt in an indirect steam coil heater at the rate of 20 m^3/min. Find the capacity of the heating coil in kW and the by-pass factor of the coil, if the surface temperature of the heating coil is 200 °C.

Solution:

Corresponding to 30 °C dbt and 22 °C wbt, the psychrometric properties are ,

$h_1 = 64.5$ kJ/kg of dry air

$\omega_1 = 0.0135$ kg/kg of dry air

$V_{sp} = 0.88$ m^3/kg dry air

$$h_2 = 1.005T_{db} + \omega[2501 + 1.88T_{db}]\text{kJ/ kg dry air}$$

$$= 1.005 \times 180 + 0.0135[2501 + 1.88 \times 180]$$

$$= 219.23 \text{ kJ/kg dry air}$$

$$\text{Capacity of heating coil in kW} = \frac{V_a}{V_{sp}} \times (h_2 - h_1)$$

t_1 t_2 t_3

$$= \frac{20}{0.88} \times (219.23 - 64.5)$$

$$= 3516.6 \text{ kJ/min}$$

$$= \mathbf{58.5\ kW}$$

$$\text{Bypass factor, } B = \frac{t_3 - t_2}{t_3 - t_1}$$

$$= \frac{200 - 180}{200 - 30}$$

$$= \mathbf{0.1176}$$

2. Sensible cooling of air:

Cooling of air without addition or subtraction of water vapour of the air is termed as sensible cooling of the air. Sensible cooling of air can be achieved by passing the air over a cooling coil like evaporating coil of the refrigeration cycle or secondary chilled water/brine coil as shown in Fig 27.3. Sensible cooling process is represented on psychrometric chart in Fig. 27.4.

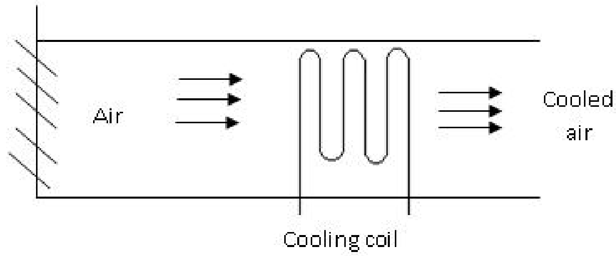

Fig. 27.3: Sensible cooling of air

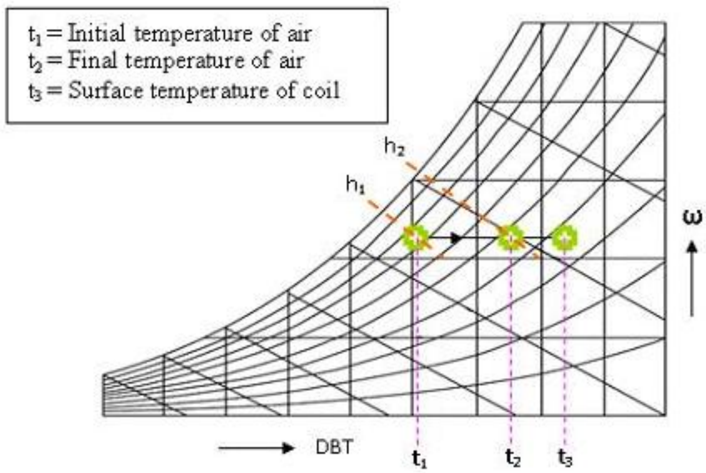

Fig.27.4: Sensible cooling process on psychrometric chart

The capacity of cooling coil can be estimated as under.

capacity of cooling coil in TR = V_a /V_{sp} x h_1-h_2 /3.5

Where, V_a = flow rate of the air, m^3/s

V_{sp} = specific volume of air, m^3/kg dry air

h_1 = initial enthalpy of the air (kJ/kg of dry air)

h_2 = enthalpy of the cooled air (kJ/kg of dry air)

The efficiency of cooling coil is expressed as bypass factor (B) of the coil.

B =t_2-t_3 / t_1-t_3

The efficiency of the cooling coil is better with lower bypass factor. Lower by-pass factor is desirable to achieve higher efficiency of the coil.

Example:

An ambient air at 40 °C dbt and 30 % RH is cooled 25 °C dbt & 21 °C wbt by a cooling coil maintained at 20 °C. The flow rate of the air is 10 m^3/min. Find the capacity of the cooling coil in ton and the by-pass factor of the coil.

Solution:

Corresponding to 40 °C dbt and 30% RH, the psychrometric properties are ,

h_1 = 76..5kJ/kg of dry air

ω_1 = 0.014 kg/kg of dry air

V_{sp} = 0.91 m^3/kg dry air

= 1.005 X 25+ 0.014[2501+1.88X25]

= 60.79 kJ/kg dry air

$$\text{Capacity of cooling coil} = \frac{V_a}{V_{sp}} \times (h_1 - h_2)$$

$$= \frac{10}{0.91} \times (76.5 - 60.79)$$

$$= 172.56 \text{ kJ/min}$$

$$= \frac{172.56}{210} \text{TR}$$

$$= \mathbf{0.8\ TR}$$

$$\text{Bypass factor, } B = \frac{t_2 - t_3}{t_1 - t_3}$$

$$= \frac{25 - 20}{40 - 20}$$

$$= \mathbf{0.25}$$

3. Dehumidification of air by cooling:

The removal of water vapour from the air is known as dehumidification of air. The dehumidification of air is achieved if the air is cooled below the dew point temperature of the air. Large scale dehumidification of air can be achieved by passing the air over a cooling coil maintained well below the dew point temperature of the air. The water vapour present in the air condenses over the surface of the cooling coil. The ambient air at point 1(t_1, ω_1, h_1) is passed over the cooling coil which is maintained at the surface temperature of t_4 (point 4). First, the air reaches to dew point temperature (t_2) and finally comes out from the cooling coil at point 3 (t_3, ω_3, h_3). The process is indicated in Fig. 27.5 and the process is represented on psychrometric chart in Fig. 27.6.

Fig

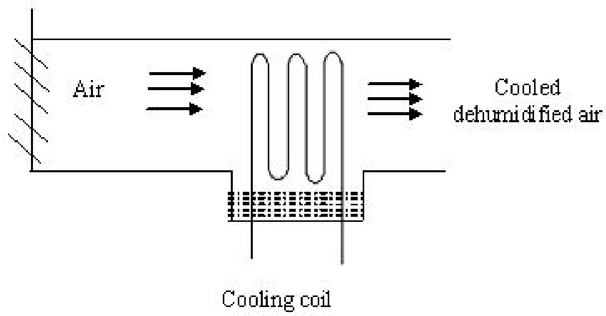

Fig. 27.5 Dehumidification of air by cooling

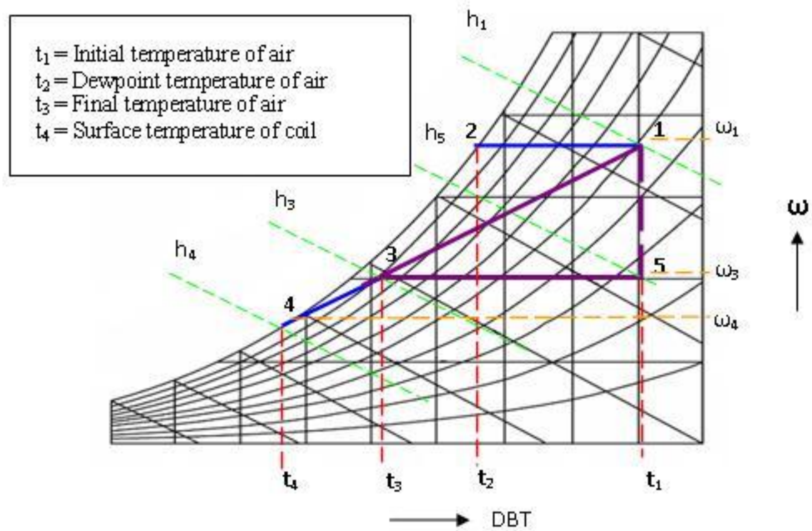

Fig. 27.6: Dehumidification by cooling process on psychrometric chart

As the cooling coil is not hundred per cent efficient, the condition of the air coming out of the coil will be at point 3. The bypass factor of the cooling coil is given by

$$B = \frac{h_3 - h_4}{h_1 - h_4} = \frac{\omega_4 - \omega_3}{\omega_1 - \omega_3} = \frac{t_3 - t_4}{t_1 - t_4}$$

As t_4 is the Apparatus Dew Point temperature (ADP) of the coil

$$\therefore B = \frac{t_3 - ADP}{t_1 - ADP}$$

As the final condition obtained is independent of the path followed by the process so that the process may be assumed to have followed the path 1-5 and 5-3 as shown in the figure.

The total heat removed from the air is given by

$Q_T = h_1 - h_3 = (h_1 - h_5) + (h_5 - h_3) = Q_L + Q_S$

where $(h_1 - h_5)$ is the latent heat removed and $(h_5 - h_3)$ is the sensible heat removed.

The ratio $\frac{Q_S}{Q_T}$ is called sensible heat factor (SHF) or sensible heat ratio (SHR) and it is given as

$$\therefore SHR = \frac{Q_S}{Q_S + Q_T} \quad \text{Where } Q_T = Q_L + Q_S$$

The slope of the line 1-3 on the psychrometric chart is fixed by SHF. Based on the initial condition of the air and SHR, it is possible to draw the sensible heat factor line on the psychrometric chart.

The SHRs for few applications are recommended as follows.

Residence or small office = 0.9

Restaurant or crowded room = 0.8

Auditorium of full capacity = 0.7

Food Processing room (wet) = 0.6

The cooling coil capacity in tons of refrigeration is given by, $\text{Capacity in tons} = \frac{(h_1 - h_3) \times m_a}{3.5}$

Where, m_a is the mass of dry air in kg per second in a mixture, passing over the coil.

4. Adiabatic Chemical Dehumidification of the air:

When humid air is passed through the solid absorbent bed or through the spray of liquid absorbent, part of water vapour will be absorbed and the water vapour content of the air decreases as indicated in Fig. 27.7. The latent heat of the vapour is liberated resulting into increase in dbt of the air without any change in total enthalpy of the air. This method of dehumidification is economical for small size dehumidification of air and the change in humidity required is smaller.

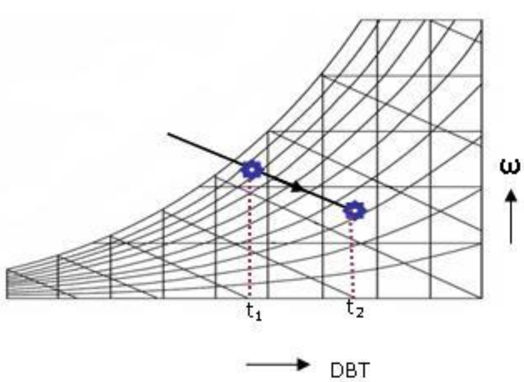

Fig.27.7: Adiabatic dehumidification

5. Adiabatic humidification of air:

Adiabatic humidification of air can be achieved by passing the air through the spray of water. When the air is passing through the spray of water, a part of water will be evaporated by using the sensible heat of the air. During this process, the specific humidity of the air increases and enthalpy of the air does not change. The process of adiabatic humidification is shown in Fig. 27.8. The latent heat required to evaporate the water is taken from the atmosphere of the area resulting into the decrease in the dbt of the air without any change in total enthalpy of the air. This method of humidification is economical for small size humidification of air and the change in humidity required is smaller. The process of adiabatic humidification is represented on psychrometric chart in Fig. 27.9.

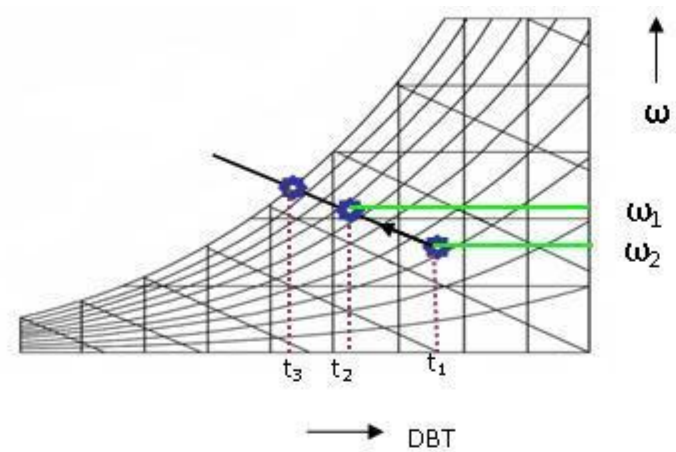

Fig. 27.9 Adiabatic Humidification on psychrometric chart

The efficiency of the humidifier can be calculated as under.

$$\%\,\eta = \frac{\text{Actual drop in dbt}}{\text{Ideal drop in dbt}} \times 100 = \frac{t_1 - t_2}{t_1 - t_3} \times 100$$

6. Mixing of air:

When two air streams having different psychrometric property are mixed together, the quality of resultant air can be obtained by taking heat and moisture balance. The process of mixing of two air streams is indicated in Fig. 27.10 and it is represented on psychrometric chart in Fig. 27.11.

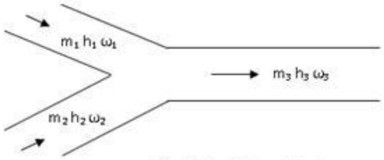

Fig. 27.10: Mixing of air streams

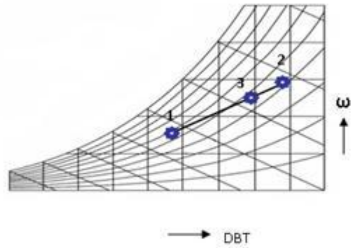

Fig. 27.11: Mixing of air on psychrometric chart

Taking heat balance,

$$m_1h_1+m_2h_2= (m_1+m_2)h_3$$

$$\therefore h_3 = \frac{m_1h_1 + m_2h_2}{m_1+m_2}$$

Taking moisture balance,

$$m_1\omega_1+m_2\,\omega_2 = (m_1+m_2)\,\omega_3$$

$$\therefore \omega_3 = \frac{m_1\omega_1 + m_2\omega_2}{m_1+m_2}$$

By calculating the values of h_3 and ω_3 the quality point can be located on the psychrometric chart and other properties can be read from the psychrometric chart.

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Lesson 28. Humidity Measurement, Dehumidifiers, Humidity Control

28.1. INTRODUCTION

The measurement of dbt and wbt is a basic way to know the psychrometric properties of air by using psychrometric chart. It is necessary to locate the point on sychrometry chart to read values of other parameters such as relative humidity, dew point temperature, specific humidity etc. The dbt and wbt of the air are measured by using following type of psychrometers.

28.2. LABORATORY PSYCHROMETER

This is the simplest type of psychrometer used for the measurement of dbt and wbt of the air. It consists of two thermometers, one measure the dbt of the air and second measures wbt as its bulb is covered with wet cloth as shown in Fig. 28.1 (a). This type of psychrometer is commonly used in laboratories.

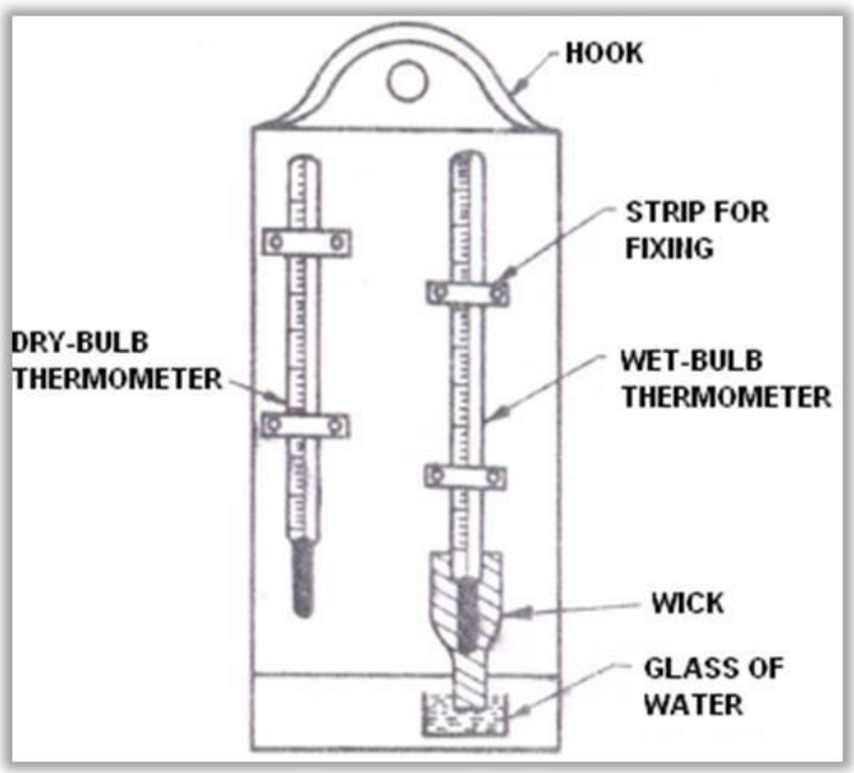

Fig. 28.1(a): Laboratory psychrometer

As the air is passed over a thermometer covered with wet wick, the moisture contained in the wick tends to evaporate. The cooling effect of evaporation lowers the temperature measured as compared to dry bulb thermometer. When the temperature measured by the wet bulb thermometer reaches steady state, then the heat absorbed from the thermometer bulb required for the evaporation of water vapour going into air is equal to the heat given by the air by convection to the thermometer. The evaporation rate of water from the wet wick depends upon the humidity of the air passing over it. The air having low humidity gives more evaporation rate and the drop in temperature measured by the wet bulb thermometer will be more and vice versa. This shows that the wbt is a measure of degree of saturation or relative humidity of the air.

28.2.1. Sling psychrometer

This psychrometer consists of two mercury thermometers mounted on a frame provided with a handle as shown in Fig. 26.2 (a). The psychrometer can be rotated with the helps of a handle to produce necessary air motion. The air velocities of 5 m/s to 10 m/s are recommended to get more accurate value of wbt. The rotating motion of sling psychrometer provides necessary air velocity over the thermometer.

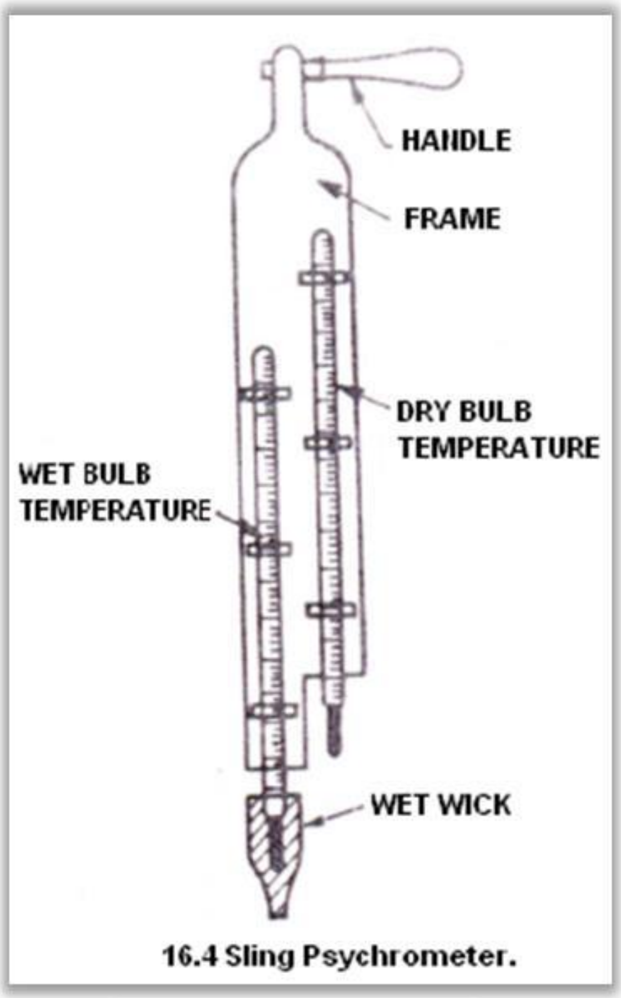

Fig. 28.2 (a): Sling Psychrometer

28.2.2. Aspirating psychrometer

The aspirating psychrometer is shown in **Fig 28.3** which has a small blower at the top for producing rapid motion of air over the thermometer bulbs. A provision is also made in the psychrometer to prevent errors due to radiant heat exchange. This type of psychrometer is used for measuring the dbt and wbt at a particular interval of time mostly for measuring the atmospheric conditions of air.

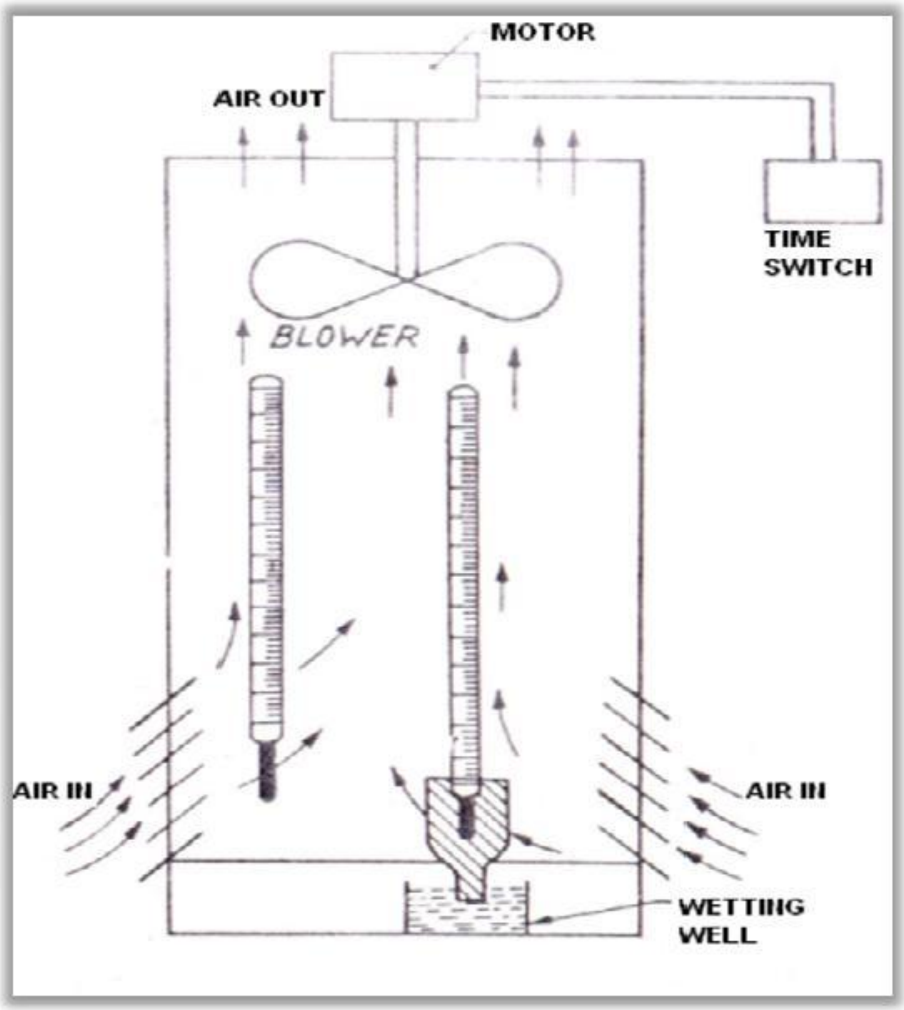

Fig. 28.3: Aspirating psychrometer

28.2.3. Continuous Recording Psychrometer/data logger

Continuous recording of psychrometers are available for measurement of dbt and relative humidity of the air using various types of sensors and transducers. The values of the air quality will be recorded and it is possible to get digital output of data.

28.3. DEHUMIDIFIERS

Dehumidification of air is required to control the RH of the air for comfort air conditioning, storage, food production, packing etc. The process of dehumidification by cooling has been explained in chapter 27. There are three main methods of dehumidification, viz.

a) Reducing the temperature of air below dew point

b) Absorption of moisture from air

c) Adsorption of moisture from air

Reducing the temperature of air below dew point: This may be carried out by two methods- either by passing the air over a cold surface below DPT, or by passing the air though a spray of cold water below DPT

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Lesson 29. Air Conditioning Systems and Numerical

29.1. INTRODUCTION

As discussed earlier, air conditioning refers the control of environmental condition of the air in terms of temperature, humidity, distribution of air and purity of air depending on the use of the air conditioning.

The design of air conditioning system is very difficult task as it involves the knowledge of variation of environmental conditions in different seasons, psychrometric processes, design of air handling system, load calculations, economic considerations etc. One type of system designed for specific place may not be suitable at some another place.

29.2 REQUIREMENT FOR COMFORT AIR CONDITIONING

The following are the important requirements for the comfort feeling of occupant.

29.2.1. Oxygen level: The human body requires about 0.65 m^3 of O_2 per hour under normal conditions and produces 0.2 m^3 of CO_2. The concentration of CO_2 in atmospheric air is nearly 0.6%. The partial pressure of oxygen gets reduced when CO_2 in air exceeds above 2% creates breathing problem. The CO_2 level around 6% creates extreme discomfort and unconsciousness occurs at 10% CO_2 level. Therefore, it is necessary to maintain the level of CO_2 by replacement of air from the air conditioning system.

29.2.2. Heat removal: The human body converts thermal energy into mechanical work with 20% efficiency and remaining amount of heat appears as heat to the atmosphere. Therefore, it is necessary provide sufficient air circulation to carry away the heat dissipated by the occupant to prevent rise in temperature of the space.

29.2.3. Moisture control: The loss of moisture from human body is nearly 50 g per hour under rest condition. The removal of vapour from the air is important to provide comfortable conditions in the air conditioned room. The majority of people feel comfortable at 60-65% relative humidity in the air. As the humidity increases, the rate of evaporation will also decrease. The control of humidity can be achieved by installing dehumidifier in the system.

29.2.4. Air distribution: The velocity of air affects the heat transfer from the body. If air temperature is lower than body temperature, increased velocity improves the comfort feeling but reverse action takes place when air temperature is higher than body temperature. It is recommended that the air velocity should not be more than 6 to 9 m/min at 20 ^{0}C and 9 to 15 m/min at 22^{0}C. Uniform air distribution at recommended air velocity is also very essential for comfort requirement in the air conditioned space.

29.2.5. Quality of air: The air supplied in the air conditioned space should be free from odour, dust, toxic gases, smoke and bacteria to prevent harmful effect on the human health. These can be eliminated by using various types of air purification equipment. Air filters are being used to remove dust, dirt, lint etc. from the air. Supply of quality air in the air conditioned space is the main requirement of centralized air conditioning system. High

Efficient Particulate Air Filters (HEPA), Ultra Low Penetration Air Filters (ULPA), Activated Carbon Filters and Plasma Air Purifiers are employed in air conditioning system of advanced countries.

29.3. DESIGN OF AIR CONDITIONING SYSTEM

Air conditioning system is defined as an assembly of different components (heating, humidifier, dehumidifier, cooling etc) used to produce required condition of air within a required space or building.

The air conditioning systems are mainly classified as (i) Central air conditioning system (ii) Unitary air conditioning system.

The central air conditioning system consists of several components grouped together in one central room and conditioned air is distributed in air conditioned rooms using air carrying duct. The central air conditioning system requires the following components.

(i) Sensible cooling coil

(ii) Cooling & dehumidifying coil

(iii) Heating coil

(iv) Air cleaning equipment

(v) Humidifier

(vi) Blower and motor

(vii) Control devices

A block diagram indicating the principle of air conditioning system is shown in Fig. 29.1.

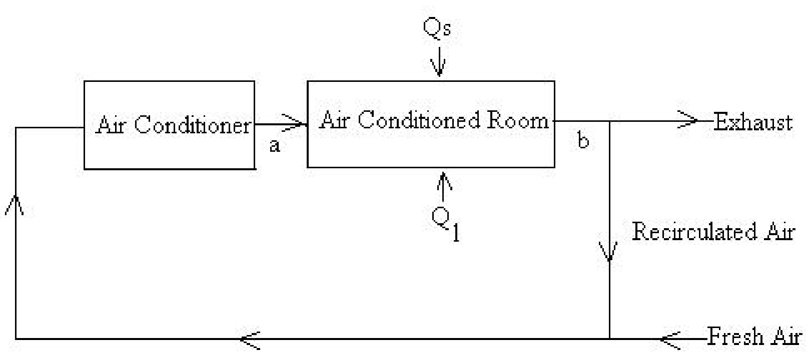

Fig.29.1: Block diagram of air conditioning system

The condition of air passing through the room follows path along the room SHR (or SHF) line depending on the Q_s (Sensible heat gain) and Q_l (latent heat gain). The condition of air entering and leaving the conditioned room is shown as 'a' and 'b' respectively on the psychrometric chart in Fig. 29.2.

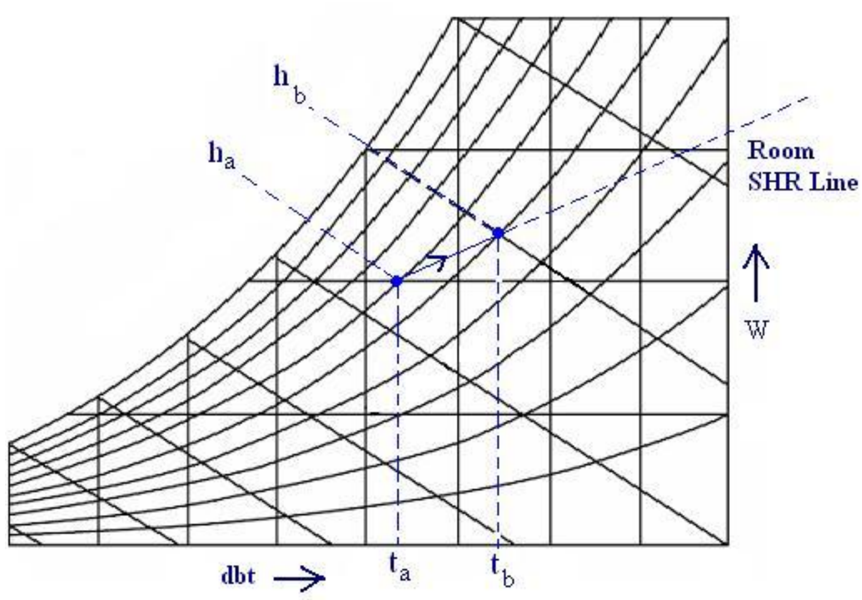

Fig 29.2: SHR of air conditioning system

The system air conditioning system, total heat removed is equal to sensible heat removed and latent heat removed. Thus,

$Q_s+Q_l = m\ (h_b-h_a)$

Where, m= flow rate of air, kg dry air/h

$SHR = Q_s\ /\ (Q_s - Q_l)$

High value of SHR (or SHF) indicates high sensible heat load. The capacity of various components of the system can be estimated based on the initial quality of air, rate of air recirculation, SHF etc. Psychrometric chart gives clear idea for the selection of different components of the air conditioning system after locating initial air quality and required comfort condition.

An air conditioning system with cooling and dehumidifying is shown in the Fig. 29.2 and corresponding psychrometric chart indicating the conditions of the air is shown in Fig. 29.3.

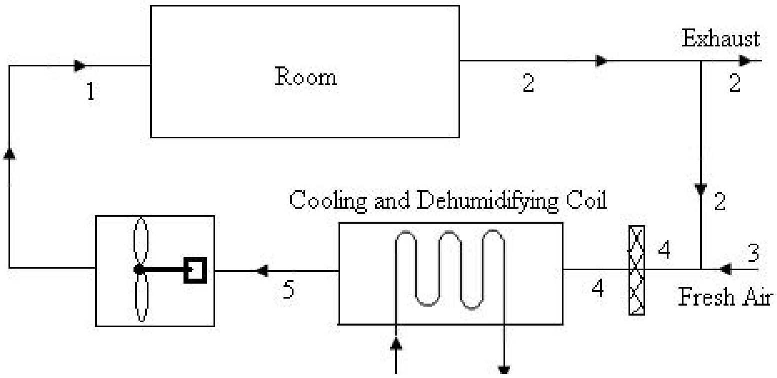

Fig. 29.3: Air conditioning system with mixing air

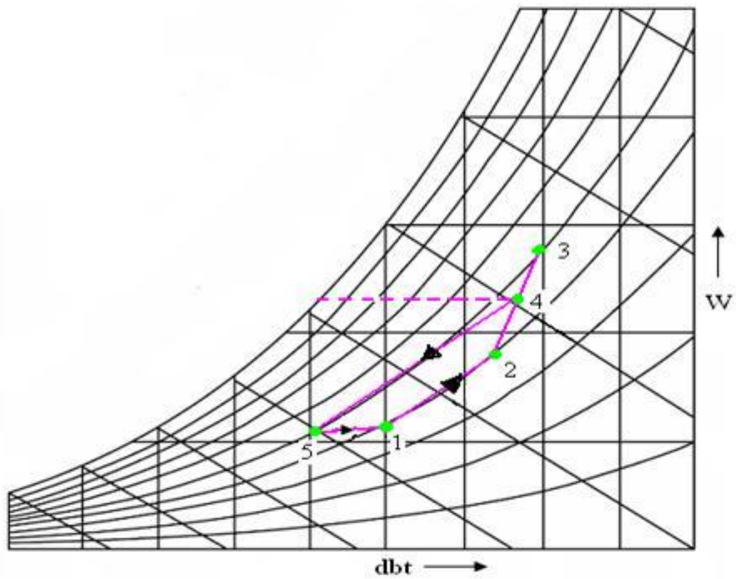

Fig 29.4: Air conditioning system on psychrometric chart

In the system, fresh air is mixed with the exhaust air. The proportion of the mixing depends on the quality of the air streams. The condition 4 is the mixing of air at condition 2 and 3. The condition 5 is the condition of air leaving the cooling and dehumidifying coil and 5-1 represents the heating of air passing through blower due to friction. The process 1-2 represents the condition of air passing through the air conditioned room depending on the RSHF of the process.

The year round air conditioning system consists of cooling, heating, humidifying, dehumidifying components. The operation of these components is regulated to achieve required conform conditions of air.

29.4. NUMERICAL

29.4.1. A psychrometer reads dbt of 40 ° C and wbt of 28 ° C. Calculate the following.

1. Specific humidity

2. Relative humidity

3. Dew point temperature

4. Enthalpy of the air

Solution:

Using the Apjohn equation

$$
\begin{aligned}
Pv &= (Pvs)wbt - \frac{1.8Pt(dbt - wbt)}{2700} \\
&= 0.03778 - \frac{(1.8 \times 1.01325)(40 - 28)}{2700} \\
&= 0.03778 - 0.00811 \\
&= 0.02968 \text{ bar}
\end{aligned}
$$

$$
\begin{aligned}
\text{Specific humidity} &= 0.622 \frac{P_V}{P_t - P_V} \\
&= 0.622 \frac{0.02968}{1.01325 - 0.02968} \\
&= 0.01877 \text{ kg/kg dry air}
\end{aligned}
$$

$$
\begin{aligned}
\text{Relative humidity} &= \frac{P_V}{P_{vs}} \\
&= \frac{0.02968}{0.07375} \\
&= 0.4024 = 40.24\%
\end{aligned}
$$

Dew point temperature is the saturation temperature of the water vapour at the Pv of the air. It is obtained from steam table corresponding to

$P_V = 0.02968$ bar

$\therefore d_{pt} = 24° C$

$$
\begin{aligned}
\text{Enthalpy of air} &= 1.005 \text{ dbt} + \omega(2500 + 1.88\text{dbt}) \\
&= 1.005 \times 40 + 0.01877(2500 + 1.88 \times 40) \\
&= \mathbf{88.54 \text{ kJ/kg dry air}}
\end{aligned}
$$

[Note: Atmospheric pressure = 1.01325 bar, 1 bar= 1.0197 kg/cm^2]

29.4.2. Atmospheric air at 15 ° C dbt and 80% RH is heated to 22 ° C dbt in steam coil heater at the rate of 100 m^3 per minute. Find

1. Heat added per minute

2. RH of the heated air

Solution:

Air at 15°C dbt:

$$\text{Specific humidity, } \emptyset = \frac{P_V}{P_{VS}}$$

$$0.8 = \frac{P_V}{0.01715}$$

$$P_V = 0.01372 \text{ bar}$$

$$\omega = 0.622 \frac{P_v}{P_t - P_v}$$

$$= 0.622 \frac{0.01372}{1.01325 - 0.01372} = \frac{0.008534}{0.9995} = 0.0085383 \text{ kg/kg dry air}$$

$$h_1 = 1.005\text{dbt} + \omega\,(2500 + 1.88\text{ dbt})$$

$$= 1.005 \times 15 + 0.0085383(2500 + 1.88 \times 15)$$

$$= 15.075 + 21.5865$$

$$= \mathbf{36.66\ \ kJ/kg\ dry\,air}$$

Air at 22°C dbt:

As the process is sensible heating, there is no change in specific humidity

Thus $\omega_2 = 0.0085383$ kg/kg dry air

$$\emptyset = \frac{P_v}{P_{vs}}$$

$$= \frac{0.01372}{0.02659} = 0.5160 = 51.60\%$$

$$h_2 = 1.005\text{dbt} + \omega\,(2500 + 1.88\text{ dbt})$$

$$= 1.005 \times 22 + 0.0085383(2500 + 1.88 \times 225)$$

$$= 22.11 + 21.6999 = 43.81\ \text{ kJ/kg day air}$$

Mass flow rate of air, $m_a = \frac{100\,(1.01325 - 0.01372)}{288 \times 287} \times 10^5 = 120.93\ kg\ dry\ air/minute$

$$\therefore Heat\ aded = m_a\,(h_2 - h_1)$$

$$= 120.93(43.81 - 36.66)$$

$$= \mathbf{864.65\ kJ/minute}$$

29.4.3. Ambient air at the rate of 50 m^3/min at 25 ° C dbt and 50% RH is mixed with another air stream having dbt of 40 ° C and wbt of 29 ° C at the rate of 100 m^3/min. Find the dbt and wbt of the mixture (Use psychrometric chart).

Figure:

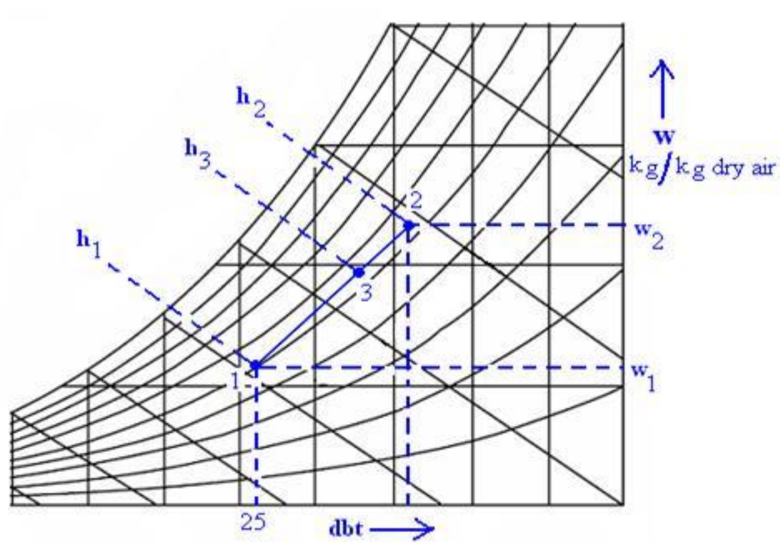

Fig.29.5: Psychrometric chart

Air at 25 °C dbt and 50% RH

$\omega_1 = 0.0100$ kg/kg dry air

$h_1 = 50.5$ kJ/kg dry air

$$m_1 = \frac{50}{0.86} = 58.14 \text{ kg dry air/min}$$

Air at 40 °C dbt and 29 °C wbt

$\omega_2 = 0.021$ kg/kg dry air

$h_2 = 95.0$ kJ/kg dry air

$$m_2 = \frac{100}{0.92} = 108.69 \text{ kg dry air/min}$$

Taking moisture balance,

$$m_1\omega_1 + m_2\omega_2 = (m_1 + m_2)\omega_3$$

$$\therefore (58.14 \times 0.0100) + (108.69 \times 0.021) = (58.14 + 108.69) \times \omega_3$$

$$\therefore 0.5814 + 2.2825 = 166.83\omega_3$$

$$\therefore \omega_3 = 0.01717 \text{ kg/kg dry air}$$

Taking heat balance,

$$m_1 h_1 + m_2 h_2 = (m_1 + m_2)h_3$$

$$\therefore 58.14 \times 50.5 + 108.69 \times 95 = (58.14 + 108.69)h_3$$

$$\therefore 2936.07 + 10325.55 = (166.83)h_3$$

$$\therefore h_3 = 79.49 \text{ kJ/kg dry air}$$

As values of ω_3 and h_3 are known, values of dbt and wbt can be obtained from the psychrometric chart.

Thus, **dbt=35° C**

wbt= 25.5° C

29.4.4. Ambient air at 35 ° C dbt and 24 ° C wbt is heated to 200 ° C in an indirect steam coil

heater. The flow rate of air is 10 m^3/s. Find the capacity of heating oil in kW (use psychrometric chart).

Solution:

Air at 35° C dbt and 24° C wbt

$h_1 = 72.0$ kJ/kg dry air

$\omega_1 = 0.0144$ kg/kg dry air

Air at 200° C dbt

As the air is heated in an indirect coil heater (sensible heating), there is no change in the value of specific humidity.

Hence, $\omega_2 = 0.0144$ kg/kg dry air

$$h_2 = 1.005\text{dbt} + \omega(2500 + 1.88\text{dbt})$$

$$= 1.005 \times 200 + 0.0144(2500 + 1.88 \times 200)$$

$$= 201.0 + 41.41$$

$$= 242.41 \text{ kJ/kg dry air}$$

Mass flow rate of air, $m = \frac{10}{0.89}$

$$= 11.24 \text{ kg/s}$$

Heater capacity, $kW = m(h_2 - h_1)$

$$= 11.24(242.41 - 72.0)$$

$$= \mathbf{1915.40\ kW}$$

29.4.5. Ambient air at dbt of 35 ° C and wbt of 24 ° C is passed through an adiabatic humidifier having efficiency of 90%. Find the dbt and specific humidity of resultant air.

Solution:

From psychrometric chart, air at 35 ° C dbt and wbt= 24 ° C

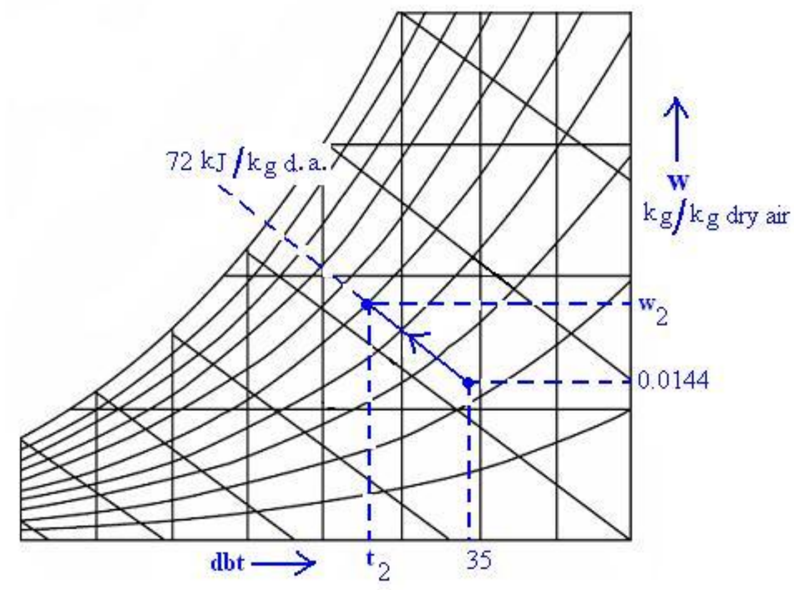

Fig. 29.6: Psychrometric chart

$\omega_1 = 0.0144$ kg/kg dry air

h_1=72.0 kJ/kg dry air

$$\text{Efficiency of humidifier} = \frac{t_1 - t_2}{t_1 - \text{wbt}}$$

$$\therefore 0.9 = \frac{35 - t_2}{35 - 24}$$

$$\therefore 9.9 = 35 - t_2$$

$$\therefore \mathbf{t_2 = 25.1° C}$$

The value of ω_2 from psychrometric chart is 0.0184 kg/kg dry air.

29.4.6. A class room is to be air-conditioned for 60 students when outdoor conditions are 30 ° C dbt and 75% RH. The quantity of air supply is 0.5 m³/min/person. Find the following.

a) Capacity of the cooling coil in TR

b) Capacity of heating Coil in kW

c) Amount of water removed by dehumidifier

d) By-pass factor of the heating coil, if the surface temperature of the heating coil is 25 ° C.

The required comfort conditions are 20 ° C dbt and 60% RH. The air is conditioned first by cooling & dehumidifying and then by sensible heating.

Solution:

The points 1 and 2 are located on the psychrometric chart as their conditions are known as shown in figure below.

The process 1-2-3 represents sensible cooling and dehumidifying and process 3-2 represents sensible heating. Point 3 is located by drawing horizontal line through point 2 which cuts the saturation line at point 3.

From the psychrometric chart,

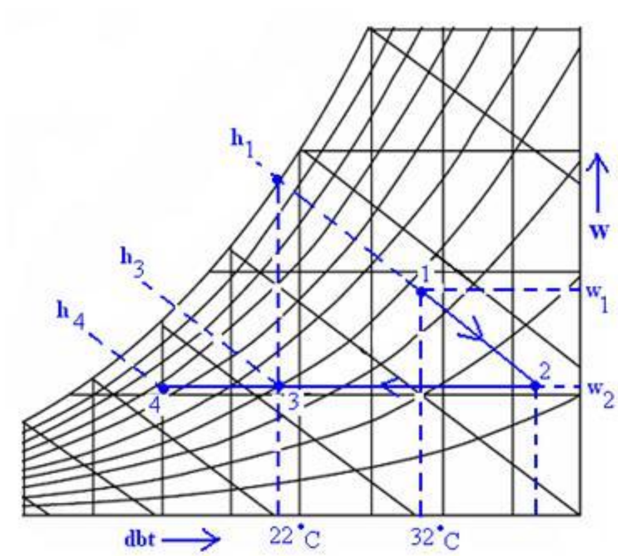

Fig. 29.7: Psychrometric chart

From the psychrometric chart,

$h_1 = 82.0$ kJ/kg dry air

$h_2 = 42.5$ kJ/kg dry air

$h_3 = 34.0$ kJ/kg dry air

$\omega_1 = 0.02$ kg/kg dry air

$\omega_2 = 0.088$ kg/kg dry air

$$\text{Mass of air supplied} = \frac{0.5 \times 60}{0.885} = 33.90 \text{ kg/min.}$$

$$\text{Capacity of cooling coil} = \frac{33 \times (h_1 - h_3)}{60 \times 3.5}$$

$$= \frac{33 \times (82.0 - 34.0)}{60 \times 3.5} = 7.5 \text{ TR}$$

$$\text{Capacity of heating coil, kW} = \frac{33 \times (h_2 - h_3)}{60}$$

$$= \frac{33 \times (42.5 - 34.0)}{60} = 4.67 \ kW$$

$$\text{Amount of water removed (per hour)} = 33 \times (\omega_1 - \omega_2) \times 60$$

$$= 33 \times (0.02 - 0.0088) \times 60$$

$$= 22.18 \text{ kg/h}$$

$$\text{By pass factor} = \frac{25 - 20}{25 - 12} = \frac{5}{13} = \mathbf{0.385}$$

29.4.7. A class room of 60 seating capacity is to be air conditioned. The out door conditions are 32° C dbt and 22° C dbt. The required comfort conditions are 22° C dbt and 55% RH. The quality of air required is 0.5 m^3/min/student. The required conditions are achieved first by chemical dehumidifying and then by sensible cooling of the air. Find

dbt of the air leaving chemical dehumidifier

Capacity of dehumidifier

Capacity of cooling coil in ton.

4. Surface temperature of the cooling coil, if the by pass factor of the cooling coil is 0.3.

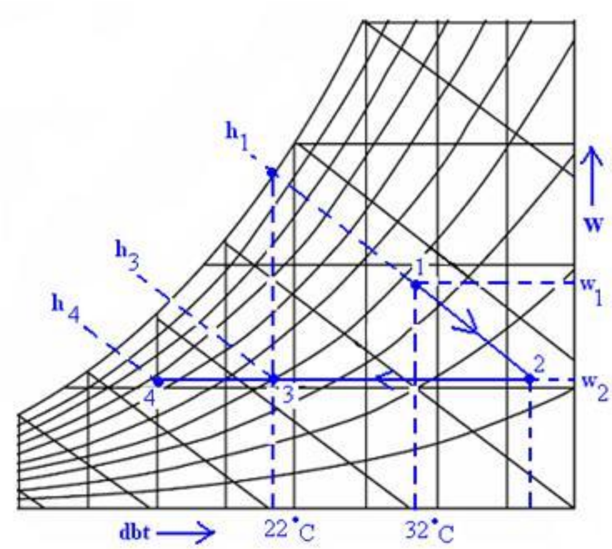

Fig. 29.7: Psychrometric chart

Solution

From psychrometric chart,

$h_1 = 65.0$ kJ/kg d. a.

$h_3 = 43.5$ kJ/kg d. a.

$\omega_1 = 0.0126$ kg/kg d. a.

$\omega_2 = 0.0084$ kg/kg d. a.

$$\text{Mass of air supplied} = \frac{0.5 \times 60}{0.882} = 34.0 \text{ kg/min}$$

$$\text{Capacity of cooling coil} = \frac{34.0 \times (h_1 - h_3)}{60 \times 3.5}$$

$$= \frac{34.0 \times (65.0 - 43.5)}{60 \times 3.5} = 3.48 \text{ TR}$$

$$\text{Capacity of dehumidifier} = 34.0 \times (\omega_1 - \omega_2) \times 60$$

$$= 34.0 \times (0.0126 - 0.0084) \times 60$$

$$= 8.57 \text{ kg/h}$$

The condition of ait at point 2 is 41.5° C dbt

$$\text{By pass factor of cooling coil} = \frac{t_3 - t_4}{t_2 - t_4}$$

$$0.3 = \frac{22 - t_4}{41.5 - t_4}$$

$$\therefore \mathbf{t_4 = 13.64°\,C}$$

Lesson 30. Cooling Load Calculations

30.1. INTRODUCTION

It is necessary to select the capacity of refrigeration plant for the cold storage to maintain required storage temperature in the cold storage. Under capacity plant, may load to higher temperature of cold storage than required while over capacity may lead to higher initial cost of the refrigeration system. The capacity of the refrigeration plant should be such that it can take care of all the heat load of the cold storage. It is also necessary to reduce cold storage load in order to reduce the energy cost for the operation of refrigeration plant. The various factors contributing the total load of the cold storage are discussed below.

30.2. WALL GAIN LOAD

The heat flow rate by conduction through the walls ceiling & floor of the cold storage from outside to inside is called wall gain load.

The value of U depends on the materials used in construction and insulation used in the construction of wall as well as on the thickness of these materials. If either U or 12âˆ†T'> are different for different walls, then it is necessary to calculate Q_w of each wall/ceiling/floor separately taking corresponding values of U and 12âˆ†T'> .

The overall heat transfer co-efficient is given by

Where, h_o = Convection heat transfer Co-efficient on the outer surface

h_i = Convection heat transfer Co-efficient on the inner surface

x_1, x_2.....= thickness of different layers of wall including insulation

k_1,k_2,....= conductivities of different layers of wall including insulation

30.3. AIR CHANGE LOAD

This is the amount of heat carried by the air when cold storage door is opened and part of cold air is replaced by outside warmer air. The air change load depends on the number of air changes occurring in the cold storage, enthalpy of outside air and inside air. The measurement of amount of air changed due to door opening is difficult and hence air change factor is used to estimate the amount of air changed.

Air change load, $Q_a = m\ (h_o-h_i)$

Where, m= mass of air entering, kg d. a. /h

h_o = Enthalpy of outside air, kJ/kg dry air

h_i = Enthalpy of inside air, kJ/kg dry air

Mass of air can be estimated by multiplying volume of cold storage with air change factor. The volume of the air is converted into amount of dry air in the volume taking specific volume of the outside air.

Air change factors (air changes/h) for different size of cold storage are given in table 30.1.

30.4. PRODUCT LOAD

It is necessary to cool the product from initial temperature to the storage temperature. The amount of heat to be removed from the product to lower the temperature of the product from initial temperature to storage temperature is called product load. It is also necessary to estimate the heat load for cooling of the packaging material along with the product as specific heat of product and material is different. For example, plastic crates are to be cooled from room temperature to the storage temperature.

Product load, $Q_p = m_p \times C_1 \times (t_1\text{-}t_2)$

Where m_p = Mass flow rate of the product in the cold storage, kg/h

C_1 = Specific heat of the product kJ/kg K

T_1 = Initial temperature of the cold storage

T_2 = Final storage temperature of the product.

Similarly, heat load of packaging materials transferred in the cold store along with the product is estimated as above taking the mass of packaging material, its specific heat and temperature difference. This load is added in the actual product load.

For frozen foods

$Q_p = m_p \text{ x } C_1 (t_1 + t_f) + m_p h_{fg} + m_p \text{ x } C_2 (t_f - t_2)$

Where t_f = Freezing temperature

h_{fg} = Latent heat of freezing

Heat produced due to respiration of the fruits and vegetables are required to be considered for such types of cold storages.

$Q_r = m_p$ (kg/h) x Respirate rate (kJ/kg)

30.5. MISCELLANEOUS LOAD

The miscellaneous load consists of primarily of heat given off by light and electric motors present in the cold storage.

Cooling load for electric appliances in terms of kJ is given by

Q_c = kW × 3600 kJ/h

Heat Load from occupants is calculated based on the data available for heat loss from human body. It is necessary to refer standard data if heat loss from human body under different temperature conditions. For example, a person at rest at 20 °C, total heat loss from the body is about 400 kJ/h (Q_l= 160 kJ/h and Qs = 240 kJ/h)

Table 30.1: Air changes per hour for cold storage due to infiltration & door openings.

Volume of Cold Storage, m²	Air Changes/h (Air Change Factor)
10	1.23
20	1.95
30	0.65
40	0.57
50	0.50
60	0.45
70	0.42
80	0.37
100	0.35
150	0.27
200	0.23
250	0.21
500	0.14
1000	0.10

Total heat load:

$Q_t = Q_w + Qa + Q_p + Q_m$

It is common practice to add 10-15% of total load as safety factor. After adding safety factor, the cooling load is multiplied by 24 hours and divided by the desired operating time in hours to find capacity of the plant required for the cold storage.

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Lesson 31. Cold Storage Design: Types of Cold Storage and Types of Loads

31.1. INTRODUCTION

The basic purpose of cold storage is to store the perishable food products at optimum temperature to enhance the self life of the products. In dairy plants, cold storages are required for storage of milk, butter, cheese, ice-cream etc. The condition of storage in these cold storages is different depending upon the nature of the product. For example, ice-cream is stored at - 25 °C while milk is stored at 3-4 °C. Similarly, many fruits and vegetables are also stored in cold storages.

31.2. TYPES OF COLD STORAGES

Cold storages are classified in different ways as indicated below.

31.2.1. Classification based on the use of cold store .

Ø Milk cold storage

Ø Cheese cold storage

Ø Butter cold storage

Ø Potato cold storage etc.

The storage conditions to be maintained as well as method of storage for these cold storages vary depending on the optimum storage conditions required for different products. For example, cheddar cheese is stored at around 10 °C and 90 % relative humidity for ripening of cheese. Appropriate method of storage of product is very important aspect. Racks are required to keep cheese blocks in the cold storages.

31.2.2. Classification based on operating temperature of cold storage

Ø Cold storage maintained above 0 °C

Ø Cold storage maintained below 0 °C

Milk cold storage is maintained above °C while ice-cream cold storage is maintained below 0 °C. Product load is one of the factors for estimation of cold storage load. It is necessary to calculate heat to be removed from the product when a part of water gets frozen at storage temperature of the product. The design of evaporator, air circulation, expansion valve etc. will be different in these cold storages. The thickness of insulation required for low temperature cold storage will be more to reduce the wall gain load.

31.2.3. Classification based on the construction

Ø Constructed cold storage

Ø Walk in cold storage

Mostly cold storage is constructed in dairy building as per the design and layout of the dairy plant. The cold storage is generally constructed by civil work and insulated either by Thermocol sheets or PUF panels.

31.3. TYPES OF LOADS IN COLD STORAGES

It is basic requirement to know the types of loads in the specific cold storage in order to find the capacity of the refrigeration system for the cold storage. It is necessary not only to cool the product to the storage temperature but also to meet the cooling load due to various heat infiltrations taking place in the cold storage. Broadly, the total load is divided into two categories as under.

31.3.1. Sensible heat load

► Heat flow through walls, ceilings, floor, doors (structural heat gain).

► Heat gain from infiltration of air due to door openings and movement of products through opening provided in the walls. For example, crates of milk enter in the milk cold storage through a gap provided in the wall using conveyer. This load is kept minimum by using appropriate strips of flexible plastic sheets to reduce the exchange of air.

► Heat received by workers working in cold storage. Though, it is very small as number of persons working in the cold storage is very few. This load is very important in air conditioning system as it is for providing comfort to large number of occupant.

► Heat load due to lighting and other motors used in the cold storage.

31.3.2. Latent heat load

► Latent heat load from infiltration of air.

► Latent heat load from occupancy.

► Latent heat generated from the stored products.

Based on the above heat load, the actual amount of heat flow rate is calculated in order to find total load to decide the capacity of evaporator of the refrigeration plant. The method of load calculation is described in lesson 32.

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Lesson 32.

Construction of Cold Storage, Insulating Materials and Vapour Barriers

32.1. INTRODUCTION

Cold storage is a basic requirement for the storage of perishable dairy and food products. For example milk is stored at around 4°C in cold storage while ice cream is stored at -30 °C . The design of cold storage requires information on the following aspects.

(i) Size of the cold storage

(ii) Products to be stored

(iii) Incoming temperature of the product

(iv) Storage temperature

(v) Ambient temperature

(vi) Air change load

(vii) Numbers of persons working in the cold storage.

32.2. LOCATION OF COLD STORAGE

The location of cold storage is important in terms of ease of product movement as well as operating and construction cost of the cold storage. The cold storage room preferably is located on the cold side of the plant. In case of more cold storages, all the cold storages should be located side by side to reduce the cost of insulation in common wall of adjacent cold stores. It should be located in such a way that finished product can be transferred to the cold stores easily and finished products can be dispatched conveniently. In cash of milk cold storage, conveyers are used to transfer the milk crates directly from packaging machine.

32.3. SIZE OF THE COLD STORE

The size of the cold storage is estimated based on the capacity requirement for the storage of product. The method of storage, working space, air circulations etc. are considered to decide the dimensions of the cold storage. Considering number of milk pouch crates which can be stacked, working space etc; the capacity of storage per m^2 area of the cold room is worked out. Similarly for potato, apples, butter etc., the dimensions of the cold storage can be calculated e.g. 600 kg butter/m^2 area, 500 lit ice-cream/m^2 etc. Storage period is one of the basic requirements to decide the size of cold storage to store the product. The capacity and number of of cold storages required for storage of cheddar cheese will be more as the cheese is stored in the cold storage for 4-5 months for ripening. This may not be the requirement for milk cold storage as it is dispatched twice a day. In addition to exact space required for storage of product, 30 to40 % space is kept for moment.

32.4. CONSTRUCTION OF COLD STORAGES

The basic construction of cold storage is just similar to other rooms except the requirement of insulation for the cold storages. The room is constructed by using masonry work and it is plastered with at least 25 mm thick plaster material (mortar). After curing of the plaster insulation of wall, ceilings & floor is carried out to make it cold storage. Thermocol or expanded polystyrene, cork etc. were widely used for insulation. Presently PUF panels are available to insulate the cold storage. The material of application of insulation varies depending on the type of insulating material and the thickness of insulation required. It is recommended to use PUF panels having stainless steel as the panel material to get long life of the insulation. Holes prepared in PUF panels for inserting support for evaporator, cables, pipes etc. should be sealed perfectly to prevent water vapour inside the insulation.

32.4.1. Insulating Materials

The materials having extremely low thermal conductivities are called insulating materials. It is necessary to insulate the cold storages to prevent the entry of heat through the walls, ceilings and floor of the cold storage when ambient air temperature is higher than the cold storage temperature. Insulation of cold storages is important to reduce the operating cost of the refrigeration plant by reducing heat gain through structure of the cold storage. Insulation is also necessary on suction pipe line of the refrigeration plant in order to reduce the super-heating of suction gas. Chilled water pipelines are also insulated to prevent surface condensation on the pipeline.

32.4.1.1. Describe properties of insulating materials

The desirable properties of insulating materials are listed below.

(i) Low thermal conductivity

(ii) Higher structural strength

(iii) Light in weight

(iv) High water repellent property

(v) Odorless

(vi) Non-inflammable

(vii) Low cost

32.4.2. Vapour barriers:

The vapour barriers are the materials which are placed on the hot side of the cold storage to prevent moisture migration and to protect the insulation from moisture condensation. Various types of vapour barriers such as structural sheet of Aluminum and S.S., thin aluminum foils, plastic film hot melt type bitumen, special type of paints etc. are used to prevent moisture transfer through the insulating material. Bitumen and aluminum foil are widely used in insulation as permeance is very low. Vapour penetration into the insulation will occur as vapour pressures are lower at lower temperature and warm air will condense which in term will form ice which may damage the panels. Panel and electrical services are carefully designed to ensure long term vapour sealing. Penetrations are required for evaporator supports, electrical wiring and refrigeration pipes. In such cases, make a hole in the panel and use PVC sleeve for the required penetration and sealing materials such as silicon may be used to make it air tight.

REFERENCES

Arora, S. C. and Domkundwar, S. 1989. A Course in Refrigeration and air conditioning. 5th ed. Dhanpat Rai and Sons, Delhi.

Arora, C. P. 2000. Refrigeration and air conditioning. Tata McGraw-Hill, New Delhi.

Ballaney, P. L. 1992. Refrigeration and air conditioning. Khanna Publ., New Delhi.

Prashad, M. 2007. Refrigeration and air conditioning. New Age International, New Delhi.

Jorden, R. C. and Priester, G. B. 1957. Refrigeration and air conditioning. Prentice-Hall, New Delhi.