University of Anbar

College of Engineering

Mechanical Engineering Department

Subject: Air Conditioning

Class: Fourth Year

Course Tutor: Assist.Prof.Dr.Obaid T.Fadhil

Air Conditioning Engineering

ME 4302-Air Conditioning (3-2-1-2)

Course Definition:

Air conditioning part of the curriculum of Mechanical Engineering Program. This course covers the key aspects of air conditioning, including the calculation of the moist air properties, the use of the psychrometric chart, the estimation of the heating and cooling loads, as well as the design of air ducts.

Course Topics:

1. Introduction to air conditioning. 2. Moist air properties. 3. Psychrometric chart and psychrometry processes. 4. Thermal comfort. 5. Indoor and outdoor design conditions. 6. Heating load calculation. 7. Cooling load calculation. 8. Air conditioning systems. 9. Air distribution systems and duct design.

Course Description:

History of air conditioning, units and dimensions, review of basic principles, vapour pressure, moisture content, relative humidity, dry and wet bulb temperatures, specific volume, dew point, enthalpy, psychrometric chart, mixtures, sensible heating and cooling, dehumidification, humidification, cooling & dehumidification with reheat, heat balance equation, comfort charts, prediction of thermal comfort, indoor & outdoor design conditions, overall heat transfer coefficient, heat loss in building, ventilation heat loss, infiltration heat loss, air required for heating space, heat gain from external & internal sources, space cooling load, cooling coil load, cooling load calculation, functions of air conditioning systems, unitary systems, central-station systems, system selection & applications, economic evaluation, basic principles of air flow in ducts, duct sizing, duct design methods, supply air outlets and air distribution patterns.

Chapter One

Introduction

1.1 Brief History of Air Conditioning

The need for air-conditioning was first realized when it was found that certain products could be produced better in the right environment. With air-conditioning, there was no dependence of the product quality on the uncertainties of the weather and the factory sites were not limited to areas with a suitable climate. First to realize the importance of treatment of air was the textile industry. Industrial air conditioning later spread to various other areas such as manufacture of rayon and plastics, colour printing, pharmaceuticals, tobacco industry, manufacture, development, reproduction of photographic materials, electrical equipment, and many hygroscopic products.

Although sporadic attempts have been recorded for improving the ventilation and thermal conditions of indoor spaces, the first use of air-conditioning by mechanical means for human comfort was introduced in the United States of America at the turn of the twentieth century. There were few comfort cooling installations before 1920, but in the mid-twenties theatres all across the country were air conditioned to draw more customers. This was followed by the use of air conditioning in other commercial premises such as restaurants, night clubs, office blocks, etc.

In the 1930s, air conditioning was introduced in transportation systems such as trains, airplanes, buses and trolleys. After the Second World War, with the rising family income there was a boom in domestic air conditioning. This trend of use in the United States was subsequently repeated in the developing countries as more and more countries achieved higher standards of living.

1.2 System International Unites (SI Units)

SI or the International System of Units is the purest form and an extension and refinement of the traditional metric system. There are six basic SI units as given in Table 1.1. The units of other thermodynamic quantities may be derived from these basic units. The unit of temperature is kelvin which measures the absolute temperature given by

$$T = t + 273.15$$

where **t** is the Celsius temperature in ^{o}C.

Table 1.1 Basic SI units

Quantity	Unit	Symbol
Length	meter	m
Mass	kilogram	kg
Time	second	s
Temperature	kelvin	K
Electrical current	ampere	A
Luminous intensity	candle	cd

1.2.1 Unit of Force

Force **F** is proportional to mass **m** and acceleration **a**, so that

$$F = C(m)(a)$$

Where C is a proportionality constant, and it's value is taken as equal to the standard gravitational acceleration as (9.80665 m/s^2). The SI unit of force is Newton denoted by the symbol **N**.

$$1\ N = 1\ kg.m/s^2$$

1.2.2 Units of Pressure

The SI unit of pressure **P** can also be derived from its definition as force per unit area

$$P = F / A \quad (N/m^2)$$

The unit is also called **Pascal** and is denoted by the symbol **Pa**.

Another common SI unit of pressure is **bar** which is equivalent to a pressure of 10^5 N/m^2 or 0.1 MN/m^2 or 100 kN/m^2.

One standard atmosphere is given by

1 atm = 1.01325 bar = 760 mm Hg = 760 torr

1 torr = 1 mm Hg = 1/760 atm = 133 N/m^2

1.2.3 Unit of Energy (Work and Heat)

The SI unit of work is Newton meter (Nm) or Joule (J). Thus

$$1\ Nm = 1\ J = 1\ kg\ m^2/s^2$$

Since both heat and work are energy, the SI unit of heat is the same as the unite of work (J).

1.2.4 Units of Power

The SI unit of power is Watt (W). It is defined as the rate of doing 1 Nm of work per second. Thus 1 W = 1 J/s = 1 Nm/s

It may also be noted that Watt also represents the electrical unit of work defined by

1 W = 1(Volt) x 1(ampere) = 1 J/s

Further, the units of energy can be derived from those of power. Thus

1 J = 1 W.s

1 kWh = 3600000 J = 3600 kJ = 860 kcal = 3410 Btu

1.2.5 Units of Enthalpy

The SI unit of specific enthalpy is J/kg or kJ/kg

1 kJ/kg = 0.239 kcal/kg = 0.42 Btu/lb

1.2.6 Units of Entropy and Specific Heat

These are expressed as

1 kJ/kg.K = 0.239 kcal/kg ^{o}C = 0.239 Btu/lb ^{o}F

1.2.7 Units of Refrigerating Capacity

The standard unit of refrigeration in vogue is ton refrigeration or simply (TR). It is equivalent to the production of cold at the rate at which heat is to be removed from one US tone of water at 32°F to freeze it to ice at 32°F in one day or 24 hours. Thus

1 TR = (1 x 2000 lb x 144 Btu/lb)/24 hr = 12000 Btu/hr = 200 Btu/min

Also, 1 Btu = 1.055 kJ

1 TR = 12000 x 1.055 = 12660 kJ/hr = 211 kJ/min = 3.5167 kW

Chapter Two

Moist Air Properties

In 1911 Willis H. Carrier made a significant contribution to the air-conditioning filed when he published relations for moist air properties together with psychrometric chart. These formulas became fundamental to the industry.

In about 1945 Goff and Gratch published thermodynamic properties for moist air that were for many year the most accurate available. New formulations have recently been developed at the National Bureau of Standards. The properties based on these formulations are the basis for the thermodynamic properties of moist air given in the ASHRAE Handbook, Fundamentals.

Moist Air and The Standard Atmosphere

Atmospheric air is a mixture of many gases plus water vapor and countless pollutants. A side from the pollutants, which may vary considerably from place to place, the composition of the dry air alone, is relatively constant, varying slightly with time, location, and altitude. The ASHRAE Handbook, Fundamentals gives the following approximate composition of dry air by volume

Nitrogen	0.78084
Oxygen	0.20948
Argon	0.00934
Carbon dioxide	0.00031
Neon, helium, sulfur dioxide, hydrogen, and other minor gases	0.00003

Based on the composition of air in Table 2.1, the molecular weight (Ma) of dry air is:

$$Ma = Mo_2 \times Fo_2 + M_N \times F_N + M_A \times F_A + M_{Co} \times F_{Co} \quad \ldots\ldots\ldots\ldots(2.1)$$

$$= 32 \times 0.209 + 28.016 \times 0.7809 + 39.94 \times 0.0093 + 44.014 \times 0.0003$$

$$= 28.965$$

Table 2.1

Constituent	Molecular Weight	Volume Fraction
Oxygen	32.000	0.2095
Nitrogen	28.016	0.7809
Argon	39.944	0.0093
Carbon dioxide	44.014	0.0003

And the gas constant of the air (Ra) can be calculate from equation (2.2)

$$Ra = \frac{R}{Ma} = \frac{8314}{28.965} = 287 \text{ J/kg.K} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.2)$$

Where:

R: Universal gas constant

Ra: gas (air) constant

The molecular weight of water is 18.015 and the gas constant for water vapor is:

$$Rv = \frac{8314}{18.015} = 461 \text{ J/kg.K}$$

The universal gas low (perfect gas) is:

$$PV = m\,R\,T \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.3)$$

$$P = \rho\,R\,T$$

Standard sea level density compute using Eq. (2.3) with the standard temperature and pressure is 1.225 kg/m^3 of dry air.

Atmospheric pressure may be computed as a function of elevation by the following relation:

$$P = a + b\,H \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.4)$$

Where the constant **a** and **b** are:

a = 101.325 , b = - 0.01153 when $H \leq 1220$ m

a = 29.42 , b = - 0.0009 when $H > 1200$ m

H : elevation above sea level in meter.

TABLE A-2 Properties of Saturated Water (Liquid–Vapor): Temperature Table

H_2O

Temp. °C	Press. bar	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Specific Volume m³/kg Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Internal Energy kJ/kg Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Enthalpy kJ/kg Evap. h_{fg}	Enthalpy kJ/kg Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Entropy kJ/kg · K Sat. Vapor s_g	Temp. °C
.01	0.00611	1.0002	206.136	0.00	2375.3	0.01	2501.3	2501.4	0.0000	9.1562	.01
4	0.00813	1.0001	157.232	16.77	2380.9	16.78	2491.9	2508.7	0.0610	9.0514	4
5	0.00872	1.0001	147.120	20.97	2382.3	20.98	2489.6	2510.6	0.0761	9.0257	5
6	0.00935	1.0001	137.734	25.19	2383.6	25.20	2487.2	2512.4	0.0912	9.0003	6
8	0.01072	1.0002	120.917	33.59	2386.4	33.60	2482.5	2516.1	0.1212	8.9501	8
10	0.01228	1.0004	106.379	42.00	2389.2	42.01	2477.7	2519.8	0.1510	8.9008	10
11	0.01312	1.0004	99.857	46.20	2390.5	46.20	2475.4	2521.6	0.1658	8.8765	11
12	0.01402	1.0005	93.784	50.41	2391.9	50.41	2473.0	2523.4	0.1806	8.8524	12
13	0.01497	1.0007	88.124	54.60	2393.3	54.60	2470.7	2525.3	0.1953	8.8285	13
14	0.01598	1.0008	82.848	58.79	2394.7	58.80	2468.3	2527.1	0.2099	8.8048	14
15	0.01705	1.0009	77.926	62.99	2396.1	62.99	2465.9	2528.9	0.2245	8.7814	15
16	0.01818	1.0011	73.333	67.18	2397.4	67.19	2463.6	2530.8	0.2390	8.7582	16
17	0.01938	1.0012	69.044	71.38	2398.8	71.38	2461.2	2532.6	0.2535	8.7351	17
18	0.02064	1.0014	65.038	75.57	2400.2	75.58	2458.8	2534.4	0.2679	8.7123	18
19	0.02198	1.0016	61.293	79.76	2401.6	79.77	2456.5	2536.2	0.2823	8.6897	19
20	0.02339	1.0018	57.791	83.95	2402.9	83.96	2454.1	2538.1	0.2966	8.6672	20
21	0.02487	1.0020	54.514	88.14	2404.3	88.14	2451.8	2539.9	0.3109	8.6450	21
22	0.02645	1.0022	51.447	92.32	2405.7	92.33	2449.4	2541.7	0.3251	8.6229	22
23	0.02810	1.0024	48.574	96.51	2407.0	96.52	2447.0	2543.5	0.3393	8.6011	23
24	0.02985	1.0027	45.883	100.70	2408.4	100.70	2444.7	2545.4	0.3534	8.5794	24
25	0.03169	1.0029	43.360	104.88	2409.8	104.89	2442.3	2547.2	0.3674	8.5580	25
26	0.03363	1.0032	40.994	109.06	2411.1	109.07	2439.9	2549.0	0.3814	8.5367	26
27	0.03567	1.0035	38.774	113.25	2412.5	113.25	2437.6	2550.8	0.3954	8.5156	27
28	0.03782	1.0037	36.690	117.42	2413.9	117.43	2435.2	2552.6	0.4093	8.4946	28
29	0.04008	1.0040	34.733	121.60	2415.2	121.61	2432.8	2554.5	0.4231	8.4739	29
30	0.04246	1.0043	32.894	125.78	2416.6	125.79	2430.5	2556.3	0.4369	8.4533	30
31	0.04496	1.0046	31.165	129.96	2418.0	129.97	2428.1	2558.1	0.4507	8.4329	31
32	0.04759	1.0050	29.540	134.14	2419.3	134.15	2425.7	2559.9	0.4644	8.4127	32
33	0.05034	1.0053	28.011	138.32	2420.7	138.33	2423.4	2561.7	0.4781	8.3927	33
34	0.05324	1.0056	26.571	142.50	2422.0	142.50	2421.0	2563.5	0.4917	8.3728	34
35	0.05628	1.0060	25.216	146.67	2423.4	146.68	2418.6	2565.3	0.5053	8.3531	35
36	0.05947	1.0063	23.940	150.85	2424.7	150.86	2416.2	2567.1	0.5188	8.3336	36
38	0.06632	1.0071	21.602	159.20	2427.4	159.21	2411.5	2570.7	0.5458	8.2950	38
40	0.07384	1.0078	19.523	167.56	2430.1	167.57	2406.7	2574.3	0.5725	8.2570	40
45	0.09593	1.0099	15.258	188.44	2436.8	188.45	2394.8	2583.2	0.6387	8.1648	45

TABLE A-2 (Continued)

Temp. °C	Press. bar	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Sat. Vapor s_g	Temp. °C
50	.1235	1.0121	12.032	209.32	2443.5	209.33	2382.7	2592.1	.7038	8.0763	50
55	.1576	1.0146	9.568	230.21	2450.1	230.23	2370.7	2600.9	.7679	7.9913	55
60	.1994	1.0172	7.671	251.11	2456.6	251.13	2358.5	2609.6	.8312	7.9096	60
65	.2503	1.0199	6.197	272.02	2463.1	272.06	2346.2	2618.3	.8935	7.8310	65
70	.3119	1.0228	5.042	292.95	2469.6	292.98	2333.8	2626.8	.9549	7.7553	70
75	.3858	1.0259	4.131	313.90	2475.9	313.93	2321.4	2635.3	1.0155	7.6824	75
80	.4739	1.0291	3.407	334.86	2482.2	334.91	2308.8	2643.7	1.0753	7.6122	80
85	.5783	1.0325	2.828	355.84	2488.4	355.90	2296.0	2651.9	1.1343	7.5445	85
90	.7014	1.0360	2.361	376.85	2494.5	376.92	2283.2	2660.1	1.1925	7.4791	90
95	.8455	1.0397	1.982	397.88	2500.6	397.96	2270.2	2668.1	1.2500	7.4159	95
100	1.014	1.0435	1.673	418.94	2506.5	419.04	2257.0	2676.1	1.3069	7.3549	100
110	1.433	1.0516	1.210	461.14	2518.1	461.30	2230.2	2691.5	1.4185	7.2387	110
120	1.985	1.0603	0.8919	503.50	2529.3	503.71	2202.6	2706.3	1.5276	7.1296	120
130	2.701	1.0697	0.6685	546.02	2539.9	546.31	2174.2	2720.5	1.6344	7.0269	130
140	3.613	1.0797	0.5089	588.74	2550.0	589.13	2144.7	2733.9	1.7391	6.9299	140
150	4.758	1.0905	0.3928	631.68	2559.5	632.20	2114.3	2746.5	1.8418	6.8379	150
160	6.178	1.1020	0.3071	674.86	2568.4	675.55	2082.6	2758.1	1.9427	6.7502	160
170	7.917	1.1143	0.2428	718.33	2576.5	719.21	2049.5	2768.7	2.0419	6.6663	170
180	10.02	1.1274	0.1941	762.09	2583.7	763.22	2015.0	2778.2	2.1396	6.5857	180
190	12.54	1.1414	0.1565	806.19	2590.0	807.62	1978.8	2786.4	2.2359	6.5079	190
200	15.54	1.1565	0.1274	850.65	2595.3	852.45	1940.7	2793.2	2.3309	6.4323	200
210	19.06	1.1726	0.1044	895.53	2599.5	897.76	1900.7	2798.5	2.4248	6.3585	210
220	23.18	1.1900	0.08619	940.87	2602.4	943.62	1858.5	2802.1	2.5178	6.2861	220
230	27.95	1.2088	0.07158	986.74	2603.9	990.12	1813.8	2804.0	2.6099	6.2146	230
240	33.44	1.2291	0.05976	1033.2	2604.0	1037.3	1766.5	2803.8	2.7015	6.1437	240
250	39.73	1.2512	0.05013	1080.4	2602.4	1085.4	1716.2	2801.5	2.7927	6.0730	250
260	46.88	1.2755	0.04221	1128.4	2599.0	1134.4	1662.5	2796.6	2.8838	6.0019	260
270	54.99	1.3023	0.03564	1177.4	2593.7	1184.5	1605.2	2789.7	2.9751	5.9301	270
280	64.12	1.3321	0.03017	1227.5	2586.1	1236.0	1543.6	2779.6	3.0668	5.8571	280
290	74.36	1.3656	0.02557	1278.9	2576.0	1289.1	1477.1	2766.2	3.1594	5.7821	290
300	85.81	1.4036	0.02167	1332.0	2563.0	1344.0	1404.9	2749.0	3.2534	5.7045	300
320	112.7	1.4988	0.01549	1444.6	2525.5	1461.5	1238.6	2700.1	3.4480	5.5362	320
340	145.9	1.6379	0.01080	1570.3	2464.6	1594.2	1027.9	2622.0	3.6594	5.3357	340
360	186.5	1.8925	0.006945	1725.2	2351.5	1760.5	720.5	2481.0	3.9147	5.0526	360
374.14	220.9	3.155	0.003155	2029.6	2029.6	2099.3	0	2099.3	4.4298	4.4298	374.14

Source: Tables A-2 through A-5 are extracted from J. H. Keenan, F. G. Keyes, P. G. Hill, and J. G. Moore, *Steam Tables*, Wiley, New York, 1969.

H_2O

TABLE A-3 Properties of Saturated Water (Liquid–Vapor): Pressure Table

Press. bar	Temp. °C	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Specific Volume m³/kg Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Internal Energy kJ/kg Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Enthalpy kJ/kg Evap. h_{fg}	Enthalpy kJ/kg Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Entropy kJ/kg · K Sat. Vapor s_g	Press. bar
0.04	28.96	1.0040	34.800	121.45	2415.2	121.46	2432.9	2554.4	0.4226	8.4746	0.04
0.06	36.16	1.0064	23.739	151.53	2425.0	151.53	2415.9	2567.4	0.5210	8.3304	0.06
0.08	41.51	1.0084	18.103	173.87	2432.2	173.88	2403.1	2577.0	0.5926	8.2287	0.08
0.10	45.81	1.0102	14.674	191.82	2437.9	191.83	2392.8	2584.7	0.6493	8.1502	0.10
0.20	60.06	1.0172	7.649	251.38	2456.7	251.40	2358.3	2609.7	0.8320	7.9085	0.20
0.30	69.10	1.0223	5.229	289.20	2468.4	289.23	2336.1	2625.3	0.9439	7.7686	0.30
0.40	75.87	1.0265	3.993	317.53	2477.0	317.58	2319.2	2636.8	1.0259	7.6700	0.40
0.50	81.33	1.0300	3.240	340.44	2483.9	340.49	2305.4	2645.9	1.0910	7.5939	0.50
0.60	85.94	1.0331	2.732	359.79	2489.6	359.86	2293.6	2653.5	1.1453	7.5320	0.60
0.70	89.95	1.0360	2.365	376.63	2494.5	376.70	2283.3	2660.0	1.1919	7.4797	0.70
0.80	93.50	1.0380	2.087	391.58	2498.8	391.66	2274.1	2665.8	1.2329	7.4346	0.80
0.90	96.71	1.0410	1.869	405.06	2502.6	405.15	2265.7	2670.9	1.2695	7.3949	0.90
1.00	99.63	1.0432	1.694	417.36	2506.1	417.46	2258.0	2675.5	1.3026	7.3594	1.00
1.50	111.4	1.0528	1.159	466.94	2519.7	467.11	2226.5	2693.6	1.4336	7.2233	1.50
2.00	120.2	1.0605	0.8857	504.49	2529.5	504.70	2201.9	2706.7	1.5301	7.1271	2.00
2.50	127.4	1.0672	0.7187	535.10	2537.2	535.37	2181.5	2716.9	1.6072	7.0527	2.50
3.00	133.6	1.0732	0.6058	561.15	2543.6	561.47	2163.8	2725.3	1.6718	6.9919	3.00
3.50	138.9	1.0786	0.5243	583.95	2546.9	584.33	2148.1	2732.4	1.7275	6.9405	3.50
4.00	143.6	1.0836	0.4625	604.31	2553.6	604.74	2133.8	2738.6	1.7766	6.8959	4.00
4.50	147.9	1.0882	0.4140	622.25	2557.6	623.25	2120.7	2743.9	1.8207	6.8565	4.50
5.00	151.9	1.0926	0.3749	639.68	2561.2	640.23	2108.5	2748.7	1.8607	6.8212	5.00
6.00	158.9	1.1006	0.3157	669.90	2567.4	670.56	2086.3	2756.8	1.9312	6.7600	6.00
7.00	165.0	1.1080	0.2729	696.44	2572.5	697.22	2066.3	2763.5	1.9922	6.7080	7.00
8.00	170.4	1.1148	0.2404	720.22	2576.8	721.11	2048.0	2769.1	2.0462	6.6628	8.00
9.00	175.4	1.1212	0.2150	741.83	2580.5	742.83	2031.1	2773.9	2.0946	6.6226	9.00
10.0	179.9	1.1273	0.1944	761.68	2583.6	762.81	2015.3	2778.1	2.1387	6.5863	10.0
15.0	198.3	1.1539	0.1318	843.16	2594.5	844.84	1947.3	2792.2	2.3150	6.4448	15.0
20.0	212.4	1.1767	0.09963	906.44	2600.3	908.79	1890.7	2799.5	2.4474	6.3409	20.0
25.0	224.0	1.1973	0.07998	959.11	2603.1	962.11	1841.0	2803.1	2.5547	6.2575	25.0
30.0	233.9	1.2165	0.06668	1004.8	2604.1	1008.4	1795.7	2804.2	2.6457	6.1869	30.0
35.0	242.6	1.2347	0.05707	1045.4	2603.7	1049.8	1753.7	2803.4	2.7253	6.1253	35.0
40.0	250.4	1.2522	0.04978	1082.3	2602.3	1087.3	1714.1	2801.4	2.7964	6.0701	40.0
45.0	257.5	1.2692	0.04406	1116.2	2600.1	1121.9	1676.4	2798.3	2.8610	6.0199	45.0
50.0	264.0	1.2859	0.03944	1147.8	2597.1	1154.2	1640.1	2794.3	2.9202	5.9734	50.0
60.0	275.6	1.3187	0.03244	1205.4	2589.7	1213.4	1571.0	2784.3	3.0267	5.8892	60.0
70.0	285.9	1.3513	0.02737	1257.6	2580.5	1267.0	1505.1	2772.1	3.1211	5.8133	70.0
80.0	295.1	1.3842	0.02352	1305.6	2569.8	1316.6	1441.3	2758.0	3.2068	5.7432	80.0
90.0	303.4	1.4178	0.02048	1350.5	2557.8	1363.3	1378.9	2742.1	3.2858	5.6772	90.0
100.	311.1	1.4524	0.01803	1393.0	2544.4	1407.6	1317.1	2724.7	3.3596	5.6141	100.
110.	318.2	1.4886	0.01599	1433.7	2529.8	1450.1	1255.5	2705.6	3.4295	5.5527	110.

TABLE A-3 (Continued)

Press. bar	Temp. °C	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Sat. Vapor s_g	Press. bar
120.	324.8	1.5267	0.01426	1473.0	2513.7	1491.3	1193.6	2684.9	3.4962	5.4924	120.
130.	330.9	1.5671	0.01278	1511.1	2496.1	1531.5	1130.7	2662.2	3.5606	5.4323	130.
140.	336.8	1.6107	0.01149	1548.6	2476.8	1571.1	1066.5	2637.6	3.6232	5.3717	140.
150.	342.2	1.6581	0.01034	1585.6	2455.5	1610.5	1000.0	2610.5	3.6848	5.3098	150.
160.	347.4	1.7107	0.009306	1622.7	2431.7	1650.1	930.6	2580.6	3.7461	5.2455	160.
170.	352.4	1.7702	0.008364	1660.2	2405.0	1690.3	856.9	2547.2	3.8079	5.1777	170.
180.	357.1	1.8397	0.007489	1698.9	2374.3	1732.0	777.1	2509.1	3.8715	5.1044	180.
190.	361.5	1.9243	0.006657	1739.9	2338.1	1776.5	688.0	2464.5	3.9388	5.0228	190.
200.	365.8	2.036	0.005834	1785.6	2293.0	1826.3	583.4	2409.7	4.0139	4.9269	200.
220.9	374.1	3.155	0.003155	2029.6	2029.6	2099.3	0	2099.3	4.4298	4.4298	220.9

Fundamental Parameters

The Gibbs Dalton law for a mixture of perfect gases states that the mixture pressure is equal to the sum of the partial pressure of the constituents

$$P = p_1 + p_2 + p_3 + \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2.5)$$

For moist air

$$P = p_{N2} + p_{O2} + p_{Co2} + p_A + p_v \ldots\ldots\ldots\ldots\ldots\ldots \quad (2.6)$$

Because the various constituents of the dry air may be considered to be one gas,

$$P_B = p_a + p_v \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2.7)$$

Where:

P_B : Barometric pressure (Atmospheric pressure)

P_a : dry air pressure

P_v : water vapor pressure

Example:

Saturated air at 26°C and atmospheric pressure 101325 N/m^2 . Find the partial pressure for each of the dry air and the water vapor?

Solution:

From table A-2 at temperature 26°C, the saturated pressure of the water vapor is

P_v = 3363 Pa (N/m^2) , and from Eqn. (2.7)

$P_B = p_a + p_v$ or $p_a = P_B - p_v$

$= 101325 - 3363 = 97962$ Pa = 97.962 kPa

Vapor partial pressure in unsaturated air

These cases are the most common in nature, the vapor pressure can be calculated from empirical formula:

$$P_v = P_{wss} - P_B\, A\, (t_d - t_w) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2.8)$$

Where:

P_v: partial pressure of water vapor

P_{wss}: saturated pressure of vapor at wet bulb temperature (t_w)

P_B: barometric pressure

t_d: dry bulb temperature

t_w: wet bulb temperature

A: constant

$A = 6.66 \times 10^{-4}$ if $t_w \geq 0^oC$

$A = 5.94 \times 10^{-4}$ if $t_w < 0^oC$

Example:

Calculate the water vapor pressure in wet air under 20°C dry bulb and 15°C wet bulb temperature and P_B = 950 mbar?

Solution:

$P_v = P_{wss} - P_B A (t_d - t_w)$

From steam table A-2 at $t_w = 15^oC$

$P_{wss} = 1.705$ kPa

$P_v = 1.705 - 95 \times 6.66 \times 10^{-4} (20 - 15) = 1.388$ kPa

Moisture content (W)

Sometimes called the specific humidity or humidity ratio, it is the ratio of the mass of water vapor (m_v) to the mass of the dry air (m_a) in the mixture:

$$W = m_v / m_a \quad \ldots\ldots \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2.8)$$

To find the values of (m_v) and (m_a) using equation (2.3)

$$m_v = P_v V_v / R_v T_v \quad , \quad m_a = P_a V_a / R_a T_a$$

but $W = m_v / m_a$

and $V_v = V_a$, $T_v = T_a$

$$W = (P_v/R_v)/(P_a/R_a) = (P_v/P_a).(R_a/R_v) = 0.622\, P_v/P_a = 0.622\, P_v/(P_B - P_v) \quad \ldots\ldots (2.9)$$

Example:

Air at 20°C dry bulb and 15°C wet bulb and the barometric pressure is 95 kPa. Calculate the moisture content of the air and the density of the vapor.

Solution:

From steam table A-2 at t_w= 15°C

P_{wss} = 1.705 kPa

$P_v = P_{wss} - P_B . A. (t_d - t_w)$

$= 1.705 - 95 \times 6.66 \times 10^{-4}(20 - 15) = 1.389$ kPa

$W = 0.622 (P_v)/(P_B - P_v) = 0.622 (1.389)/ (95 - 1.389) = 0.00923$ kg/kg dry air

Density of the vapor, $P_v = \rho_v.R_v.T_V \rightarrow \rho_v = (1389)/(461)(20+273) = 0.01028$ kg/m^3

Relative humidity (ϕ)

Is the ratio of the mole fraction of the water vapor (X_v) in a mixture to the mole fraction of the water vapor in a saturated mixture (X_s) at the same temperature and pressure.

$\Phi = [X_v/X_s]_{t, p}$ …… (2.10)

For a mixture of perfect gases, the mole fraction is equal to the partial pressure ratio of each constituent. The mole fraction of water vapor is:

$X_v = P_v /P$

Then; $\Phi = (P_v /P)/(P_{ss} /P) = P_v/P_{ss}$ …… (2.11)

Since the temperature of the dry air and the water vapor are assumed to be the same in the mixture;

$\Phi = (P_v /R_v.T)/(P_{ss} /R_v.T) = [\rho_v /\rho_{ss}]_{t, p}$ …….. (2.12)

By using the perfect gas law we can derive a relation between the relative humidity (Φ) and the moisture content (W). Combining Eqns. (2.9) and (2.11)

$\Phi = (W. P_B) / (0.622 +W) P_{ss}$ ……. (2.13)

Example:

Air at 24°C and 40% RH, and P_B= 92 kPa. Find the vapor density (ρ_v) and vapor pressure (P_v).

Solution:

From Table A-2 at t_d=24°C, υ = 45.883 m^3/kg

ρ_{ss}= 1/ υ = 1/45.883 = 0.02179 kg/m^3

Φ =[ρ_v /ρ_{ss}]$_{t, p}$ → ρ_v= Φ. ρ_{ss} = 0.4 x 0.02179 = 0.008716 kg/m^3

P_v= ρ_v R_v T = 0.008716 x 461x (24+273) = 1193.4 Pa

Or by other way;

Φ =[ρ_v /ρ_{ss}]$_{t, p}$, from Table A-2 at t=24°C → P_{ss}= 2985 Pa

P_v= 0.4 x 2985 = 1194 Pa

Degree of saturation (μ)

Is the ratio of the moisture content (W) to the moisture content of a saturated mixture (W_{ss}) at the same temperature and pressure.

μ =[W/W_{ss}]$_{t, p}$ …….. (2.14)

W= m_s/m_a = (P_s/(P_B – P_s)) .(R_a/R_s)

W_{ss}= (P_{ss}/(P_B – P_{ss})) .(R_a/R_s)

μ = (P_s/(P_B – P_s)) .((P_B – P_{ss})/ P_{ss}) .(R_a/R_s)(R_s/R_a) = (P_s/P_{ss}).(P_B – P_{ss})/(P_B –P_s)

= (P_s/P_{ss}). (1-P_{ss}/P_B)/(1- P_s/P_B) = Φ[(1- P_{ss}/P_B)/(1- Φ. P_{ss}/P_B)] …….. (2.15)

Example:

Moist air at 40°C DBT, 30°C WBT and 101 kPa barometric pressure, calculate for the air:

a) Relative humidity (Φ)

b) Moisture content (W)

c) Degree of saturation (μ)

Solution:

P_v = P_{wss} – P_B. A. (t_d – t_w) , from Table A-2

P_{wss}= 4.246 kPa at 30°C, P_{dss}= 7.384 kPa at 40°C

$P_v = 4.246 – 101 \times 6.66\times10^{-4} (40 – 30) = 3.57$ kPa

a) $\Phi = P_v/P_{dss} \times 100\% = (3.57)/(7.384) \times 100\% = 48.4\%$

b) $W = 0.622\ P_v/ (P_B – P_v) = 0.622(3.57)/(101 – 3.57) = 0.0228\ kg/kg_{d.a}$

c) $\mu =[W/W_{ss}]_{t, p}$

$W_{ss}= 0.622(7.384)/(101 – 7.384) = 0.049\ kg/kg_{d.a}$

$\mu = 0.0228/0.049 = 0.465 = 46.5\%$

Or:

$\mu = \Phi[(1- P_{ss}/P_B)/(1- \Phi.\ P_{ss}/P_B)] = (0.484)[1 – (7.384/101)]/[1 – 0.484(7.384/101)]$

$= 46.5\%$

Dew point temperature (t_{dp})

This is defined as the temperature of saturated air which has the same vapor pressure as the moist air under consideration. It may also be stated as a mixture is cooled at constant pressure, the temperature at which condensation first begins is the dew point.

Example:

Find the dew point temperature for air at 20°C DBT and 15°C WBT and atmospheric pressure is 95 kPa.

Solution:

From Table A-2 at $t_w = 15°C$, $P_{wss}= 1.705$ kPa

$P_v = P_{wss} – P_B.\ A.\ (t_d – t_w)$

$= 1.705 – 95 \times 6.66 \times10^{-4} (20 – 15) = 1.388$ kPa

From Table A-2, we find the saturation temperature that corresponds to P_v

$t_{sat} = t_{dp} = 11.84°C$

Specific volume (υ)

This is defined as the volume of one kg of moist air, it can calculated by three ways:

1. Used of dry air mass and partial pressure

2. Used of vapor mass and partial pressure

3. Used of mass of mixture and its total pressure.

Example:

Calculate the specific volume (υ) of air at 35°C DBT and 25°C WBT at a barometric pressure of 95 kPa.

Solution:

$P_a V_a = m_a R_a T_a \rightarrow \upsilon_a = R_a T_a / P_a$

But; $P_a = P_B - P_v$

$P_{wss} = 3.169$ kPa at $t = 25°C$

$P_v = 3.169 - 95 \times 6.66 \times 10^{-4} (35 - 25) = 2.5363$ kPa

Then $P_a = 95 - 2.5363 = 92.46$ kPa

$\upsilon_a = (0.287)(35 + 273)/(92.46) = 0.956\ m^3/kg$

Or by other way;

$W = 0.622 (P_v)/(P_B - P_v) = 0.622(2.5363)/(95 - 2.5363) = 0.01706\ kg/kg_{d.a}$

So for 1 kg of dry air there is 0.01706 kg of steam ; $m_s = 0.01706$ kg

$\upsilon_v = (0.01706)(0.461)(35 + 273)/(2.5363) = 0.956\ m^3/kg$

Enthalpy (h)

If a heat exchange occurs at constant pressure, as well as a change in internal energy taking place, work may be done.

This leads to a definition of enthalpy, H:

$H = U + pV$

The equation is strictly true for a pure gas of mass m, pressure p, and volume V. However it may be applied without appreciable error to the mixtures of gases associated with air conditioning.

The enthalpy, h, used in psychrometry is the specific enthalpy of moist air, expressed in kJ/kg dry air, defined by the equation:

$$h = h_a + W\, h_v \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.16)$$

where ; h_a is the enthalpy of dry air, h_v is the enthalpy of water vapor, both expressed in kJ kg^{-1}, and W is the moisture content.

An approximate equation for the enthalpy of dry air over the range 0°C to 60°C is, however

$$h_a = 1.007\, t_d - 0.026 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.17)$$

and for lower temperatures, down to – 10°C, the approximate equation is

$$h_a = 1.005\, t_d \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.18)$$

For purposes of approximate calculation, we may assume that, in the range 0°C to 60°C, the vapor is generated from liquid water at 0°C and that the specific heat of superheated steam is a constant. The following equation can then be used for the enthalpy of water vapor:

$$h_v = 2501 + 1.84\, t_d \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.19)$$

Equations (2.17) and (2.19) can now be combined, as typified by equation (2.16), to give an approximate expression for the enthalpy of humid air at a barometric pressure of 101.325 kPa:

$$h = (1.007\, t_d - 0.026) + W(2501 + 1.84\, t_d) \quad \ldots\ldots\ldots\ldots\ldots\ldots (2.20)$$

Example:

Compute the enthalpy of saturated air at 15°C and standard atmospheric pressure.

Solution:

Because the air is saturated, $W = W_s$

$W_s = 0.622(P_{vs})/(P_B - P_{vs})$ from Table A-2 at t=15°C, $P_{vs} = 1705$ Pa

$W_s = 0.622(1705)/(101325 - 1705) = 0.01065\ kg/kg_{d.a}$

$h = (1.007\, t_d - 0.026) + W(2501 + 1.84\, t_d)$

$= (1.007x\ 15 - 0.026) + 0.01065(\ 2501 + 1.84x\ 15) = 42$ kJ/kg

Example:

Compute the enthalpy of moist air at 20°C DBT and 15°C WBT and standard atmospheric pressure.

Solution:

$P_v = P_{wss} - P_B.\ A.\ (t_d - t_w)$

$= 1.705 - 101.325 \times 6.66 \times 10^{-4}(20 - 15) = 1.3676$ kPa

$W_s = 0.622(P_{vs})/(P_B - P_{vs})$

$= 0.622(1.3676)/(101.325 - 1.3676) = 0.00843$ kg/kg$_{d.a}$

$h = (1.007 \times 20 - 0.026) + 0.00843(2501 + 1.84 \times 20) = 41.508$ kJ/kg

Example:

Moist air at 42°C DBT, 26°C WBT and 100 kPa barometric pressure. Calculate:

a) vapor pressure b) relative humidity c) moisture content d) specific enthalpy

e) specific volume f) dew point temperature g) degree of saturation

Solution:

From Table A-2

$P_{wss} = 3.363$ kPa at $t_w = 26°C$

$P_{dss} = 8.268$ kPa at $t_d = 42°C$

a) $P_v = 3.363 - 100 \times 6.66 \times 10^{-4}\ (42 - 26) = 2.2974$ kPa

b) $\Phi = P_v/P_{dss} = 2.2974/8.268 = 0.2779 = 27.79\%$

c) $W = 0.622\ (2.2974)/(100 - 2.2974) = 0.0146$ kg/kg$_{d.a}$

d) $h = (1.007\ t_d - 0.026) + W(2501 + 1.84\ t_d)$

$= (1.007 \times 42 - 0.026) + 0.0146(2501 + 1.84 \times 42) = 79.91$ kJ/kg

e) $\upsilon_a = R_a\ T_a / P_a = (0.287)(42+273)/(100 - 2.2974) = 0.9253$ m^3/kg

f) the dew point temperature is the temperature corresponds to P_v

$t_{dp} = 19.7°C$

g) $\mu = \Phi[(1- P_{ss}/P_B)/(1- \Phi . P_{ss}/P_B)] = 0.2779 [(1- 8.268/100)/(1 – 0.2779 \times 8.268/100)]$

$= 0.26092$

Example:

In a check on an air-conditioning system, a room was maintained at 22°C DBT and 53% RH, by air supply at 1.6 m^3/s with 10°C DBT and 9°C WBT. The barometric pressure was 920 mbar. Find:

a) sensible heat gain b) latent heat gain c) total heat gain

Solution:

From Table A-2 at $t_w = 9°C$, $P_{wss} = 1.15$ kPa

$P_v = 1.15 – 92 \times 6.66 \times 10^{-4}(10 – 9) = 1.0887$ kPa

$P_a = 92 – 1.0887 = 90.91$ kPa

$\dot{m}_a = (P_a V_a)/(R_a T_a) = (90.91 \times 1.6)/(0.287 \times 295) = 1.718$ kg/s

$h_{a1} = 1.007 \times 22 – 0.026 = 22.128$ kJ/kg

$h_{a2} = 1.007 \times 10 – 0.026 = 10.044$ kJ/kg

$Q_s = \dot{m}_a(h_{a1} – h_{a2}) = 1.718(22.128 – 10.044) = 20.7$ kW

Also, from Table A-2 at t=22°C $P_{dss} = 2.645$ kPa

$\Phi = P_v/P_{dss} \rightarrow P_v = 0.53 \times 2.645 = 1.4$ kPa

$W_1 = 0.622(1.4)/(92 – 1.4) = 0.00961$ kg/kg$_{d.a}$

$W_2 = 0.622(1.0887)/(92 – 1.0887) = 0.00745$ kg/kg$_{d.a}$

$h_{v1} = 2501 + 1.84 \times 22 = 2541.48$ kJ/kg

$h_{v2} = 2501 + 1.84 \times 10 = 2519.4$ kJ/kg

$Q_L = \dot{m}_a(W_1 . h_{v1} – W_2 . h_{v2}) = 1.718(0.00961 \times 2541.48 - 0.00745 \times 2519.4) = 9.714$ kW

$Q_{Tot} = Q_s + Q_L = 20.7 + 9.714 = 30.414$ kW

Adiabatic saturation and thermodynamic wet bulb temperature

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

As shown in Fig.2.1 an adiabatic saturator is a device in which air flows through an infinitely long duct containing water. As the air comes in contact with water in the duct, there will be heat and mass transfer between water and air. If the duct is infinitely long, then at the exit, there would exist perfect equilibrium between air and water at steady state. Air at the exit would be fully saturated and its temperature is equal to that of water temperature. The device is adiabatic as the walls of the chamber are thermally insulated. In order to continue the process, make-up water has to be provided to compensate for the amount of water evaporated into the air. The temperature of the make-up water is controlled so that it is the same as that in the duct.

After the adiabatic saturator has achieved a steady-state condition, the temperature indicated by the thermometer immersed in the water is the *thermodynamic wet-bulb temperature*. The thermodynamic wet bulb temperature will be less than the entering air DBT but greater than the dew point temperature.

Certain combinations of air conditions will result in a given sump temperature, and this can be defined by writing the energy balance equation for the adiabatic saturator. Based on a unit mass flow rate of dry air, this is given by:

$$h_1 = h_2 - (W_2 - W_1)h_f \qquad \text{..... (2.21)}$$

where h_f is the enthalpy of saturated liquid at the sump or thermodynamic wet-bulb temperature, h_1 and h_2 are the enthalpies of air at the inlet and exit of the adiabatic saturator, and W_1 and W_2 are the humidity ratio of air at the inlet and exit of the adiabatic saturator, respectively.

It is to be observed that the thermodynamic wet-bulb temperature is a thermodynamic property, and is independent of the path taken by air. Assuming the humid specific heat to be constant, from the enthalpy balance, the thermodynamic wet-bulb temperature can be written as:

$$t_2 = t_1 - \frac{h_{fg,2}}{c_{pm}}(w_2 - w_1) \qquad \text{..... (2.22)}$$

where $h_{fg,2}$ is the latent heat of vaporization at the saturated condition 2. Thus measuring the dry bulb (t_1) and wet bulb temperature (t_2) one can find the inlet humidity ratio (W_1) from the above expression as the outlet saturated humidity ratio (W_2) and latent heat heat of vaporizations are functions of t_2 alone (at fixed barometric pressure).

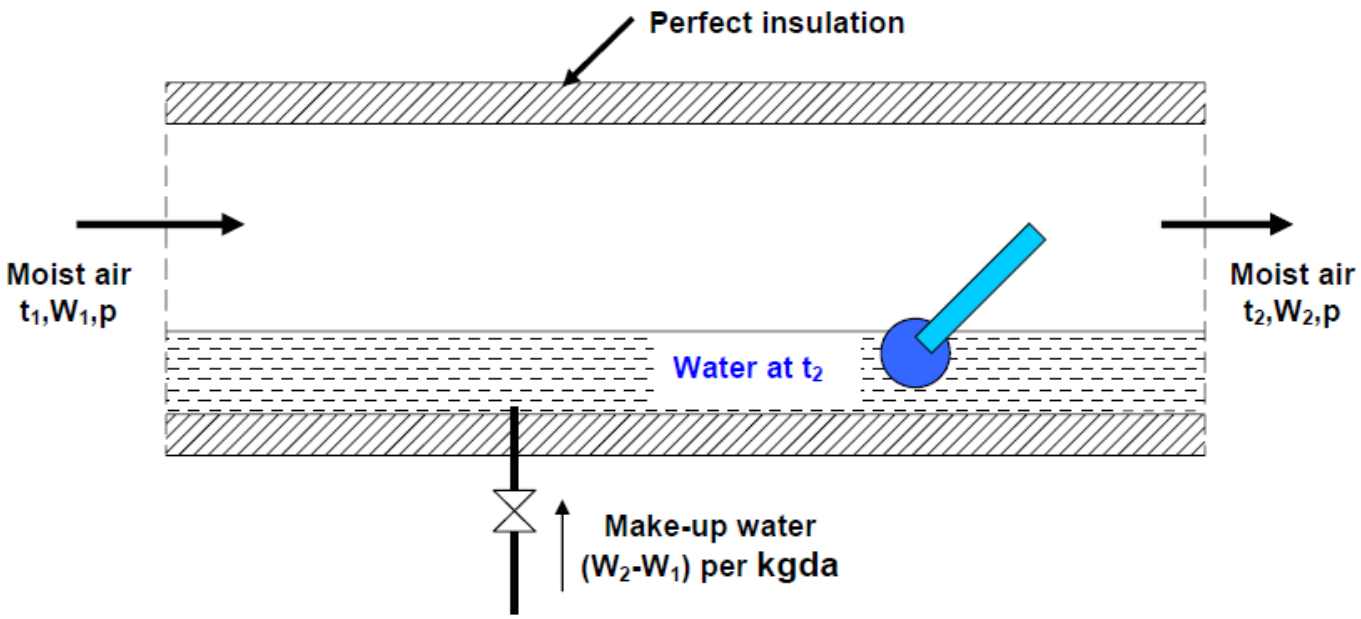


Fig.2.1 The process of adiabatic saturation of air

Example:

Moist air at 26°C dry bulb and 16°C wet bulb enters to adiabatic device. The air leaving the device completely saturated at standard atmospheric pressure (P_{atm}=101.325 kPa)

Water flow at 16°C, Calculate:

a) moisture content at the entry and exit W_1 & W_2

b) degree of saturation at the entry μ_1

c) specific volume at the entry and exit υ_1 & υ_2

d) enthalpy at the entry and exit h_1 & h_2

Solution:

a) From Table A-2 at $t_w = 16^oC$, $P_{wss} = 1.818$ kPa

$P_v = 1.818 - 101.325 \times 6.66 \times 10^{-4}(26 - 16) = 1.1432$ kPa

$W_1 = 0.622(1.1432)/(101.325 - 1.1432) = 0.00709$ kg/kg dry air

$W_2 = W_{sw} = 0.622(1.818)/(101.325 - 1.818) = 0.01136$ kg/kg dry air

b) $\mu = [W/W_{ss1}]_{t,p}$

where (W_{ss}) is the moisture content for saturated air at entry state

From Table A-2 at $t = 26^oC$, $P_{ss} = 3.360$ kPa

$W_{ss1} = 0.622(3.360)/(101.325 - 3.360) = 0.0213$ kg/kg dry air

Then; $\mu = (0.00709)/(0.0213) = 0.333 = 33.3\%$

c) $P_{a1} = P_B - P_1 = 101.325 - 1.1432 = 100.182$ kPa

$P_{a2} = P_B - P_2 = 101.325 - 1.818 = 99.507$ kPa

$\upsilon_{a1} = R_a T_{a1}/ P_{a1} = 0.287 (273+26)/(100.183) = 0.857$ m^3/kg dry air

$\upsilon_{a2} = R_a T_{a2}/ P_{a2} = 0.287(273 +16)/(99.507) = 0.833$ m^3/kg dry air

d) $h = (1.007\, t_d - 0.026) + W(2501 + 1.84\, t_d)$

$h_1 = (1.007 \times 26 - 0.026) + 0.00709(2501 + 1.84 \times 26) = 44.23$ kJ/kg dry air

$h_2 = (1.007 \times 16 - 0.026) + 0.01136(2501 + 1.84 \times 16) = 44.83$ kJ/kg dry air

Also, can be calculate (h_2) from Eqn.(2.21)

$h_2 = h_1 + (W_2 - W_1)\, h_f$

From Table A-2 at $t = 16^oC$ $h_f = 67.19$ kJ/kg

$h_2 = 44.23 + (0.01136 - 0.00709) \times 67.19 = 44.517$ kJ/kg dry air

- ***Note here the enthalpy of the dry air decreases about 10 kJ/kg (from 26.208 to 16.138) while the enthalpy of water vapor is increases within the same amount 10 kJ/kg (from 18.071 to 28.745). This is an important fact in adiabatic saturation process.***

Exercises

1. The atmospheric condition of air are 25°C dry bulb temperature and moisture content of 0.01 kg/kg dry air. Find:

(a) partial pressure of vapor (b) relative humidity (c) dew point temperature.

[Ans. 0.016 bar, 50.6%, 14.1°C]

2. A sling psychrometer reads 40°C dry bulb temperature and 28°C wet bulb temperature. Calculate the following:

(a) moisture content (b) relative humidity (c) vapor density in air

(d) dew point temperature (e) enthalpy of mixture per kg of dry air.

[Ans. 0.019 kg/kg of dry air, 40.7%, 0.0208 kg/m^3, 24°C, 88.38 kJ/kg dry air]

3. A sample of moist air has a dry bulb temperature of 25°C and a relative humidity of 50%. The barometric pressure is 740 mm of Hg. Calculate:

(a) partial pressure of water vapor and dry air (b) dew point temperature

(c) specific humidity (moisture content) (d) enthalpy of air

[Ans. 0.01583 bar, 14°C, 0.0101 kg/kg dry air, 50.81 kJ/kg dry air]

4. The moist air exists at a total pressure of 1.01325 bar and 25°C dry bulb temperature. If the degree of saturation is 50%, determine the following using steam tables:

(a) moisture content (b) dew point temperature (c) specific volume of moist air.

[Ans. 10.03 g/kg dry air, 14°C, 0.857 m^3/kg]

5. The atmospheric conditions of air are 35°C dry bulb temperature, 60% relative humidity and 1.01325 bar pressure. If 0.005 kg of moisture per kg of dry air is removed, the temperature becomes 25°C. Determine the final relative humidity and dew point temperature.

[Ans. 88.6%, 23°C]

6. An atmospheric air enters the adiabatic saturator at 33°C dry bulb temperature and23°C wet bulb temperature. The barometric pressure is 740 mm Hg. Determine the moisture content and vapor pressure at 33°C.

[Ans. 0.012 kg/kg dry air, 13 mm Hg]

7. Air at 40°C DBT and 15% RH is passed through the adiabatic humidifier at the rate of 200 m^3/min. The outlet conditions of air are 25°C DBT and 20°C WBT. Find:

(a) dew point temperature (b) relative humidity of exit air

(c) amount of water vapor added to the air per minute.

[Ans. 17.8°C, 65%, 1.26 kg/min]

8. The atmospheric air has 35°C DBT and 50% RH and standard atmospheric pressure. Find:

(a) wet bulb temperature (b) moisture content (c) dew point temperature (d) enthalpy

[Ans. 26.20°C, 0.0178 $kg/kg_{d.a}$, 23°C, 81 kJ/kg]

9. An atmospheric air at 15°C DBT and 80% RH is supplied to the heating chamber at the rate of 100 m^3/min.. The leaving air has a temperature of 22°C without change in its moisture content. Determine the heat added to the air per min. and final relative humidity of the air.

[Ans. 865.3 kJ/min, 53%]

10. Moist air at 20°C DBT and the barometric pressure is 82.5 kPa. Calculate the enthalpy and the degree of saturation, if the air was:

(a) saturated (b) at a relative humidity of 50%

[Ans. 66.124 kJ/kg, 1.0 and 42.792 kJ/kg, 0.493]

Chapter Three

The Psychrometry of Air Conditioning Processes

3.1 The Psychrometric Chart

This provides a picture of the way in which the state of moist air alters as an air conditioning process takes place or a physical change occurs. Familiarity with the psychrometric chart is essential for a proper understanding of air conditioning.

Any point on the chart is termed a *state point*, the location of which, at a given barometric pressure, is fixed by any two of the psychrometric properties discussed in chapter 2. It is customary and convenient to design charts at a constant barometric pressure because barometric pressure does not alter greatly over much of the inhabited surface of the earth. When the barometric pressure is significantly different from the standard adopted for the chart or psychrometric tables to hand, then the required properties can be calculated using the equations derived earlier.

The British standard is that adopted by the Chartered Institution of Building Services Engineers for their Tables of Psychrometric Data and for their psychrometric chart. It is 101.325 kPa. The American standard is also 101.325 kPa and this value is used by the American Society of Heating, Refrigeration and Air Conditioning Engineers.

The psychrometric chart published by the CIBSE uses two fundamental properties, mass and energy, in the form of moisture content and enthalpy, as co-ordinates. As a result, mixture states lie on the straight line which joins the state points of the two constituents. Lines of constant dry-bulb temperature are virtually straight but divergent, only the isotherm for 30°C being vertical. The reason for this is that to preserve the usual appearance of a psychrometric chart, in spite of choosing the two fundamental properties as co-ordinates, the co-ordinate axes are oblique, not rectangular. Hence, lines of constant enthalpy are both straight and parallel, as are lines of constant moisture content. Since both these properties are taken as linear, the lines of constant enthalpy are equally spaced as are, also, the lines of constant moisture content. This is not true of the lines of constant humid volume and constant wet-bulb temperature, which are slightly curved and divergent. Since their curvature is only slight in the region of practical use on the chart, they can be regarded as straight without significant error resulting. In the sketches of psychrometric charts used throughout this text to illustrate changes of state, only lines of percentage saturation are shown curved. All others are shown straight, and dry-bulb isotherms are shown as vertical, for convenience.

The chart also has a protractor which allows the value of the ratio of the sensible heat gain to the total heat gain to be plotted on the chart. This ratio is an indication of the slope

of the room ratio line and is of value in determining the correct supply state of the air that must be delivered to a conditioned space. The zero value for the ratio is parallel to the isotherm for 30°C because the enthalpy of the added vapour depends on the temperature at which evaporation takes place, it being assumed that most of the latent heat gain to the air in a conditioned room is by evaporation from the skin of the occupants and that their skin surface temperature is about 30°C.

Figure 3.1 shows a state point on a psychrometric chart.

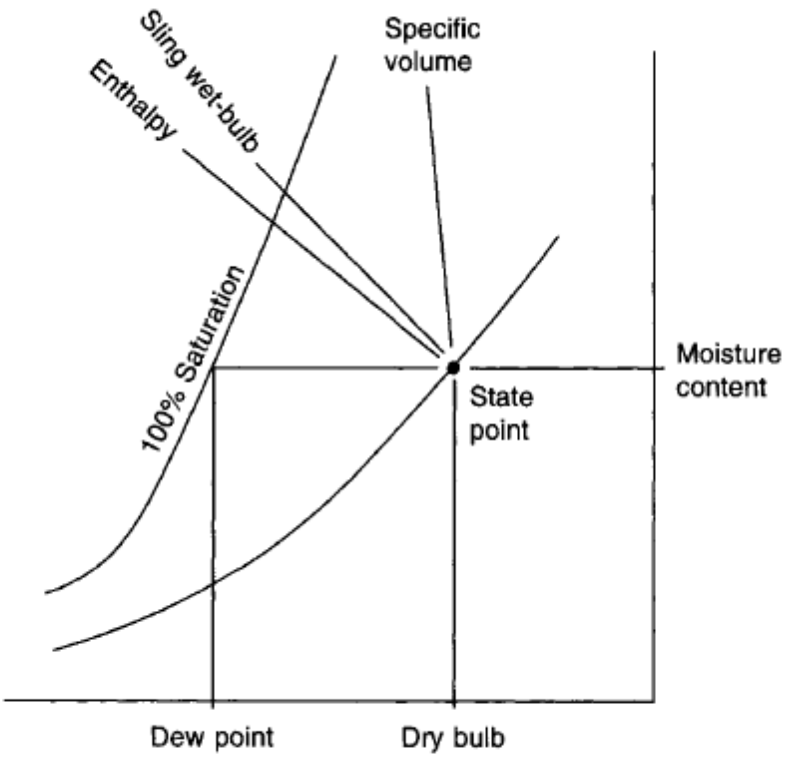


Fig. 3.1 The variables shown on the psychrometric chart.

3.2 Mixtures

Figure 3.2 shows what happens when two airstreams meet and mix adiabatically. Moist air at state 1 mixes with moist air at state 2, forming a mixture at state 3. The principle of the conservation of mass allows two mass balance equations to be written:

$m_{a1} + m_{a2} = m_{a3}$ for the dry air and

$g_1 m_{a1} + g_2 m_{a2} = g_3 m_{a3}$ for the associated water vapour

Hence

$$(g_1 - g_3)m_{a1} = (g_3 - g_2)m_{a2}$$

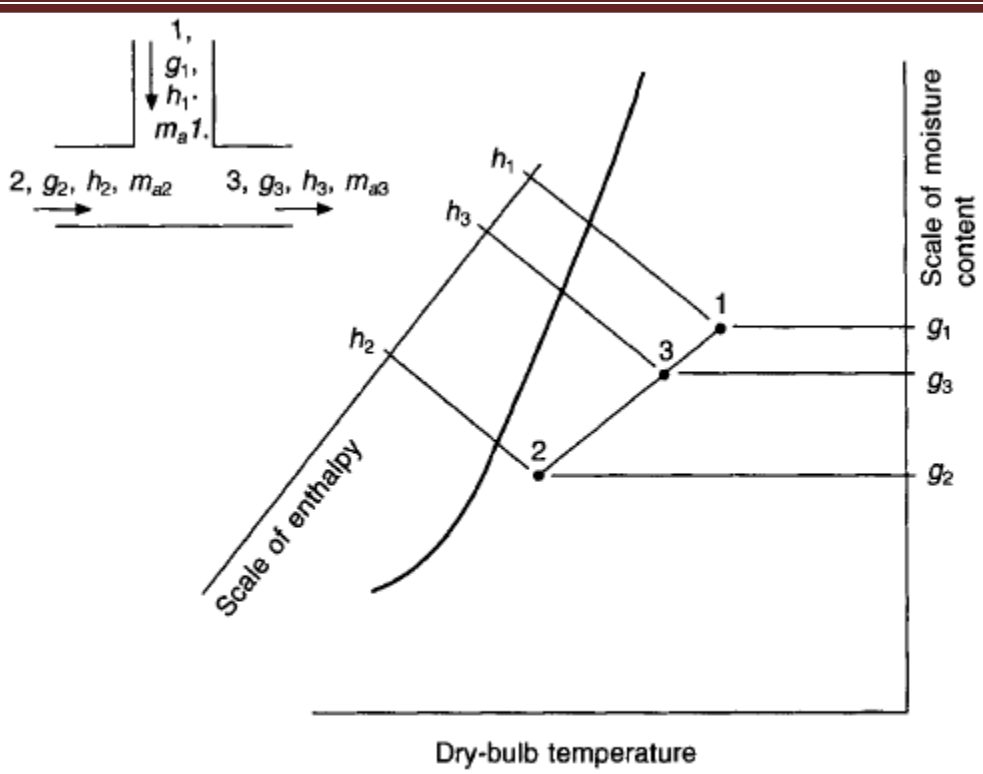


Fig. 3.2 The adiabatic mixing of two airstreams.

Therefore

$$\frac{g_1 - g_3}{g_3 - g_2} = \frac{m_{a2}}{m_{a1}}$$

Similarly, making use of the principle of the conservation of energy,

$$\frac{h_1 - h_3}{h_3 - h_2} = \frac{m_{a2}}{m_{a1}}$$

From this it follows that the three state points must lie on a straight line in a mass–energy co-ordinate system. When two airstreams mix adiabatically, the mixture state lies on the straight line which joins the state points of the constituents, and the position of the mixture state point is such that the line is divided inversely as the ratio of the masses of dry air in the constituent airstreams.

EXAMPLE 3.1

Moist air at a state of 60°C dry-bulb, 32.1°C wet-bulb (sling) and 101.325 kPa barometric pressure mixes adiabatically with moist air at 5°C dry-bulb, 0.5°C wet-bulb (sling) and 101.325 kPa barometric pressure. If the masses of dry air are 3 kg and 2 kg, respectively, calculate the moisture content, enthalpy and dry-bulb temperature of the mixture.

Answer

g_1 = 18.400 g per kg dry air

$g_2 = 2.061$ g per kg dry air
$h_1 = 108.40$ kJ per kg dry air
$h_2 = 10.20$ kJ per kg dry air

The principle of the conservation of mass demands that

$$g_1 m_{a1} + g_2 m_{a2} = g_3 m_{a3}$$
$$= g_3(m_{a1} + m_{a2})$$

hence

$$g_3 = \frac{g_1 m_{a1} + g_2 m_{a2}}{m_{a1} + m_{a2}}$$
$$= \frac{18.4 \times 3 + 2.061 \times 2}{3 + 2}$$
$$= \frac{59.322}{5}$$
$$= 11.864 \text{ g per kg dry air}$$

Similarly, by the principle of the conservation of energy,

$$h_3 = \frac{h_1 m_{a1} + h_2 m_{a2}}{m_{a1} + m_{a2}}$$
$$= \frac{108.40 \times 3 + 10.20 \times 2}{3 + 2}$$
$$= 69.12 \text{ kJ per kg dry air}$$

To determine the dry-bulb temperature, the following practical equation must be used

$$h = (1.007t - 0.026) + g(2501 + 1.84t)$$

Substituting the values calculated for moisture content and enthalpy, this equation can be solved for temperature:

$$h = 69.12 = (1.007t - 0.026) + 0.011\,86(2501 + 1.84t)$$

$$t = \frac{39.48}{1.029} = 38.4°\text{C}$$

On the other hand, if the temperature were calculated by proportion, according to the masses of the dry air in the two mixing airstreams, a slightly different answer results:

$$t = \frac{3 \times 60 + 2 \times 5}{5}$$
$$= 38°\text{C}$$

EXAMPLE 3.2

A stream of moist air at a state of 21°C dry-bulb and 14.5°C wet-bulb (sling) mixes with another stream of moist air at a state of 28°C dry-bulb and 20.2°C wet-bulb (sling), the respective masses of the associated dry air being 3 kg and 1 kg. With the aid of CIBSE tables of psychrometric data calculate the dry-bulb temperature of the mixture (*a*) using the principles of conservation of energy and of mass and, (*b*), using a direct proportionality between temperature and mass.

3.3 Sensible heating and cooling

Sensible heat transfer occurs when moist air flows across a heater battery or over the coils of a sensible cooler. In the heater, the temperature of the medium used to provide the heat is not critical. The sole requirement for heat transfer is that the temperature shall exceed the final air temperature. In sensible cooling there is a further restriction: the lowest water temperature must not be so low that moisture starts to condense on the cooler coils. If such condensation does occur, through a poor choice of chilled water temperature, then the process will no longer be one of sensible cooling since dehumidification will also be taking place.

Figure 3.3 shows the changes of state which occur, sketched upon a psychrometric chart. The essence of both processes is that the change of state must occur along a line of constant moisture content. The variations in the physical properties of the moist air, for the two cases, are summarised below:

	Sensible heating	*Sensible cooling*
Dry-bulb	increases	decreases
Enthalpy	increases	decreases
Humid volume	increases	decreases
Wet-bulb	increases	decreases
Percentage saturation	decreases	increases
Moisture content	constant	constant
Dew point	constant	constant
Vapour pressure	constant	constant

EXAMPLE 3.3

Calculate the load on a battery which heats 1.5 $m^3 s^{-1}$ of moist air, initially at a state of 21°C dry-bulb, 15°C wet-bulb (sling) and 101.325 kPa barometric pressure, by 20 degrees. If low temperature hot water at 85°C flow and 75°C return is used to achieve this, calculate the flow rate necessary, in kilograms of water per second.

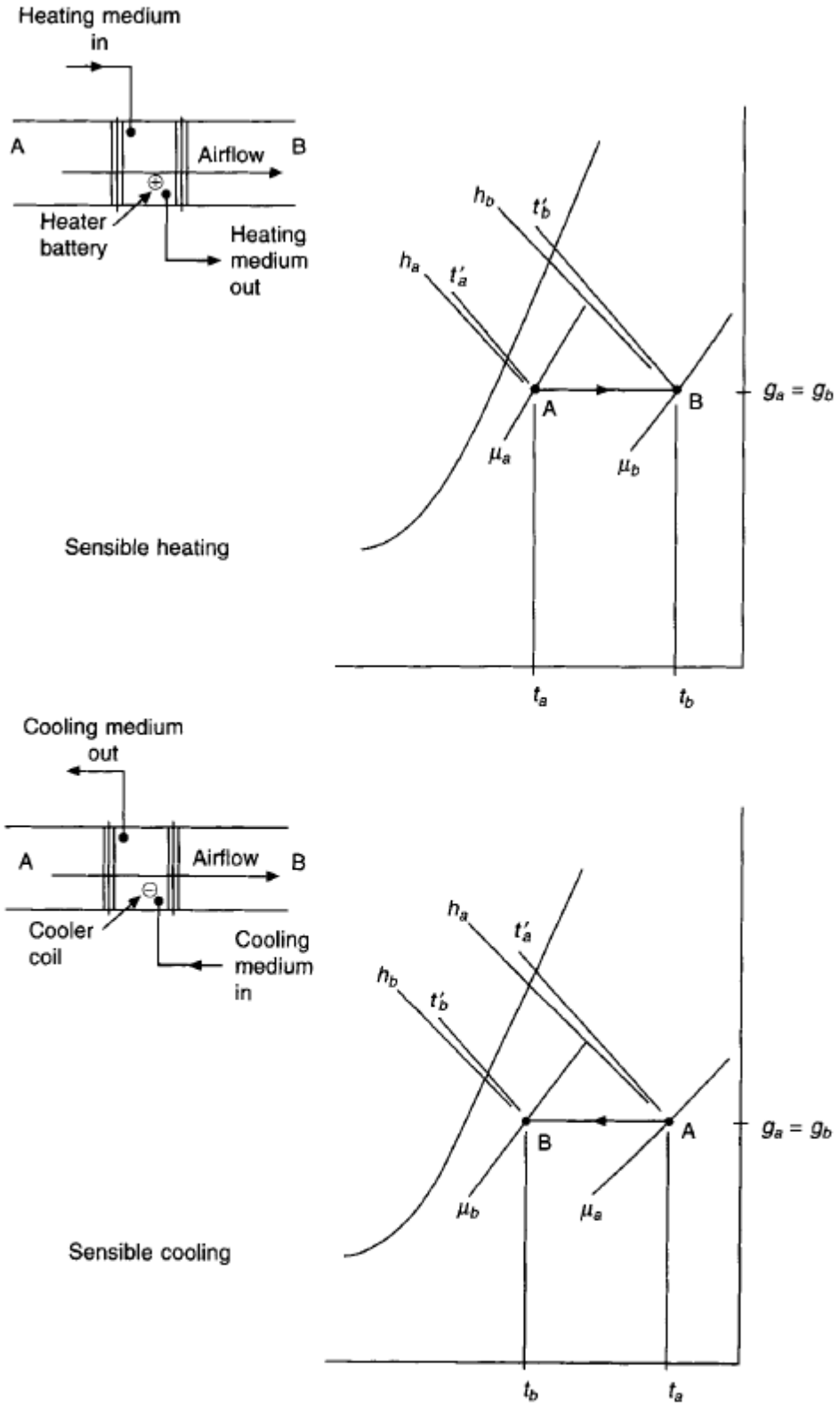


Fig. 3.3 Psychrometry for sensible heating and cooling.

Answer

$$\text{Heating load} = \begin{pmatrix}\text{mass flow of moist air expressed} \\ \text{in kg s}^{-1}\text{ of associated dry air}\end{pmatrix} \times$$

$$\begin{pmatrix}\text{increase in enthalpy of moist air expressed} \\ \text{in kJ per kg of associated dry air}\end{pmatrix}$$

From CIBSE tables of psychrometric data (or, less accurately, from the CIBSE psychrometric chart), the initial enthalpy is found to be 41.88 kJ kg^{-1}, the moisture content to be 8.171 g kg^{-1} and the humid volume to be 0.8439 m^3 per kg of dry air. Since the air is being heated by 20 degrees, reference must now be made to tables in order to determine the enthalpy at the same moisture content as the initial state but at a dry-bulb temperature of 41°C. By interpolation, the enthalpy of the moist air leaving the heater battery is found to be 62.31 kJ per kg of dry air.

$$\text{Heating load} = \left(\frac{1.5}{0.8439}\right) \times (62.31 - 41.88) = 36.3 \text{ kW}$$

$$\text{Flow rate of LTHW} = \frac{36.3}{(85° - 75°) \times 4.2} = 0.864 \text{ kg s}^{-1}$$

where 4.2 kJ/kg K is the specific heat capacity of water.

EXAMPLE 3.4

Calculate the load on a cooler coil which cools the moist air mentioned in example 3.3 by 5 degrees. What is the flow rate of chilled water necessary to effect this cooling if flow and return temperatures of 10°C and 15°C are satisfactory?

Answer

The initial enthalpy and humid volume are the same as in the first example. The final dry-bulb temperature of the moist air is 16°C but its moisture content is still 8.171 g per kg of dry air. At this state, its enthalpy is found from tables to be 36.77 kJ per kg of dry air.

$$\text{Cooling load} = \left(\frac{1.5}{0.8439}\right) \times (41.88 - 36.77) = 9.1 \text{ kW}$$

$$\text{Chilled water flow rate} = \frac{9.1}{5 \times 4.2} = 0.433 \text{ kg s}^{-1}$$

It is to be noted that, for sensible cooling, the selection of cooler coils and the choice of chilled water flow temperature require some care. See section 10.6.

3.4 Dehumidification

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

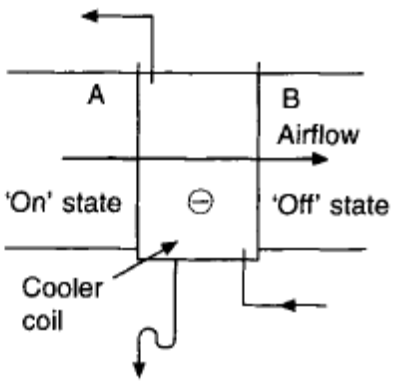
(i) cooling to a temperature below the dew point,
(ii) adsorption,
(iii) absorption,
(iv) compression followed by cooling.

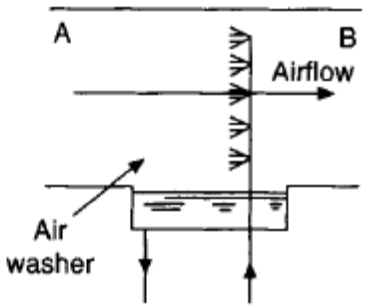
The first method forms the subject matter of this section.

Cooling to a temperature below the dew point is done by passing the moist air over a cooler coil or through an air washer provided with chilled water.

Figure 3.4 shows on a sketch of a psychrometric chart what happens when moist air is cooled and dehumidified in this fashion. Since dehumidification is the aim, some of the spray water or some part of the cooler coil must be at a temperature less than the dew point of the air entering the equipment. In the figure, t_d is the dew point of the moist air 'on' the coil or washer. The temperature t_c, corresponding to the point C on the saturation curve, is termed the *apparatus dew point.* This term is in use for both coils and washers but, in the case of cooler coils alone, t_c is also sometimes termed the *mean coil surface temperature.* The justification for this latter terminology is offered in chapter 10.

For purposes of carrying out air conditioning calculations, it is sufficient to know the





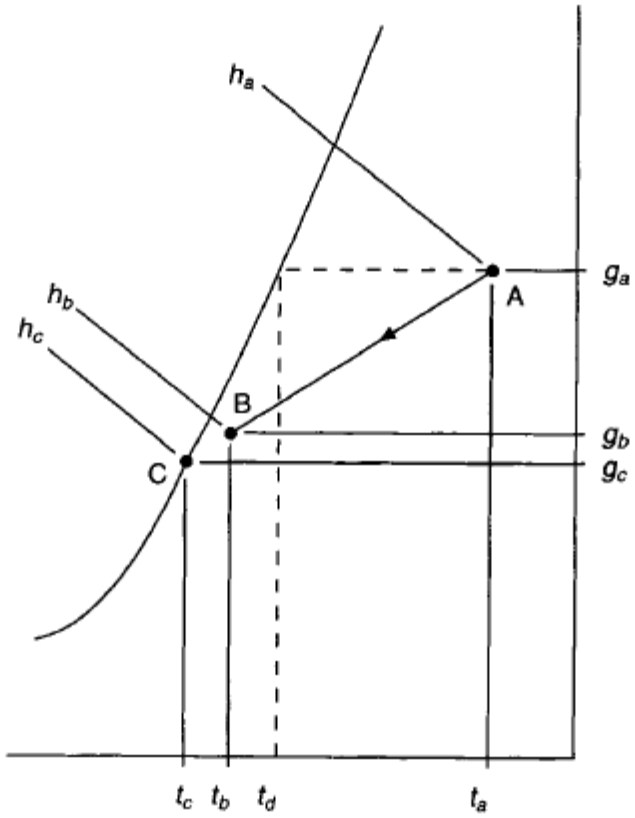


Fig. 3.4 Cooling and dehumidification by a cooler coil or an air washer.

state A of the moist air entering the coil, the state B of the air leaving the coil, and the mass flow of the associated dry air. What happens to the state of the air as it passes between points A and B is seldom of more than academic interest. Consequently, it is quite usual to show the change of state between the 'on' and the 'off' conditions as occurring along a straight line. In fact, as will be seen in chapter 10, the change of state follows a curved path, the curvature of which is a consequence of the heat transfer characteristics of the process, not of the construction of the psychrometric chart.

It can be seen from Figure 3.4 that the moisture content of the air is reduced, as also is its enthalpy and dry-bulb temperature. The percentage saturation, of course, increases. It might be thought that the increase of humidity would be such that the 'off' state, represented by the point B, would lie on the saturation curve. This is not so for the very good reason that no air washer or cooler coil is a hundred per cent efficient. It is unusual to speak of the efficiency of a cooler coil. Instead, the alternative terms, *contact factor* and *by-pass factor*, are used. They are complementary values and contact factor, sometimes denoted by β, is defined as

$$\beta = \frac{g_a - g_b}{g_a - g_c}$$

$$= \frac{h_a - h_b}{h_a - h_c} \qquad (3.1)$$

Similarly, by-pass factor is defined as

$$(1 - \beta) = \frac{g_b - g_c}{g_a - g_c}$$

$$= \frac{h_b - h_c}{h_a - h_c} \qquad (3.2)$$

It is sufficient, for all practical purposes, to assume that both these expressions can be rewritten in terms of dry-bulb temperature, namely that

$$\beta = \frac{t_a - t_b}{t_a - t_c} \qquad (3.3)$$

and that

$$(1 - \beta) = \frac{t_b - t_c}{t_a - t_c} \qquad (3.4)$$

It is much less true to assume that they can also be written, without sensible error, in terms of wet-bulb temperature, since the scale of wet-bulb values on the psychrometric chart is not at all linear. The assumption is, however, sometimes made for convenience, provided the values involved are not very far apart and that some inaccuracy can be tolerated in the answer.

Typical values of β are 0.82 to 0.92 for practical coil selection in the UK. In hot, humid climates more heat transfer surface is necessary and higher contact factors are common.

EXAMPLE 3.5

1.5 $m^3 s^{-1}$ of moist air at a state of 28°C dry-bulb, 20.6°C wet-bulb (sling) and 101.325 kPa flows across a cooler coil and leaves the coil at 12.5°C dry-bulb and 8.336 g per kg of dry air.

Determine (*a*) the apparatus dew point, (*b*) the contact factor and (*c*) the cooling load.

Answer

Figure 3.5 shows the psychrometric changes involved and the values immediately known from the data in the question. From CIBSE tables (or from a psychrometric chart) it is established that $h_a = 59.06$ kJ kg^{-1} and that $h_b = 33.61$ kJ kg^{-1}. The tables also give a value for the humid volume at the entry state to the coil, of 0.8693 m^3 kg^{-1}.

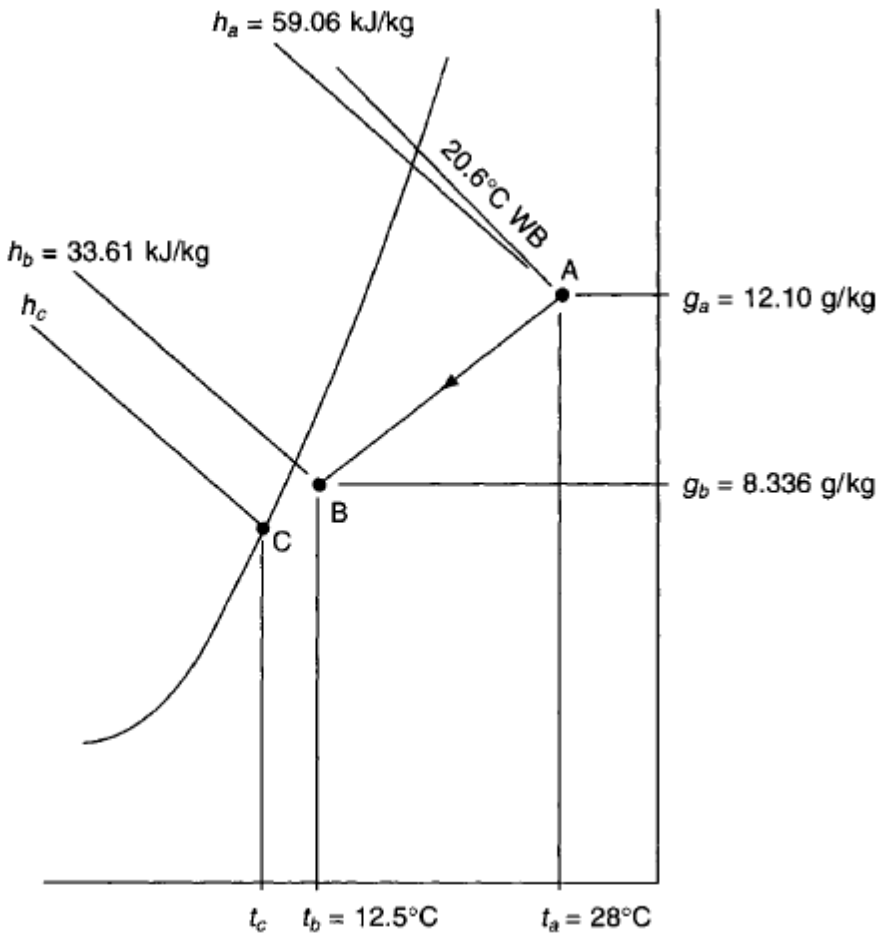


Fig. 3.5 The psychrometry for Example 3.5.

(*a*) Mark state points A and B on a psychrometric chart. Join them by a straight line and extend the line to cut the saturation curve. Observation of the point of intersection shows that this is at a temperature of 10.25°C. It is not easy to decide this value with any exactness unless a large psychrometric chart is used. However, it is sufficiently accurate for present purposes (and for most practical purposes) to take the value read from an ordinary chart. Hence the apparatus dew point is 10.25°C.

(*b*) Either from the chart or from tables, it can be established that the enthalpy at the apparatus dew point is 29.94 kJ kg^{-1}. Using the definition of contact factor,

$$\beta = \frac{59.06 - 33.61}{59.06 - 29.94} = 0.874$$

One might also determine this from the temperatures:

$$= \frac{28° - 12.5°}{28° - 10.25°} = 0.873$$

Clearly, in view of the error in reading the apparatus dew point from the chart, the value of β obtained from the temperatures is quite accurate enough in this example and, in fact, in most other cases.

(*c*) Cooling load = mass flow rate × decrease of enthalpy

$$= \frac{1.5}{0.8693} \times (59.06 - 33.61) = 43.9 \text{ kW}$$

3.5 Humidification

This, as its name implies, means that the moisture content of the air is increased. This may be accomplished by either water or steam but the present section is devoted to the use of water only, section 3.7 being used for steam injection into moving airstreams.

It is customary to speak of the humidifying efficiency or the effectiveness of an air washer (although neither term is universally accepted) rather than a contact or by-pass factor. There are several definitions, some based on the extent to which the dry-bulb temperature of the entering moist airstream approaches its initial wet-bulb value, and others based on the change of state undergone by the air. In view of the fact that the psychrometric chart currently in use by the Chartered Institution of Building Services

Engineers is constructed with mass (moisture content) and energy (enthalpy) as oblique, linear co-ordinates, the most suitable definition to use with the chart is that couched in terms of these fundamentals. There is the further advantage that such a definition of effectiveness, *E*, is identical with the definition of contact factor, β, used for cooler coils.

Although humidifying efficiency is often expressed in terms of a process of adiabatic saturation, this is really a special case, and is so regarded here.

Figure 3.6(*a*) shows an illustration of the change of state experienced by an airstream as it passes through a spray chamber.

The effectiveness of the spray chamber is then defined by

$$E = \frac{h_b - h_a}{h_c - h_a} \tag{3.5}$$

and humidifying efficiency is defined by

$$\eta = 100E \tag{3.6}$$

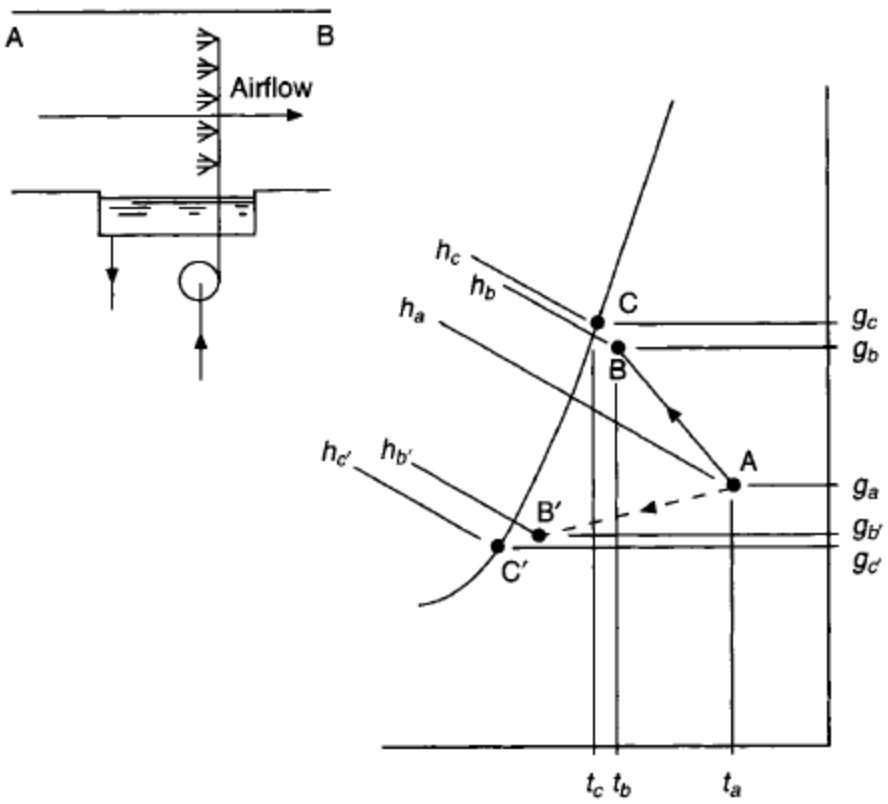
It is evident that, because moisture content is the other linear co-ordinate of the psychrometric chart and the points A, B and C lie on the same straight line, C being obtained by the extension of the line joining AB to cut the saturation curve, then it is possible to put forward an alternative and equally valid definition of effectiveness, expressed in terms of moisture content:

$$E = \frac{g_b - g_a}{g_c - g_a} \tag{3.7}$$

and

$\eta = 100E$, as before

Figure 3.6(*a*) shows that h_b is greater than h_a.



(*a*)

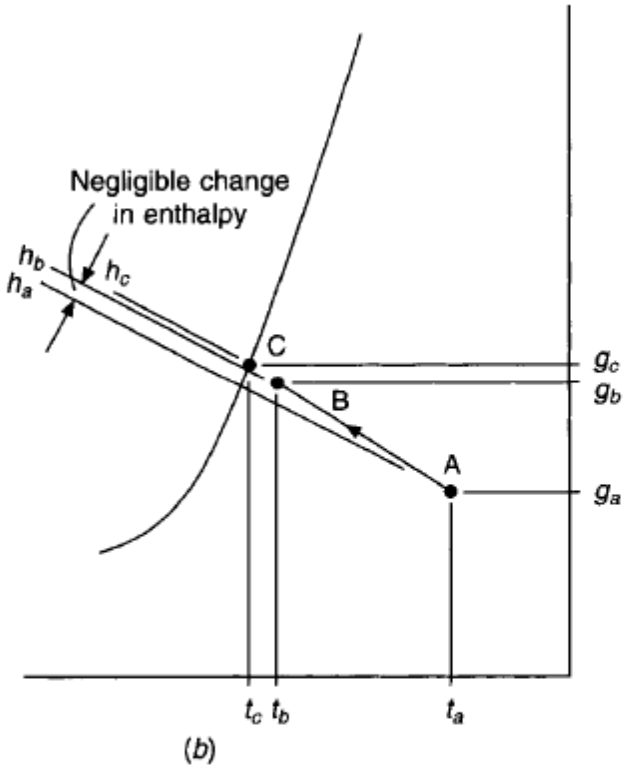


Fig. 3.6 (a) The change of state when air flows through a spray chamber having spray water at a controlled temperature. (b) Adiabatic saturation when the spray water is entirely recirculated and neither cooled nor heated.

If the spray water had been chilled, instead of heated, then a change of state might have been as shown dotted on Figure 3.6(*a*), from the point A to the point B′. Under these circumstances, the effectiveness would have been expressed by

$$E = \frac{h_a - h_{b'}}{h_a - h_{c'}}$$

or

$$E = \frac{g_a - g_{b'}}{g_a - g_{c'}}$$

Consider the special case of adiabatic saturation: for this to occur it is necessary that

(i) the spray water is totally recirculated, no heat exchanger being present in the pipelines or in the washer tank;
(ii) the spray chamber, tank and pipelines are perfectly lagged; and
(iii) the feed water supplied to the system to make good the losses due to evaporation is at the temperature of adiabatic saturation

Under these conditions it may be assumed that the change of state follows a line of constant wet-bulb temperature (since the Lewis number for air–water vapour mixtures is unity), the system having been given sufficient time to settle down to steady-state operation. There must be a change of enthalpy during the process because feed water is being supplied at the wet-bulb temperature (virtually) and this will not, as a general rule, equal the datum temperature for the enthalpy of water, 0°C. Strictly speaking then, it is incorrect to speak of the process as being an adiabatic one. The use of the term stems from the phrase 'temperature of adiabatic saturation' and so its use is condoned. One thing is fairly certain though: at the temperatures normally encountered the change of enthalpy during the process is neligible, and so effectiveness must be expressed in terms of change of moisture content. Figure 3.6(*b*) shows a case of adiabatic saturation, in which it can be seen that there is no significant alteration in enthalpy although there is a clear change in moisture content. It can also be seen that a fall in temperature accompanies the rise in moisture content. This temperature change provides an approximate definition of effectiveness or efficiency which is most useful, and, for the majority of practical applications, sufficiently accurate. It is

$$E \simeq \frac{t_a - t_b}{t_a - t_c} \tag{3.8}$$

The way in which the psychrometric chart is constructed precludes the possibility of this being an accurate expression; lines of constant dry-bulb temperature are not parallel and equally spaced, such properties being exclusive to enthalpy and moisture content.

EXAMPLE 3.6

1.5 $m^3 s^{-1}$ of moist air at a state of 15°C dry-bulb, 10°C wet-bulb (sling) and 101.325 kPa barometric pressure, enters the spray chamber of an air washer. The humidifying efficiency of the washer is 90 per cent, all the spray water is recirculated, the spray chamber and the tank are perfectly lagged, and mains water at 10°C is supplied to make good the losses due to evaporation.

Calculate (*a*) the state of the air leaving the washer, (*b*) the rate of flow of make-up water from the mains.

Answer

(*a*) This is illustrated in Figure 3.7.

Using the definitions of humidifying efficiency quoted in equations (3.6) and (3.7), and referring to tables of psychrometric data for the properties of moist air at states A and C, one can write

$$\frac{90}{100} = \frac{g_b - 5.558}{7.659 - 5.558}$$

Hence,

$$g_b = 7.449 \text{ g per kg of dry air}$$

The state of the moist air leaving the air washer is 10°C wet-bulb (sling), 7.449 g per kg and 101.325 kPa barometric pressure. The use of equation (3.8) shows that the approximate dry-bulb temperature at exit from the washer is 10.5°C.

(*b*) The amount of water supplied to the washer must equal the amount evaporated. From tables (or less accurately from a psychrometric chart), the humid volume at state A is 0.8232 m^3 kg^{-1}. Each kilogram of dry air passing through the spray chamber has its associated water vapour augmented by an amount equal to $g_b - g_a$, that is, by 1.891 g.

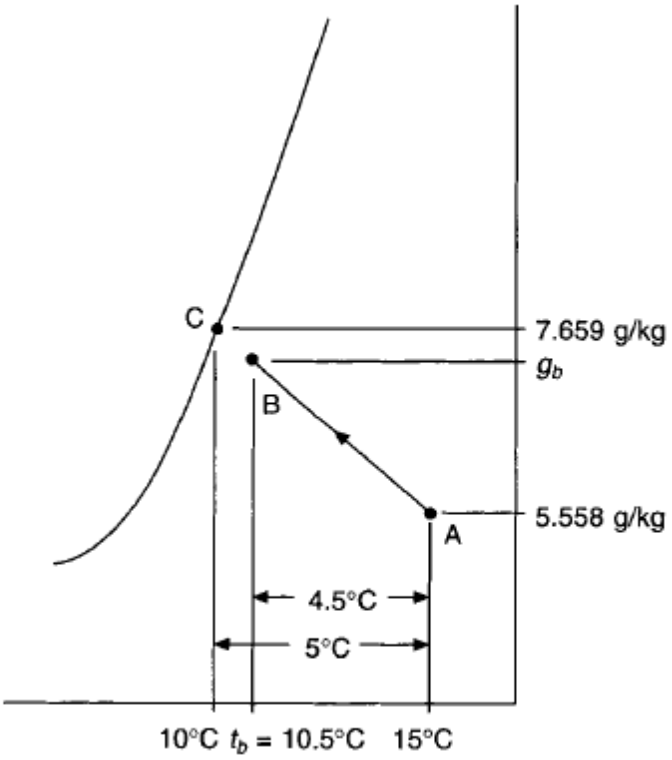


Fig. 3.7 Adiabatic saturation along a wet-bulb line.

Thus, the rate of make-up is

$$= \frac{1.5 \times 0.001\,891 \times 3600}{0.8232}$$

$$= 12.40 \text{ kg of water per hour}$$

3.6 Water injection

The simplest case to consider, and the one that provides the most insight into the change of state of the airstream subjected to humidification by the injection of water, is where all the injected water is evaporated. Figure 3.8 shows what happens when total evaporation occurs.

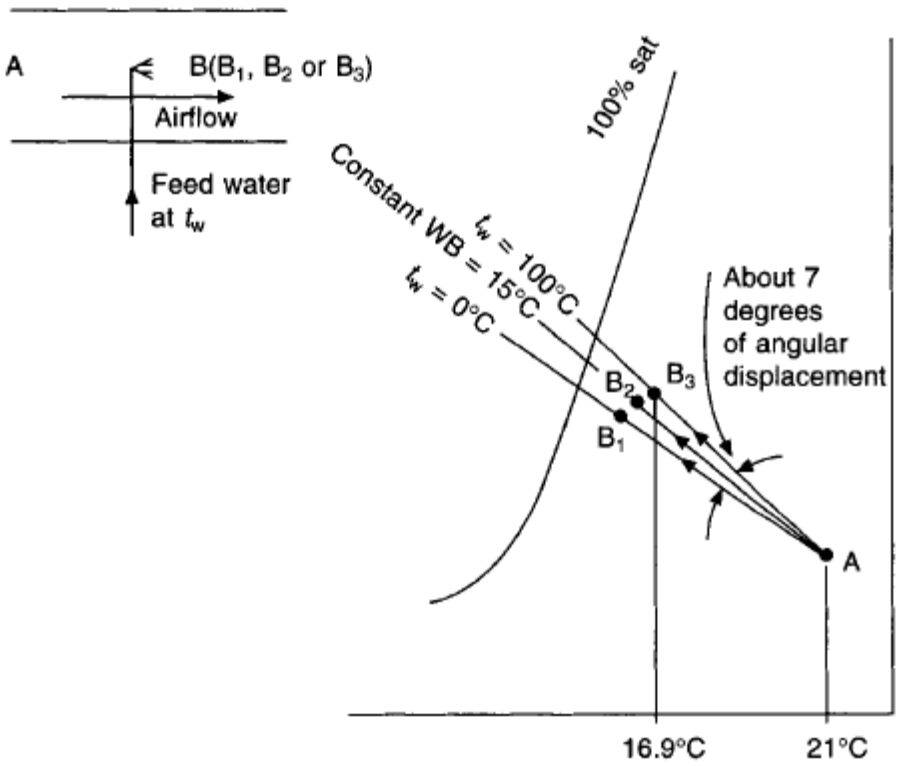


Fig. 3.8 A humidification process by the entire evaporation of injected spray water.

Air enters a spray chamber at state A and leaves it at state B, all the injected water being evaporated, none falling to the bottom of the chamber to run to waste or to be recirculated. The feed-water temperature is t_w. It is important to realise that since total evaporation has occurred, state B must lie nearer to the saturation curve, but just how much nearer will depend on the amount of water injected.

Two equations, a heat balance and a mass balance, provide the answer required.

Striking a heat balance, we can write

$$h_a + h_w = h_b$$

and, in a similar way, the mass balance may be written as

$$m_a + m_w = m_b$$

Knowing the amount of feed water evaporated, the mass balance gives the necessary information about the moisture content of the airstream leaving the area of the water injection. Applying the mass balance to the water vapour only (since the associated kilogram of dry air may be ignored),

$$g_b = g_a + m_w$$

where m_w is the amount of feed water evaporated in kg per kg of dry air flowing through the spray chamber.

Applying the heat balance:

$$h_a + h_w = h_b$$
$$= (1.007t_b - 0.026) + g_b(2501 + 1.84t_b) \qquad (2.24)$$

EXAMPLE 3.7

Moist air at a state of 21°C dry-bulb, 15°C wet-bulb (sling) and 101.325 kPa barometric pressure enters a spray chamber. If, for each kilogram of dry air passing through the chamber, 0.002 kg of water at 100°C is injected and totally evaporated, calculate the moisture content, enthalpy and dry-bulb temperature of the moist air leaving the chamber.

Answer

From CIBSE tables of psychrometric data,

$$h_a = 41.88 \text{ kJ per kg dry air}$$
$$g_a = 0.008\,171 \text{ kg per kg dry air}$$

Since the feed water has a temperature of 100°C, its enthalpy is 419.06 kJ per kg of water injected, from CIBSE tables of properties of water at saturation.

Use of the equation for mass balance yields the moisture content of the moist air leaving the spray chamber:

$$g_b = 0.008\,171 + 0.002$$
$$= 0.010\,171 \text{ kg per kg dry air}$$

Use of the energy balance equation gives the enthalpy of the air leaving the chamber and hence, also, its dry-bulb temperature by equation (2.24)

$$h_b = 41.88 + 0.002 \times 418.06$$
$$= 42.716 \text{ kJ per kg dry air}$$
$$= (1.007t_b - 0.026) + 0.010\,171(2501 + 1.84t_b)$$

thus,

$$42.716 = 1.007t_b - 0.026 + 25.44 + 0.0187t_b$$
$$t_b = 16.9°C$$

3.7 Steam injection

As in water injection, steam injection may be dealt with by a consideration of a mass and energy balance. If m_s kg of dry saturated steam are injected into a moving airsteam of mass flow 1 kg of dry air per second, then we may write

$$g_b = g_a + m_s$$

and

$$h_b = h_a + h_s$$

If the initial state of the moist airstream and the condition of the steam is known, then the

final state of the air may be determined, provided none of the steam is condensed. The change of state takes place almost along a line of constant dry-bulb temperature between limits defined by the smallest and largest enthalpies of the injected steam, provided the steam is in a dry, saturated condition. If the steam is superheated then, of course, the dry-bulb temperature of the airstream may increase by any amount, depending on the degree of superheat. The two limiting cases for dry saturated steam are easily considered. The lowest possible enthalpy is for dry saturated steam at 100°C. It is not possible to use steam at a lower temperature than this since the steam must be at a higher pressure than atmospheric if it is to issue from the nozzles. (Note, however, that steam could be generated at a lower temperature from a bath of warm water, which was not actually boiling.)

The other extreme is provided by the steam which has maximum enthalpy; the value of this is 2803 kJ kg^{-1} of steam and it exists at a pressure of about 30 bar and a temperature of about 234°C.

What angular displacement occurs between these two limits, and how are they related to a line of constant dry-bulb temperature? These questions are answered by means of two numerical examples.

EXAMPLE 3.8

Dry saturated steam at 100°C is injected at a rate of 0.01 kg s^{-1} into a moist airstream moving at a rate of 1 kg of dry air per second and initially at a state of 28°C dry-bulb, 11.9°C wet-bulb (sling) and 101.325 kPa barometric pressure. Calculate the leaving state of the moist airstream.

Answer

From psychrometric tables, h_a = 33.11 kJ per kg dry air,

$$g_a = 0.001\,937 \text{ kg per kg}$$

From NEL steam tables, h_s = 2675.8 kJ per kg steam.

$$g_b = 0.001\,937 + 0.01$$
$$= 0.011\,937 \text{ kg per kg dry air, by the mass balance}$$
$$h_b = 33.11 + 0.01 \times 2675.8$$
$$= 59.87 \text{ kJ per kg dry air}$$

Hence,

$$59.87 = (1.007t_b - 0.026) + 0.011937(2501 + 1.84t_b)$$
$$t_b = 29.2°C$$

EXAMPLE 3.9

Dry saturated steam with maximum enthalpy is injected at a rate of 0.01 kg s^{-1} into a moist airstream moving at a rate of 1 kg of dry air per second and initially at a state of 28°C dry-bulb, 11.9°C wet-bulb (sling) and 101.325 kPa barometric pressure. Calculate the leaving state of the moist airstream.

Answer

The psychrometric properties of state A are as for the last example. The moisture content at state B is also as in example 3.8.

From NEL steam tables, h_s = 2803 kJ per kg of steam, at 30 bar and 234°C saturated. Consequently,

$$h_b = 33.11 + 0.01 \times 2803$$
$$= 61.14 \text{ kJ kg}^{-1}$$
$$g_b = 0.001\,937 + 0.01$$
$$= 0.011\,937 \text{ kg kg}^{-1} \text{ dry air}$$

Hence, as before:

$$61.14 = (1.007t_b - 0.026) + 0.011\,937(2501 + 1.84t_b)$$
$$t_b = 30.4°\text{C}$$

Figure 3.9 illustrates what occurs. It can be seen that for the range of states considered, the change in dry-bulb value is not very great. In fact, the angular displacement between the two condition lines for the last two examples of steam injection is only about 3 or 4 degrees. We can conclude that, although there is an increase in temperature, it is within the accuracy usually required in practical air conditioning to assume that the change of state following steam injection is up a line of constant dry-bulb temperature.

In the injection of superheated steam, every case should be treated on its merits, as the equation for the dry-bulb temperature resulting from a steam injection process shows:

$$t_b = \frac{h_b + 0.026 - 2501g}{1.007 + 1.84g} \tag{3.9}$$

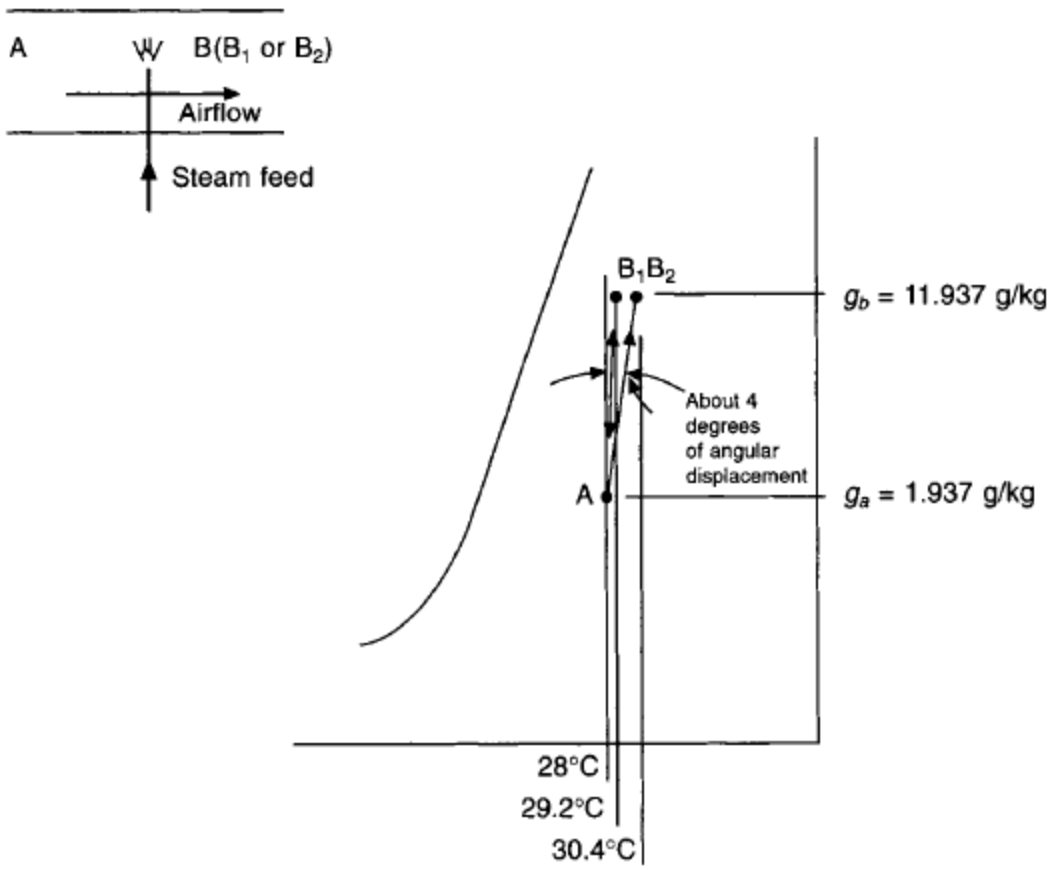


Fig. 3.9 A humidification process by the entire acceptance of injected dry saturated steam. See Examples 3.8 and 3.9.

3.8 Cooling and dehumidification with reheat

Figure 3.10(*a*) shows, in diagrammatic form, the sort of plant required. Moist air at a state A passes over the finned tubes of a cooler coil through which chilled water is flowing. The amount of dehumidification carried out is controlled by a dew point thermostat, C1, positioned after the coil. This thermostat regulates the amount of chilled water flowing through the coil by means of the three-way mixing valve R1. Air leaves the coil at state B, with a moisture content suitable for the proper removal of the latent heat gains occurring in the room being conditioned. The moisture content has been reduced from g_a to g_b and the cooler coil has a mean surface temperature of t_c, Figure 3.10(*b*) illustrating the psychrometric processes involved.

The cooling load is proportional to the difference of enthalpy between h_a and h_b, and the load on the heater battery is proportional to h_d minus h_b. It follows from this that part of the cooling load is being wasted by the reheat. This is unavoidable in the simple system illustrated, and is a consequence of the need to dehumidify first, and heat afterwards. It should be observed, however, that in general it is undesirable for such a situation to exist during maximum load conditions. Reheat is usually only permitted to waste cooling capacity under partial load conditions, that is, the design should be such that state B can adequately deal with both maximum sensible and maximum latent loads. These points are illustrated in the following example.

EXAMPLE 3.10

Moist air at 28°C dry-bulb, 20.6°C wet-bulb (sling) and 101.325 kPa barometric pressure flows over a cooler coil and leaves it at a state of 10°C dry-bulb and 7.046 g per kg of dry air.

(*a*) If the air is required to offset a sensible heat gain of 2.35 kW and a latent heat gain of 0.31 kW in a space to be air-conditioned, calculate the mass of dry air which must be supplied to the room in order to maintain a dry-bulb temperature of 21°C therein.

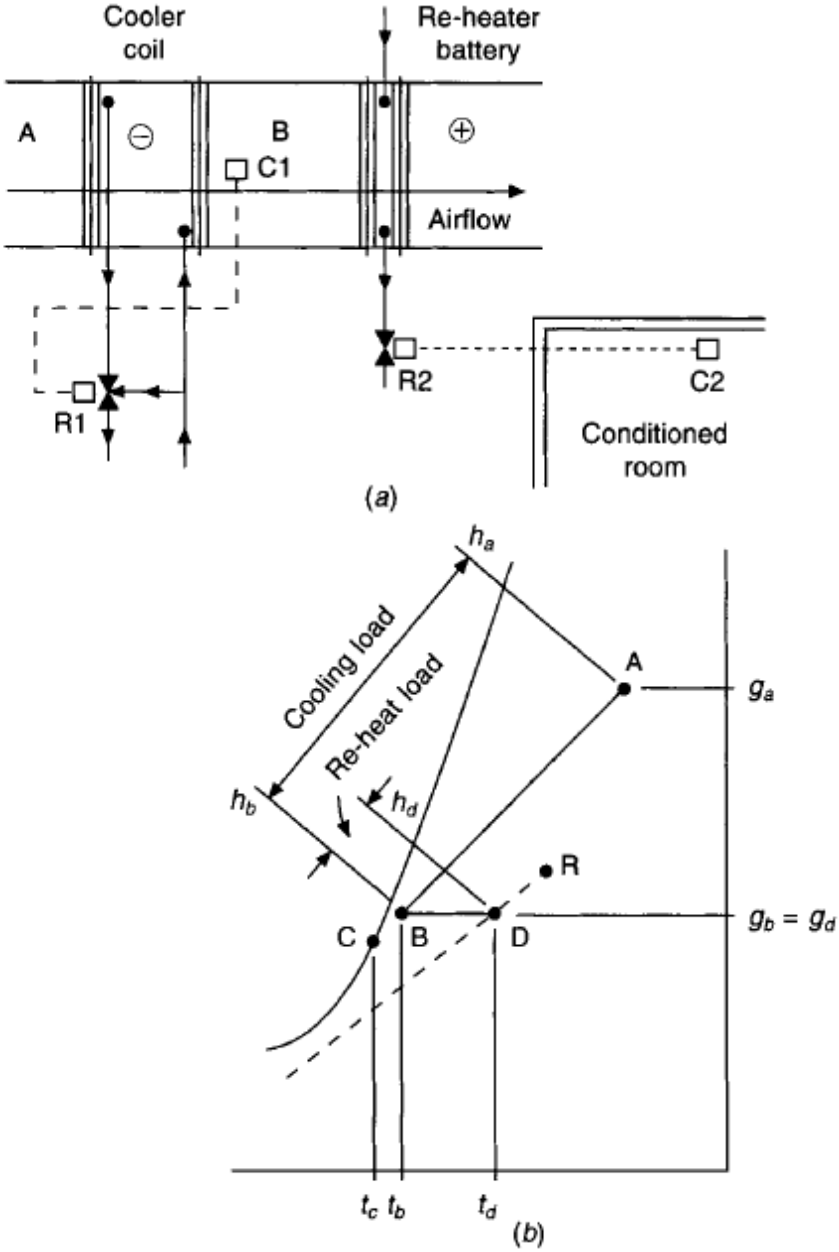


Fig. 3.10 (a) Plant arrangement for cooling and dehumidification with reheat. (b) Psychrometry for cooling and dehumidification with reheat.

(*b*) What will be the relative humidity in the room?

(*c*) If the sensible heat gain diminishes by 1.175 kW but the latent heat gain remains unchanged, at what temperature and moisture content must the air be supplied to the room?

Answer

(*a*) If m_a kg per s of dry air are supplied at a temperature of 10°C, they must have a sensible cooling capacity equal to the sensible heat gain, if 21°C is to be maintained in the room. Thus assuming the specific heat capacity of air is 1.012 kJ kg^{-1} K^{-1}

$$m_a \times c_a \times (21° - 10°) = 2.35 \text{ kW}$$

then,

$$m_a = \frac{2.35}{1.012 \times 11°} = 0.211 \text{ kg s}^{-1}$$

(*b*) 0.211 kg s^{-1} of dry air with an associated moisture content of 7.046 g per kg of dry air must take up the moisture evaporated by the liberation of 0.31 kW of latent heat. Assuming a latent heat of evaporation of, say, 2454 kJ per kg of water (at about 21°C), then the latent heat gain corresponds to the evaporation of

$$\frac{0.31 \text{ kJ s}^{-1}}{2454 \text{ kJ kg}^{-1}} = 0.000\,126\,3 \text{ kg of water per second}$$

The moisture associated with the delivery to the room of 0.211 kg s^{-1} of dry air will increase by this amount. The moisture picked up by each kg of dry air supplied to the room will be

$$\frac{0.1263}{0.211} = 0.599 \text{ g kg}^{-1}$$

Thus, the moisture content in the room will be equal to 7.046 + 0.599, that is, 7.645 g kg^{-1}. The relative humidity at this moisture content and 21°C dry-bulb is found from tables of psychrometric data to be 49.3 per cent. If the relevant data are taken from a psychrometric chart, instead of from psychrometric tables, then similar but slightly different results are obtained, of adequate practical accuracy in most cases. The percentage saturation at 21°C dry-bulb is then just under 49 per cent.

(*c*) If 0.211 kg s^{-1} of dry air is required to absorb only 1.175 kW of sensible heat then, if 21°C is still to be maintained, the air must be supplied at a higher temperature:

$$\text{temperature rise} = \frac{1.175}{1.012 \times 0.211} = 5.5 \text{ K}$$

Thus, the temperature of the supply air is 15.5°C.

Since the latent heat gains are unrelated, the air supplied to the room must have the same ability to offset these gains as before; that is, the moisture content of the air supplied must still be 7.046 g per kg of dry air.

EXAMPLE 3.11

(*a*) For sensible and latent heat gains of 2.35 and 0.31 kW, respectively, calculate the load on the cooler coil in example 3.10.

(*b*) Calculate the cooling load and the reheater load for the case of 1.175 kW sensible heat gains and unchanged latent gains.

Answer

(*a*) The load on the cooler coil equals the product of the mass flow of moist air over the coil and the enthalpy drop suffered by the air. Thus, as an equation:

$$\text{cooling load} = m_a \times (h_a - h_b)$$

The notation adopted is the same as that used in Figure 3.10. In terms of units, the equation can be written as

$$\text{kW} = \frac{\text{kg of dry air}}{\text{s}} \times \frac{\text{kJ}}{\text{kg of dry air}}$$

Using enthalpy values obtained from tables for the states A and B, the equation becomes

$$\begin{aligned}\text{cooling load} &= 0.211 \times (59.06 - 27.81)\\ &= 0.211 \times 31.25\\ &= 6.59 \text{ kW}\end{aligned}$$

(*b*) Since it is stipulated that the latent heat gains are unchanged, air must be supplied to the conditioned room at the same state of moisture content as before. That is, the cooler coil must exercise its full dehumidifying function, the state of the air leaving the coil being the same as in example 3.11(*a*), above. Hence the cooling load is still 6.59 kW. To deal with the diminished sensible heat gains it is necessary to supply the air to the room at a temperature of 15.5°C, as was seen in example 3.10(*c*). The air must, therefore, have its temperature raised by a reheater battery from 10°C to 15.5°C. Since the moisture content is 7.046 g kg^{-1} at both these temperatures, we can find the corresponding enthalpies at states B and D directly from a psychrometric chart or by interpolation (in the case of state D) from psychrometric tables.

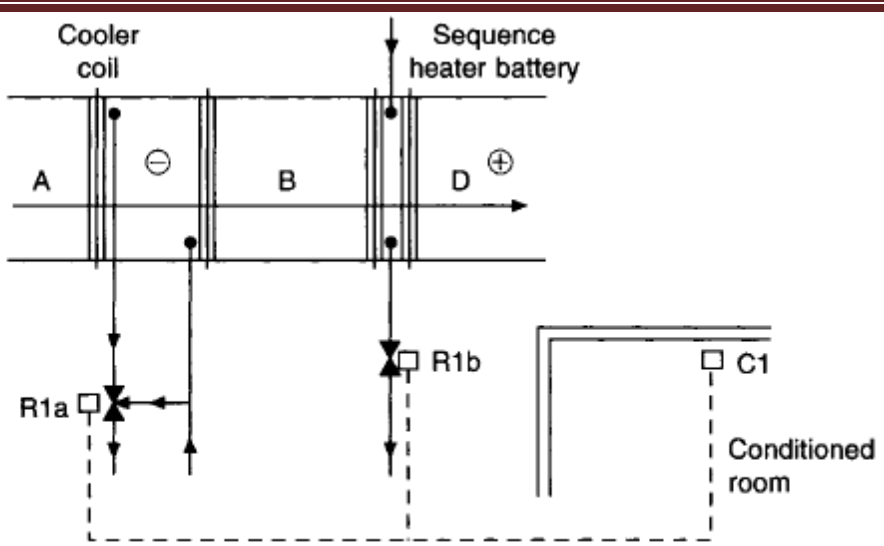
Hence, we can form an equation for the load on the reheater battery:

$$\text{reheater load} = 0.211 \times (33.41 - 27.81) = 1.18 \text{ kW}$$

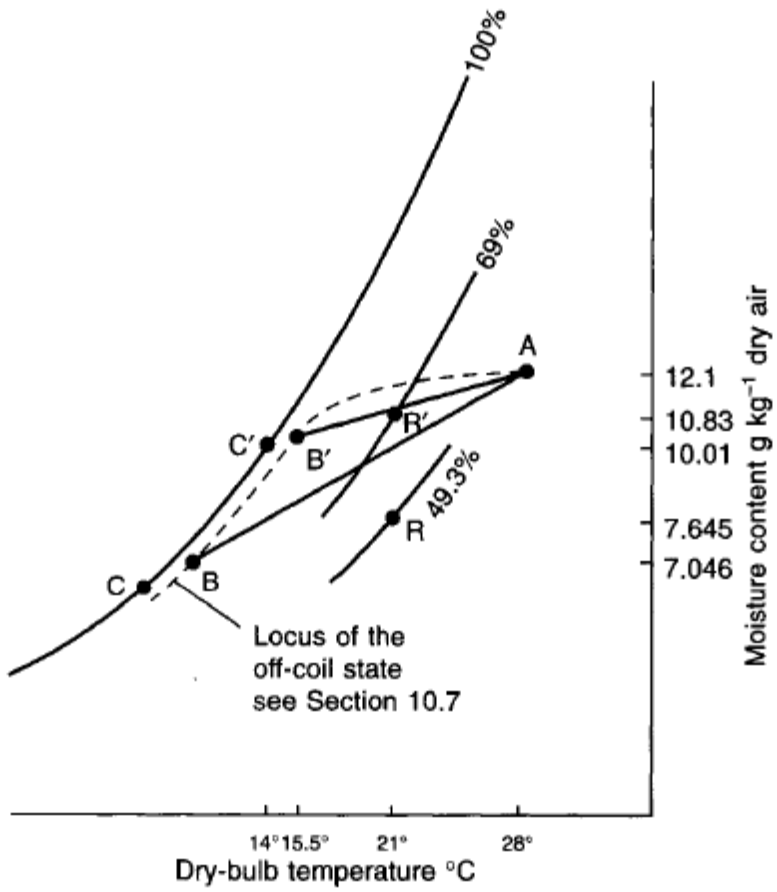
EXAMPLE 3.12

If the plant in example 3.10 is arranged for sequence control over the cooler coil and heater battery, and if under the design conditions mentioned in that example the plant is able to maintain 21°C dry-bulb and 49.3 per cent relative humidity in the conditioned space, calculate the relative humidity that will be maintained there, under sequence control, if the sensible heat gains diminish to 1.175 kW, the latent gains remaining unaltered.

It is given that the mean coil surface temperature under the condition of partial load is 14°C. (See section 10.7 on the partial load performance of cooler coils.)



(a)



(b)

Fig. 3.11 (a) Plant arrangement for heating and cooling in sequence. (b) Psychrometry for heating and cooling in sequence, for Example 3.12.

Answer

As was mentioned in section 3.4, the mean coil surface temperature (identified by the state point C), lies on the straight line joining the cooler coil 'on' and 'off' states (points A and B) where it cuts the saturation curve. Figure 3.11(*b*) illustrates this, and also shows that the moisture content of state B′ is greater than that of B. It is possible to calculate the moisture content of the supply air by assuming that the dry-bulb scale on the psychrometric chart is linear. By proportion, we can then assess a reasonably accurate value for the moisture content of the air leaving the cooler coil under the partial load condition stipulated in this example:

$$\frac{g_a - g_{b'}}{g_a - g_{c'}} = \frac{t_a - t_{b'}}{t_a - t_{c'}}$$

From tables, or from a psychrometric chart, $g_a = 12.10$ g kg^{-1} and $g_{c'} = 10.01$ g kg^{-1}. Also, it was calculated in example 3.10(*b*) that the supply temperature must be 15.5°C dry-bulb for the partial sensible load condition.

Hence we can write

$$g_{b'} = g_a - (g_a - g_{c'}) \times \frac{(t_a - t_{b'})}{(t_a - t_{c'})}$$

$$= 12.10 - (12.10 - 10.01) \times \frac{(28 - 15.5)}{(28 - 14)} = 10.23 \text{ g kg}^{-1}$$

Since the moisture pick-up has been previously calculated as 0.599 g kg^{-1}, the moisture content in the conditioned room will be 10.83 g kg^{-1}; at 21°C dry-bulb the relative humidity will be about 69 per cent.

3.9 Pre-heat and humidification with reheat

Air conditioning plants which handle fresh air only may be faced in winter with the task of increasing both the moisture content and the temperature of the air they supply to the conditioned space. Humidification is needed because the outside air in winter has a low moisture content, and if this air were to be introduced directly to the room there would be a correspondingly low moisture content there as well. The low moisture content may not be intrinsically objectionable, but when the air is heated to a higher temperature its relative humidity may become very low. For example, outside air in winter might be at –1°C, saturated (see chapter 5). The moisture content at this state is only 3.484 g per kg of dry air. If this is heated to 20°C dry-bulb, and if there is a moisture pick-up in the room of 0.6 g per kg of dry air, due to latent heat gains, then the relative humidity in the room will be as low as 28 per cent. This value may sometimes be seen as too low for comfort. The plant must increase the temperature of the air, either to the value of the room temperature if there is background heating to offset fabric losses, or to a value in excees of this if it is intended that the air delivered should deal with fabric losses.

The processes whereby the moisture content of air may be increased were discussed in section 3.5. The actual method chosen depends on the application but, for comfort air

conditioning or for any application where people are present in the conditioned space, the use of an air washer or any method involving the recirculation of spray water, or the exposure of a wetted surface area to the airstream, is not recommended. This is because of the risk to health caused by the presence of micro-organisms in the water. The use of dry steam injection is much preferred.

As an exercise in psychrometry, it is worth considering the case where an air washer is used. If 100 per cent fresh air is handled in cold weather it is pre-heated, passed through an air washer where it undergoes adiabatic saturation, and reheated to the temperature at which it must be supplied to the room. Pre-heating and adiabatic saturation permit the relative humidity in the room to be controlled, and reheating allows the temperature therein to be properly regulated, in winter.

Figure 3.12(*a*) shows, in a diagrammatic form, a typical plant. Opening the modulating valve R1 in the return pipeline from the pre-heater increases the heating output of the battery and provides the necessary extra energy for the evaporation of more water in the washer, if an increase in the moisture content of the supply air is required. Similarly, opening the control value R2, associated with the reheater, allows air at a higher temperature to be delivered to the room being conditioned. C1 and C2 are a room humidistat and a room thermostat, respectively.

EXAMPLE 3.13

The plant shown in Figure 3.12(*a*) operates as illustrated by the psychrometric changes in Figure 3.12(*b*).

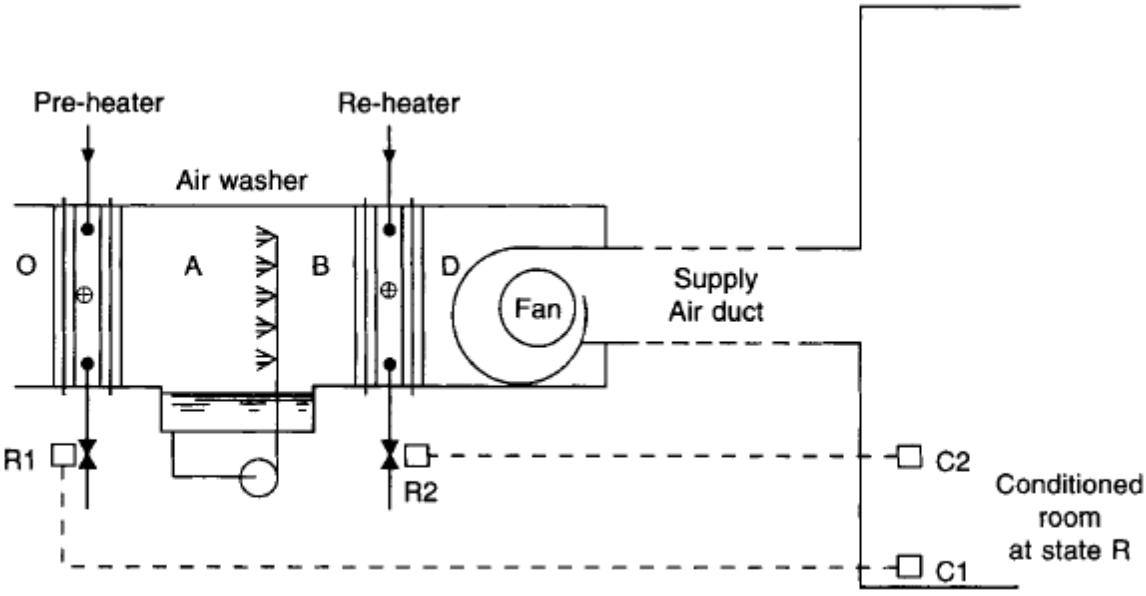
Air is pre-heated from –5.0°C dry-bulb and 86 per cent saturation to 23°C dry-bulb. It is then passed through an air washer having a humidifying efficiency of 85 per cent and using recirculated spray water. Calculate the following:

(*a*) The relative humidity of the air leaving the washer
(*b*) The cold water make-up to the washer in litres s^{-1}, given that the airflow rate leaving the washer is 2.5 m^3 s^{-1}
(*c*) The duty of the pre-heater battery in kW
(*d*) The temperature of the air supplied to the conditioned space if the sensible heat losses from it are 24 kW and 20°C dry-bulb is maintained there
(*e*) The duty of the reheater battery
(*f*) The relative humidity maintained in the room if the latent heat gains therein are 5 kW

Answer

(*a*) At state O the moisture content is found from tables to be 2.137 g kg^{-1}. Consequently, at 23°C dry-bulb and 2.137 g kg^{-1} the wet-bulb (sling) value is found from tables to be 10°C. This state is represented by the point A in the diagram. Assuming that the process of adiabatic saturation occurs up a wet-bulb line, we can establish from tables that the moisture content at state C, the apparatus dew point, is 7.659 g kg^{-1}. If the further assumption is made that the dry-bulb scale is linear, we can evaluate the dry-bulb temperature at state B, leaving the washer, by proportion:

$$t_b = 23 - 0.85(23 - 10)$$
$$= 23 - 11.05$$
$$= 12°C \text{ dry-bulb}$$



(*a*)

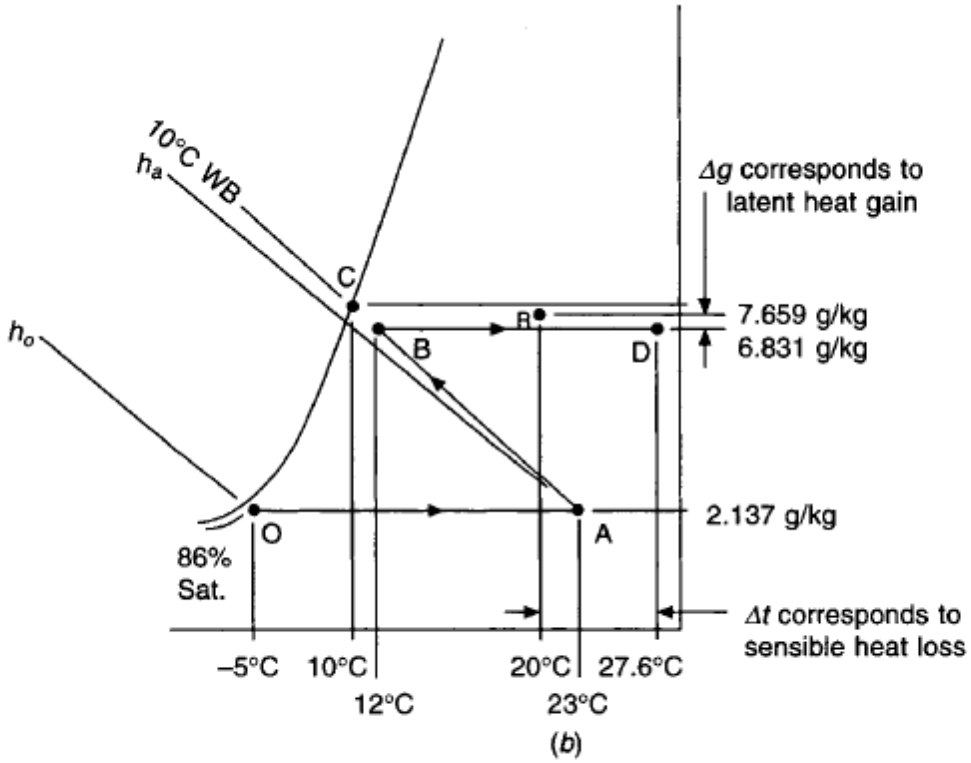


Fig. 3.12 (a) Plant arrangement for pre-heating 100% fresh air with adiabatic saturation and reheat. (b) The psychrometry for Example 3.13.

We know that the dry-bulb temperature at the point C is 10°C because the air is saturated at C and it has a wet-bulb temperature of 10°C.

The above answer for the dry-bulb at B is approximate (because the dry-bulb scale is not linear) but accurate enough for all practical purposes. On the other hand, we may calculate the moisture content at B with more exactness, since the definition of humidifying efficiency is in terms of moisture content changes and this scale is linear on the psychrometric chart.

$$\begin{aligned} g_b &= 7.659 - (1 - 0.85) \times (7.659 - 2.137) \\ &= 7.659 - 0.828 \\ &= 6.831 \text{ g kg}^{-1} \end{aligned}$$

We can now refer to tables or to a chart and determine that the relative humidity at 12°C dry-bulb and 6.831 g kg^{-1} is 78 per cent. Percentage saturation is virtually the same as relative humidity. More accurately, we might perhaps determine the humidity at a state of 10°C wet-bulb (sling) and 6.831 g kg^{-1}. The yield in accuracy is of doubtful value and the method is rather tedious when using tables.

(*b*) The cold water make-up depends on the mass flow of dry air. It may be determined that the humid volume of the air leaving the washer at state B is 0.8162 m^3 kg^{-1} of dry air. The cold water fed from the mains to the washer serves to make good the losses due to evaporation within the washer. Since the evaporation rate is (6.831 – 2.137) g kg^{-1}, we may calculate the make-up rate:

$$\text{make-up rate} = \frac{(6.831 - 2.137) \times 2.5}{1000 \times 0.8162}$$
$$= 0.0144 \text{ kg s}^{-1} \text{ or litres s}^{-1}$$

(*c*) The pre-heater must increase the enthalpy of the air passing over it from h_o to h_a. Reference to tables establishes that $h_o = 0.298$ kJ kg^{-1} and $h_a = 28.57$ kJ kg^{-1}. Then,

$$\text{pre-heater duty} = \frac{2.5 \times (28.57 - 0.298)}{0.8162}$$
$$= 3.063 \text{ kg s}^{-1} \times 28.27 \text{ kJ s}^{-1}$$
$$= 86.59 \text{ kW}$$

(*d*) 3.063 kg s^{-1} of air diffuses throughout the room and its temperature falls from t_d to 20°C as it offsets the heat loss. Assuming the specific heat capacity of water vapour is 1.89 kJ kg^{-1} K^{-1} and that of dry air 1.012 kJ kg^{-1} K^{-1}, a heat balance equation can be written:

Sensible heat loss = (mass flow rate of dry air × specific heat of dry air + associated moisture × specific heat of water vapour) × (supply air temperature – room temperature)

$$24 = 3.063 \times (1.012 + 0.006\,831 \times 1.89) \times (t_d - 20)$$
$$= 3.063 \times 1.025 \times (t_d - 20)$$

whence $t_d = 27.6°\text{C}$

The term involving the specific heat of dry air plus the moisture content times the specific heat of water vapour, is termed the specific heat of humid air or the humid specific heat. In this case its value is 1.025 kJ kg^{-1} K^{-1}.

(*e*) The reheater battery thus has to heat the humid air from the state at which it leaves the washer to the state at which it enters the room, namely from 12°C dry-bulb and 6.831 g kg^{-1} to 27.6° dry-bulb and 6.831 g kg^{-1}.

$$\text{Reheater duty} = 3.063 \times 1.025 \times (27.6 - 12)$$
$$= 49 \text{ kW}$$

Alternatively, we can determine the enthalpies of the states on and off the heater, by interpolating from tables or reading directly from a psychrometric chart:

$$\text{reheater duty} = 3.063 \times (45.2 - 29.3)$$
$$= 48.7 \text{ kW}$$

(*f*) A mass balance must be struck to determine the rise in moisture content in the room as a consequence of the evaporation corresponding to the liberation of the latent heat gains:

latent heat gain in kW = (kg of dry air per hour delivered to the room)
× (the moisture pick-up in kg of water per kg of dry air)
× (the latent heat of evaporation of water in kJ per kg of water)

$$5.0 = 3.063 \times (g_r - 0.006\,831) \times 2454$$

whence $g_r = 0.007\,496$ kg per kg dry air

From tables or from a chart it may be found that at a state of 20°C dry-bulb and 7.496 g per kg dry air, the relative humidity is about 51 per cent and, for use in Example 3.14(*b*), $h_r = 39.14$ kJ kg^{-1}.

3.10 Mixing and adiabatic saturation with reheat

Figure 3.13(*a*) shows a plant arrangement including an air washer which, although undesirable for comfort air conditioning, is retained here to illustrate the psychrometry involved. Air at state R is extracted from a conditioned room and partly recirculated, the remainder being discharged to atmosphere. The portion of the extracted air returned to the air conditioning plant mixes with air at state O, drawn from outside, and forms a mixture state M. The air then passes through an air washer, the spray water of which is only recirculated and adiabatic saturation occurs, the state of the air changing from M to W (see Figure 3.13(*b*)) along a line of constant wet-bulb temperature (see sections 3.5 and 3.6). An extension of the line MW cuts the saturation curve at a point A, the apparatus dew point. To deal with a particular latent heat gain in the conditioned room it is necessary to supply the air to the room at a moisture content g_s, it being arranged that the difference of moisture content $g_r - g_s$, in conjunction with the mass of air delivered to the room, will offset the latent gain. In other words, the air supplied must be dry enough to absorb the moisture liberated in the room.

It is evident that the moisture content of the air leaving the washer must have a value g_w, equal to the required value, g_s. This is amenable to calculation by making use of the definition of the effectiveness of an air washer, in terms of g_a, g_w and g_m (see section 3.5).

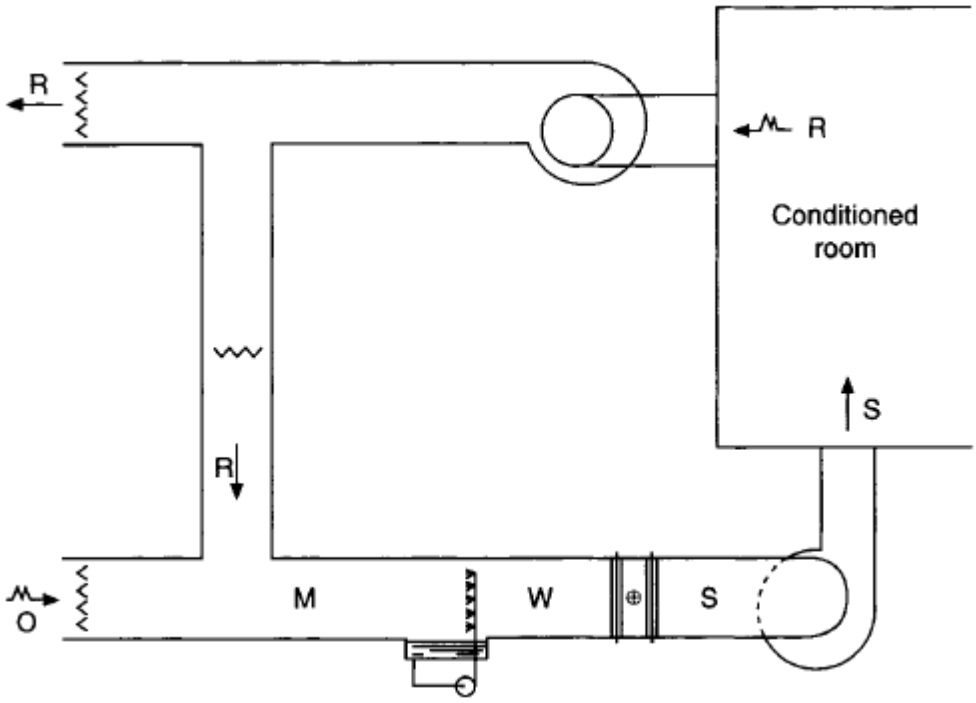
EXAMPLE 3.14

If the room mentioned in example 3.13 is conditioned by means of a plant using a mixture of recirculated and fresh air, of the type illustrated in Figure 3.13(*a*), calculate:

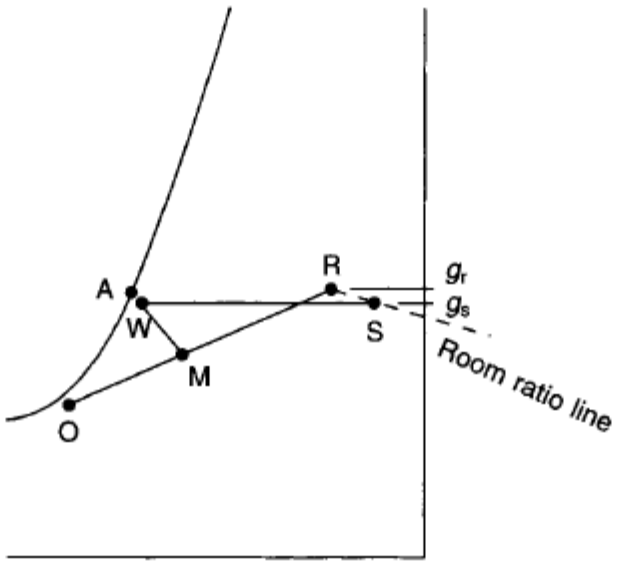
(*a*) the percentage of the air supplied to the room by mass which is recirculated, and
(*b*) the humidifying efficiency of the air washer.

Answer

(*a*) Since the wet-bulb scale is not linear, it is not accurate enough to calculate the mixing proportions on this basis. Instead, one must make use of changes of enthalpy or moisture content. Bearing in mind that lines of constant enthalpy are not parallel to lines of constant wet-bulb temperature, some slight inaccuracy is still present if the assumption is made that the change of state accompanying a process of adiabatic saturation is along a line of constant enthalpy. However, this is unavoidable, and so such an assumption is made.



(*a*)



(*b*)

Fig. 3.13 (a) Plant arrangement to permit variable mixing proportions of fresh and recirculated air. (b) Psychrometry for example 3.14.

Referring to Figure 3.13(*b*) it can be seen that

$$h_a \simeq h_w \simeq h_m$$

From tables and Figure 3.12(*b*) it is established that h_w (at 12°C dry-bulb and 6.831 g kg^{-1}) is 29.30 kJ kg^{-1}.

From the principles set out in section 3.2, governing the change of state associated with a mixing process, it is clear that the percentage of recirculated air, by mass,

$$= \frac{h_m - h_o}{h_r - h_o} \times 100$$

$$= \frac{29.30 - 0.298}{39.14 - 0.298} \times 100$$

$$= 75 \text{ per cent}$$

Thus, 75 per cent of the air supplied to the room, if recirculated and mixed with 25 per cent of air from outside, will have an enthalpy of 29.30 kJ kg^{-1} and a wet-bulb of 10°C (sling). If adiabatic saturation is then to produce a state of 12°C dry-bulb and 6.831 g kg^{-1}, the humidifying efficiency of the washer used can no longer be the value used for example 3.13, namely, 85 per cent. An entirely different washer must be used for the above calculations to be valid and this must have an effectiveness which may be calculated as follows:

(*b*) Since efficiency is expressed in terms of moisture content, it is necessary to determine the value of g_m, the values of g_a and g_w being already known.

$$g_m = 0.75g_r + 0.25g_o$$
$$= 0.75 \times 7.497 + 0.25 \times 2.137$$
$$= 6.157 \text{ g kg}^{-1}$$

$$\text{Humidifying efficiency} = \frac{g_w - g_m}{g_a - g_m} \times 100$$
$$= \frac{6.831 - 6.157}{7.659 - 6.157} \times 100$$
$$= 45 \text{ per cent}$$

In practical terms, this is a low efficiency.

If the washer used in this example had an efficiency of 85 per cent, as in example 3.13, then the calculations would not have been so easy. The line AWM would have had to have been at a lower wet-bulb value in order to fulfil two requirements:

(i) $g_w = 6.831$ g kg^{-1}

(ii) $\dfrac{g_w - g_m}{g_a - g_m} \times 100 = 85$ per cent

For this to be the case, the dry-bulb temperature of W must obviously be less than 10°C. The easiest way to achieve a practical solution is by drawing a succession of lines representing processes of adiabatic saturation on a psychrometric chart and calculating several values of efficiency until one of acceptable accuracy is obtained.

3.11 The use of dry steam for humidification

It is only if the air can accept the additional moisture, that humidification may be achieved by the injection of spray water or dry steam. If the state of the air into which it is proposed to inject moisture is saturated, or close to saturation, some or all of the moisture added will not be accepted and will be deposited downstream in the air handling plant or the duct system. It is therefore essential to ensure that the airstream is sufficiently heated prior to moisture injection.

EXAMPLE 3.15

A room is to be maintained at a state of 20°C dry-bulb and 50 per cent saturation by a plant handling 0.5 $m^3\ s^{-1}$ of outside air at a state of –2°C saturated. The airstream is heated to a temperature warm enough to offset a heat loss of 2.5 kW and dry steam is then injected to maintain the humidity required in the room. Calculate the supply air temperature and the heating and humidification loads. See Figure 3.14.

Answer

From psychrometric tables or a psychrometric chart, air at the outside state has a moisture content of 3.205 g kg^{-1}, an enthalpy of 5.992 kJ kg^{-1} and a specific volume of 0.7716 m^3 kg^{-1}. Assuming specific heats of 1.012 and 1.89 kJ kg^{-1} K^{-1}, respectively, for dry air and water vapour, the humid specific heat of the fresh air handled is

$$1 \times 1.012 + 0.003\,205 \times 1.89 = 1.018 \text{ kJ kg}^{-1}\text{ K}^{-1}$$

The heat loss is offset by the supply of air at a temperature t_s, warmer than the room temperature of 20°C. Hence

$$2.5 = (0.5/0.7716) \times 1.018 \times (t_s - 20)$$

whence t_s = 23.8°C.

From tables or a chart the enthalpy at state B, leaving the heater battery, is 32.10 kJ kg^{-1}. See Figure 3.14. Hence the heater battery load is

$$(0.5/0.7716) \times (32.10 - 5.992) = 16.92 \text{ kW}$$

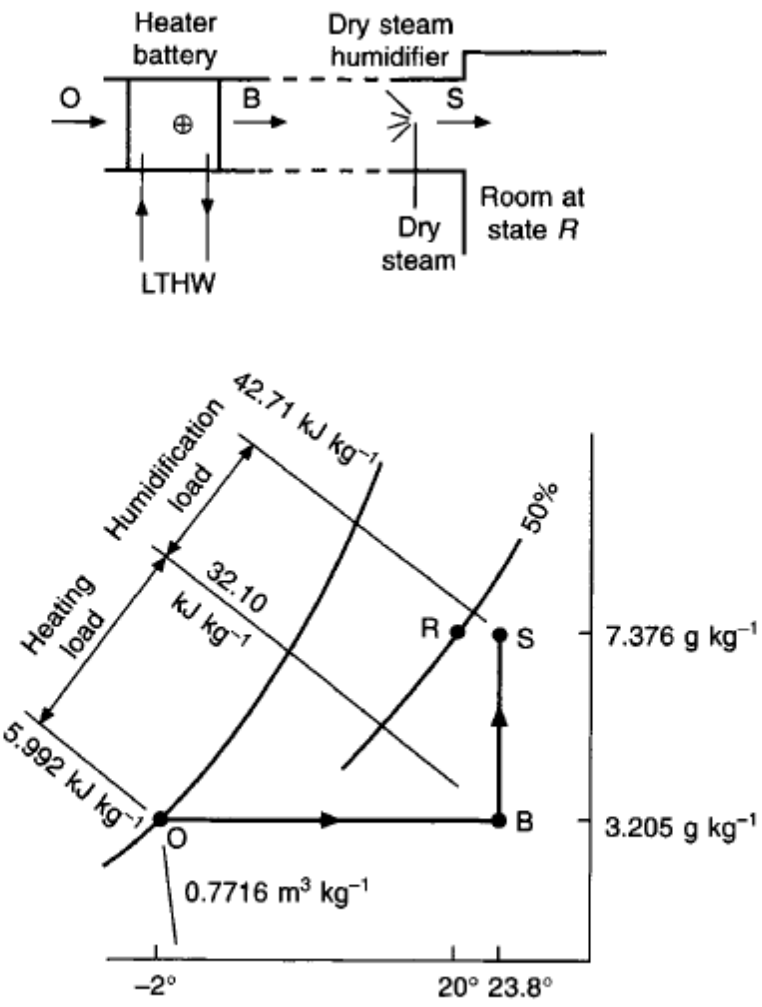


Fig. 3.14 The use of dry steam for humidification. See Example 3.15.

Alternatively

$$(0.5/0.7716) \times 1.018 \times (23.8 + 2) = 17.02 \text{ kW}$$

The first method is preferred because enthalpy values are based on well-established thermodynamic properties of dry air and water vapour whereas, on the other hand, the values of the specific heats used for the second method may not be exactly correct.

Assuming the change of state resulting from the injection of dry saturated steam is up a dry-bulb line, the state of the air supplied to the room is 23.8°C dry-bulb and 7.376 g kg^{-1}, at which the enthalpy is determined as 42.71 kJ kg^{-1} from tables or a chart. Hence the humidification load is

$$(0.5/0.7716) \times (42.71 - 32.10) = 6.88 \text{ kW}$$

Exercises

1. Dry saturated steam at 1.2 bar and 104.8°C is injected at a rate of 0.01 kg s^{-1} into an airstream having a mass flow rate of dry air of 1 kg s^{-1} and an initial state of 28°C dry-bulb, 11.9°C wet-bulb (sling). Calculate the leaving state of the airstream if all the injected steam is accepted by it. Use equation (2.24) as necessary.

Answers

29.3°C, 11.937 g kg^{-1}.

2. Air at 40°C dry bulb temperature and 30% relative humidity is passed through an adiabatic air washer at the rate of 28 m^3/min. If the effectiveness of the air washer is 80%, find the conditions of the air leaving the air washer, and the amount of water vapor added to the air per minute.

Answer

W_2= 0.019 kg/$kg_{d.a}$, t_{d2}= 28°C, ϕ_2= 80%, $\dot{m}_s$= 0.154 kg/min

3. An air conditioning system controls the indoor conditions of a room at 20°C dry bulb temperature and 50% relative humidity, when the outdoor conditions are at 5°C dry bulb temperature and 70% relative humidity. 2 m^3/s of recycled air from the room is mixing with 1 m^3/s of the fresh air, the mixture then flows across a heater and come out of it at 27°C, and supply to the room. Sketch the process on the psychrometric chart and determine:

(a) the amount of heat that gives the heater to the air in kW.

(b) the heating load of the room in kW.

Answer

45.13 kW, 14.56 kW

4. Air at 28°C dry bulb temperature and 50% relative humidity passes through a cooling coil at a rate of 1 m^3/s, the air leaves the coil at 10°C dry bulb temperature and 0.007 kg/$kg_{d.a}$ moisture content. Calculate the following by using the psychrometric chart.

(a) mass flow rate of the air (b) total heat removed from the air

(c) contact factor of the coil (d) rate of condensing of water vapor

Answer: 1.150 kg/s, 34.5 kW, 0.81, 20.286 kg/h.

5. 0.95 m^3/s of outside air passes over a cooling coil. The outdoor air is at 35°C dry bulb and 25°C wet bulb temperature, and the conditioned space is being maintained at 26°C dry bulb temperature and 45% relative humidity. the sensible heat ratio is calculated as 0.72. the air leaving the coil is 90% saturated.

(a) find the apparatus dew point, and the temperature of the air leaving the coil

(b) how much cooling in kW is the unit doing

(c) how much moisture in $kg/kg_{d.a}$ is condensed out of the incoming air per hour.

Answer

(a) 8°C, 10.4°C (b) 51.39 kW (c) 34.47 kg/h

6. The sensible heat gain of a room is 4.8 kW and its latent heat gain is 1.4 kW. A conditioned air supply of 0.5 m^3/s is to be delivered to the room. If the room is to be maintained at 25°C DBT, find the relative humidity that will result in the conditioned room if the air supply is 17°C and 90% RH.

Answer

60%

7. Air enters a chamber at 10°C DBT and 5°C WBT at a rate of 100 m^3/min. The barometer reads a pressure of 1.01325 bar. While passing through the chamber, the air absorbs sensible heat at the rate of 40 kW and picks up 45 kg/h of saturated steam at 105°C. Determine the dry and wet bulb temperatures of the air leaving the chamber.

Answer

26.5°C, 18.1°C

Chapter Four

Thermal Comfort

One of the main purposes of an air-conditioning system is to provide conditions for human thermal comfort. Comfort is related to levels of optimum acceptability which has been established by response tests for all types of subjects under varying environmental conditions of temperature and humidity. It is a subjective quality, personal to individuals depending on sex, age, state of health, clothing, environmental conditions, and very often depending on individual preferences. In general, comfort occurs when body temperatures are held within narrow ranges, and skin moisture is low. The level of noise in the controlled environment also affects the feeling of comfort.

4.1 Heat Balance Equation

The physical basis of comfort lies in the thermal balance of the body, i.e. the heat produced by the body's metabolism must be dissipated to the environment, otherwise the body would overheat.

The total energy production rate of the body is the sum of the production rates of heat $\dot{Q}$ and work $\dot{W}$ and can be written in the form

$$\dot{Q} + \dot{W} = MA_{skin} \quad \text{..... (4.1)}$$

Where

M is the rate of metabolic energy production per unit surface area

A_{skin} is the total surface area of skin.

The thermal balance of the body can be expressed by the equation,

$$S = (M - W) - E \pm R \pm C \quad \text{..... (4.2)}$$

Where

(M – W) is the net surplus heat to be liberated or stored (metabolic rate minus the useful rate of working)

E is the heat loss by evaporation

R is the heat gain or loss by radiation

C is the heat gain or loss by convection

S is the rate at which heat is stored within the body.

Under steady state conditions, the body remains comfortable and healthy because S is zero. In an oppressively hot environment, the load imposed upon E, R and C may be so great that S is positive and the body temperature will rise, eventually resulting in heat stroke.

Heat dissipation from the body (Table 4.1) to immediate surroundings occurs by several modes of heat exchange:

- Sensible heat flow from the skin
- Latent heat flow from evaporation of sweat and from evaporation of moisture diffused through the skin
- Sensible heat flow during respiration
- Latent heat flow due to evaporation of moisture during respiration.

Table 4.1 Heat output of the body in various activities

Activity	Watts
Sleeping	min. 70
Sitting, moderate movement, e.g. typing on computer	160 – 190
Sitting, heavy arm and leg movements	190 – 230
Standing, moderate work, some walking	220 – 290
Walking, moderate lifting or pushing	290 – 410
Intermittent heavy lifting, digging	440 – 580
Hard, sustained work	580 – 700

Sensible and latent heat losses from the skin are typically expressed in terms of environmental factors, skin temperature, and skin wettedness. The main independent environmental variables can be summarized as air temperature, mean radiant temperature and relative air velocity and ambient water vapor pressure.

4.2 Thermal Interchange with Environment

The human body is continually gaining and producing heat as well as losing heat to its surroundings to maintain temperature equilibrium. Body heat gains come from two source:

- Heat produced within the body itself as a result of metabolic processes.

- Heat gained by body from external sources, by radiation from the sun or other hot objects or surfaces, and by convection from the surrounding air.

Heat is lost from the body by:

(a) **Conduction:** Heat loss by conduction depends on the temperature difference between the body surface and the object with which the body is in direct contact. Heat lost by conduction from the body can be neglected as the amount of body surface in contact with an external surface is usually too small and the period of contact is short too.

(b) **Convection** (about 30%): Heat loss due to convection takes place from the body to the air in contact with the skin or clothing. The rate of convection heat loss is increased by a faster rate of air movement, by a lower air temperature and a higher skin temperature.

(c) **Radiation** (about 45%): Radiant heat loss depends on the temperature of the body surface and the temperature of the opposing surfaces. Thus the human body will radiate heat to walls, ceilings, floors, windows, and to the out of doors if these surfaces are at a lower temperature than the body surface. Conversely, the body gains by radiation from the sun or from any surface warmer than the skin surface. Body skin temperature ranges between 30°C and 34°C with an average of 32.2^{o}C for a healthy person engaged in light activity.

(d) **Evaporation** (about 25%): Heat loss by evaporation fan is governed by the rate of evaporation, which in turn depends on the humidity of air (the dryer the air, the faster the evaporation) and on the amount of moisture available for evaporation.

Metabolic heat generation

In choosing optimal conditions for comfort and health, knowledge of the energy expended during the course of routine physical activities is necessary, since heat production increases in proportion to exercise intensity. The unit used to measure the metabolic rate is *met*. One met represents the average heat produced by a sedentary average person at normal mean radiant temperature, i.e. 1 met = 58.2 W/m^2. Table 4.2 lists the typical metabolic heat generation for various activities.

Table 4.2 Typical metabolic heat generation for various activities

Activities	W/m^2	met
Resting		
Sleeping	40	0.7
Reclining	45	0.8
Seated, quiet	60	1.0
Standing, relaxed	70	1.2
Walking		
3.2 km/h (0.9 m/s)	115	2.0
4.3 km/h (1.2 m/s)	150	2.6
6.4 km/h (1.8 m/s)	220	3.8
Office activities		
Reading, seated	55	1.0
Writing	60	1.0
Typing	65	1.1
Filing, seated	70	1.2
Filing, standing	80	1.4
Walking about	100	1.7
Lifting/packing	120	2.1

Clothing affects comfort, since it acts as an insulation. The unit measuring the insulating effect of clothing on a human subject is *clo*, where, 1 *clo* = 0.155 km^2/W.

4.3 Environmental Parameters and Indices

Environmental parameters

Environmental parameters that affect human comfort can be categorized into (a) directly measured parameters and (b) calculated parameters.

The following are the frequently used directly measured psychrometric parameters:

- Dry bulb temperature
- Wet bulb temperature
- Dew point temperature
- Water vapor pressure
- Total atmospheric pressure
- Relative humidity
- Humidity ratio
- Air velocity

The mean radiant temperature is derivable and, hence, a *calculated parameter*. It is the temperature of a uniform black enclosure in which a solid body or occupant would exchange the same amount of radiant heat as in the existing non-uniform environment. Fanger identified two additional calculated parameters, which are *activity level* and *clothin*g. In addition to the above, the other secondary factors such as day-to-day temperature variation, age, adaptability, sex, etc. also influence comfort.

Environmental indices

An environmental index combines two or more parameters, such as air temperature, mean

radiant temperature, humidity or air velocity into a single variable. The *effective temperature* (ET^*) is probably the most common environmental index and has the widest range of applications.

The *effective temperature* (ET^*) is defined as the dry bulb temperature of a uniform enclosure at 50% RH in which humans would have the same net heat exchange by radiation, convection, and evaporation as they would in the varying humidities of the test environment.

Another approach used to evaluate the combined effect of temperature and humidity is *the Heat Stress Index*. This index is the ratio of the total evaporative heat loss required for thermal equilibrium to the maximum evaporative heat loss possible for the environment, multiplied by 100 for steady-state conditions (skin temperature is held constant at 35°C in order to limit the rise in body temperature, the sweat rate should not exceed one liter per hour to limit the loss of body fluid). The heat stress index is therefore defined as

$$\text{Heat Stress Index} = \dot{Q}_E / \dot{Q}_{E,max} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.3)$$

Where

$\dot{Q}_E$ is the actual evaporative loss

$\dot{Q}_{E,max}$ is the maximum evaporative heat loss with the skin temperature at 35°C.

4.4 Comfort Charts

In identical environments, different people perceive comfort in different ways. In the same built environment, some may feel chilly while others may feel warm. Dry bulb temperature is not a reliable indication of how warm or cold an occupant will feel in a room. The effects of both relative humidity and air velocity need also to be considered.

In the same context, ASHRAE and other researchers have conducted extensive research over the years to relate the above factors to human comfort. From the results of these tests emerged the concept of an *effective temperature*. This index is a measure of comfort which involves the combined effect of dry bulb, wet bulb, and air movement as judged by the subjects in the research studies. There were a number of different combinations of dry bulb and relative humidity which would give the same feeling of comfort to a high percentage of the subjects for a given air velocity.

A typical comfort chart shown in Figure 4.1 could then be constructed by drawing lines through the points at which the majority of people equally clothed and equally active reported the same feeling of comfort. These lines are called the *effective temperature* (ET) lines.

The range of the summer effective temperatures from around 19 to 24°C, while the range of the effective temperatures are from 17°C to 22°C.

Over the years a number of similar charts have been developed by ASHRAE and other researchers including Fanger who developed General Comfort Charts based on clothing, activities, air temperatures, etc.

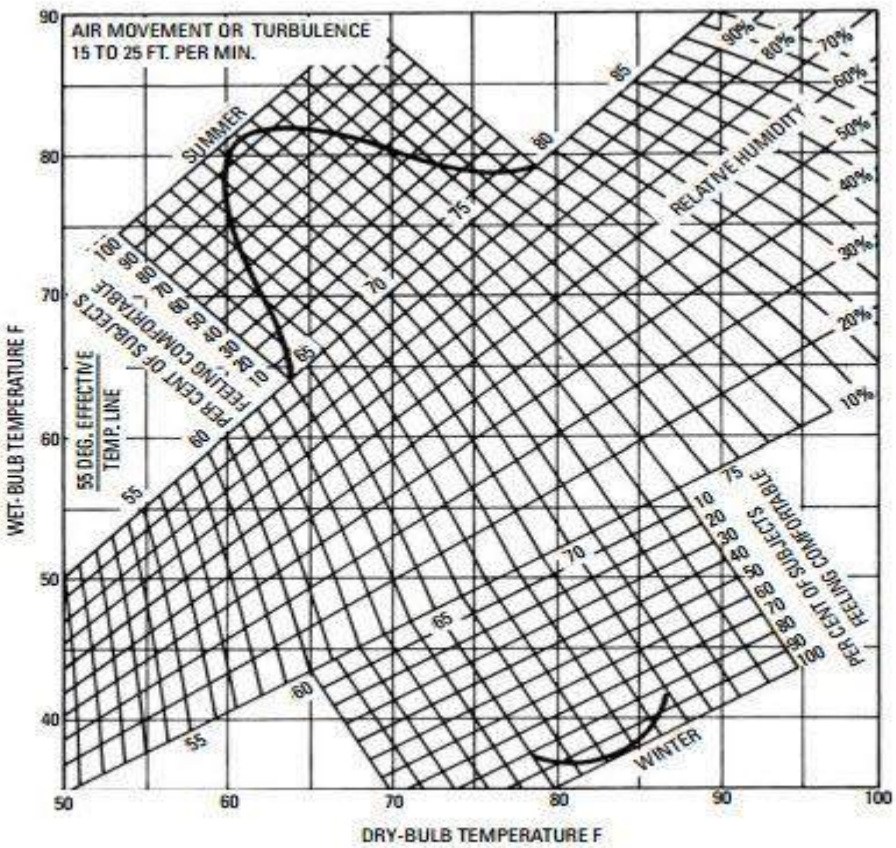
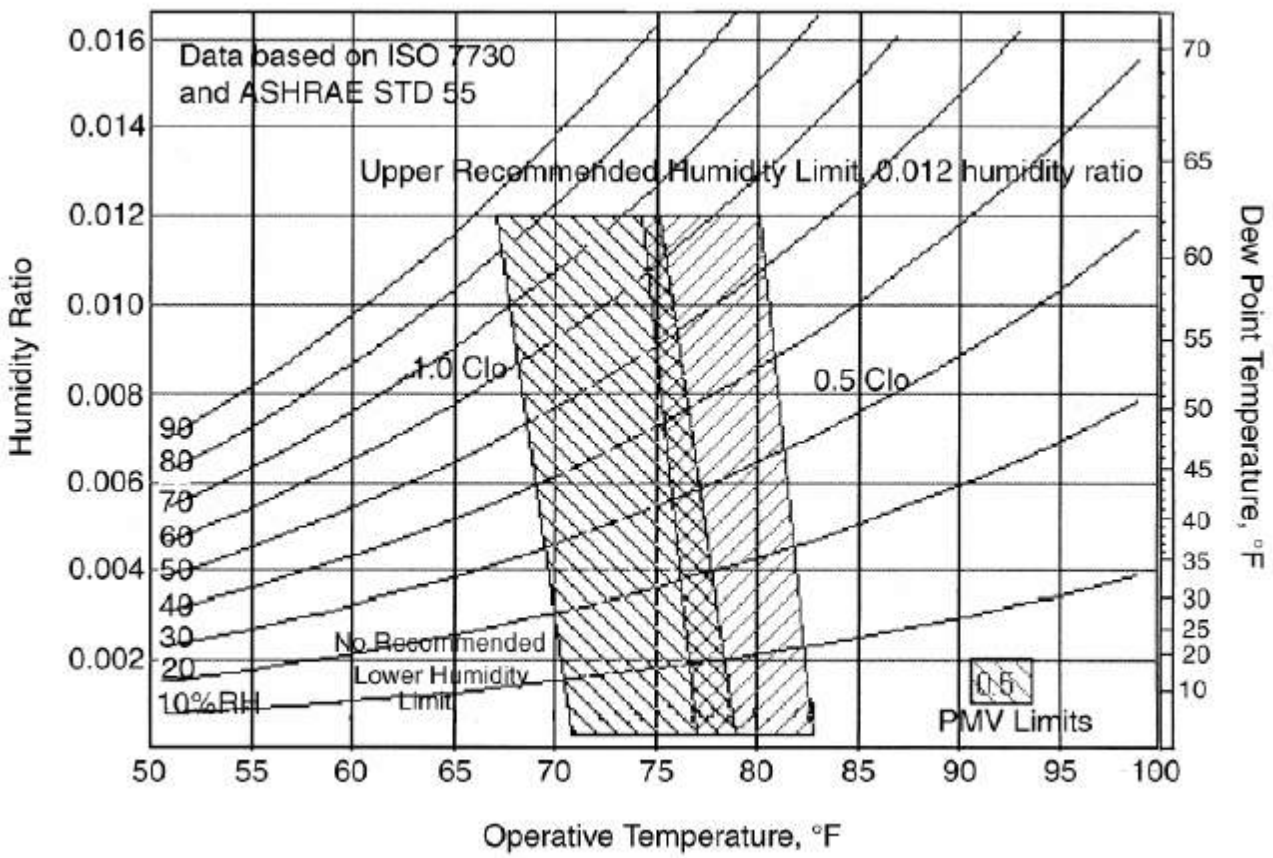


Figure 8-8 Comfort chart for still air. *(Courtesy ASHRAE 1960 Guide)*

(a) Comfort chart for still air



(b) ASHRAE comfort zones for summer and winter

Figure 4.1 (a) Comfort chart for still air (b) ASHRAE comfort zones for summer and winter.

4.5 Prediction of Thermal Comfort

Human thermal is influenced by physiological factors, it is difficult to specify a single quantity for evaluating human comfort. The usual comfort parameters are ambient air temperature, humidity, air motion, body activity level, and clothing. However, it has been observed that if the surrounding surfaces are below the air dry bulb temperature, comfort would occur at a higher effective temperature than that indicated by Fig. 4.1. This implies that radiant cooling affects comfort parameters/sensation appreciably. Studies have also indicated that women of all ages prefer an effective temperature about one degree higher than that preferred by men, while both men and women over 40 years age prefer an effective temperature about one degree higher than that desired by younger people. People of all climatic regions have identical preferred temperatures. The activity level of the occupants and the duration of occupancy also affect human thermal comfort sensation.

Thermal comfort and thermal sensation can be predicted by (a) a comfort chart and (b) numerically by the predicted mean vote (**PMV**) and the predicted percentage of dissatisfied

(**PPD**). The predicted mean vote predicts the mean response of a large group of people. This comfort sensation scale as developed by Rohles and Nevins is shown in Table 4.3.

Table 4.3 ASHRAE thermal sensation scale

+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

The test results have been correlated with the dry bulb temperature, humidity level, sex, and duration of exposure. The basic equation used to compute the **PMV** is

$$PMV = a^{*}t + b^{*}p_v + c^{*} \quad \text{.....} \quad (4.4)$$

Where, **t** is the dry bulb temperature (^{o}C) and $\mathbf{p_v}$ is the corresponding saturation pressure (kPa). a^{*}, b^{*} and $c^{*}$are the coefficients used for calculating PMV. The values of a^{*}, b^{*} and c^{*}, can be obtained from Table 4.4.

Table 4.4 Coefficient a^{*}, b^{*} and c^{*} used to calculate the predicted mean vote (PMV)

Exposure period (hr)	Sex	a^{*}	b^{*}	c^{*}
1.0	Male	0.220	0.233	-5.673
1.0	Female	0.272	0.248	-7.245
1.0	Combined	0.245	0.248	-6.475
3.0	Male	0.212	0.293	-5.949
3.0	Female	0.275	0.255	-8.622
3.0	Combined	0.243	0.278	-6.802

For young adult subjects with sedentary activity and wearing clothing with a thermal resistance of approximately 0.5 clo, air velocity (0.2 m/s).

After calculating the PMV, the PPD is estimated for the same condition (Fig. 4.2). The dissatisfied occupants are defined as those who do not vote either +1, 0 or -1 on the PMV scale. The PMV- PPD model is widely used and accepted for design and field assessment of comfort conditions.

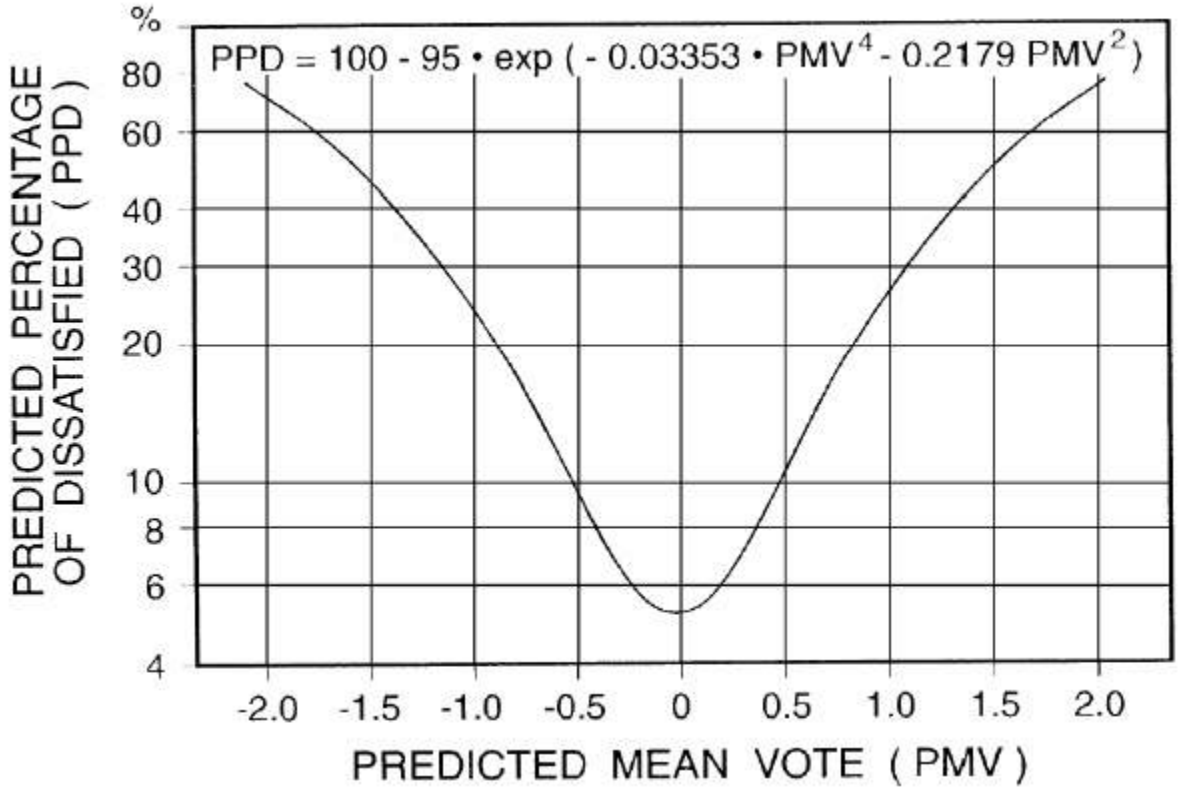


Figure 4.2 PPD as a function of PMV

Example 4.1:

A number of male and female subjects took part in a climate chamber test. Determine the difference between the PMV of male and that of female occupants with the dry bulb temperature being 24°C and the dew point temperature being 20°C, one hour after entry into the space.

Solution:

$$PMV = a^{*}t + b^{*}p_v + c^{*}$$

PMV for men:

$$PMV = 0.22(24) + 0.233(2.339) - 5.673 = 0.051$$

Similarly, PMV for women:

PMV = 0.272(24) + 0.248(2.339) – 7.245 = - 0.136

From the above, both males and females are predicted to be thermally neutral.

The P_v values can be obtained from the table of thermodynamic properties of water at saturation.

4.6 Indoor Design Conditions

The most commonly recommended design conditions for comfort are:

$ET^* = 24^oC$

Dry bulb temperature = mean radiant temperature

Relative humidity = 50% (30 – 70)%

Air velocity less than 0.2 m/s

The indoor conditions to be maintained within a building are the dry bulb temperature and relative humidity of the air at the breathing line, 1 to 1.5 m above the floor, in an area that would indicate average condition at that level and which would not be affected by abnormal or unusual heat gains or losses from the interior or exterior.

Table 4.5 shows the guideline room air temperatures for different applications.

Table 4.5

GUIDELINE ROOM AIR TEMPERATURES

Type of Space	°F Summer	°F Winter	°C Summer	°C Winter
Residences, apartments, hotel and motel guest rooms, convalescent homes, offices, conference rooms, classrooms, courtrooms, and hospital patient rooms	74–78	68–72	23–26	20–22
Theaters, auditoriums, churches, chapels, synagogues, assembly halls, lobbies, and lounges	76–80	70–72	24–27	21–22
Restaurants, cafeterias, and bars	72–78	68–70	22–26	20–21
School dining and lunch rooms	75–78	65–70	24–26	18–21
Ballrooms and dance halls	70–72	65–70	21–22	18–21
Retail shops and supermarkets	74–80	65–68	23–27	18–20
Medical operating rooms[a]	68–76	68–76	20–24	20–24
Medical delivery rooms[a]	70–76	70–76	21–24	21–24
Medical recovery rooms and nursery units	75	75	24	24
Medical intensive care rooms[a]	72–78	72–78	22–26	22–26
Special medical care nursery units[a]	75–80	75–80	24–27	24–27
Kitchens and laundries	76–80	65–68	24–27	18–20
Toilet rooms, service rooms, and corridors	80	68	27	20
Bathrooms and shower areas	75–80	70–75	24–27	21–24
Steam baths	110	110	43	43
Warm air baths	120	120	49	49
Gymnasiums and exercise rooms	68–72	55–65	20–22	13–18
Swimming pools	75 or above	75	24	24
Locker rooms	75–80	65–68	24–27	18–20
Children's play rooms	75–78	60–65	24–26	16–18
Factories and industrial shops	80–85	65–68	27–29	18–20
Machinery spaces, foundries, boiler shops, and garages	—	50–60	—	10–16
Industrial paint shops	—	75–80	—	24–27

[a] Variable temperature range required with individual room control.

4.7 Quality and Quantity of Air

The air in an occupied space should, at all times, be free from toxic, unhealthful or disagreeable fumes such as carbon dioxide. It should also be free from dust and odors. In order to obtain these conditions, enough clean outside air must always be supplied to an occupied space to counteract or adequately dilute the source of contamination. The concentration of odor in a room depends upon many factors such as dietary and hygienic habits of occupants, type and amount of outdoor air supplied, room volume and type of odor sources. In general, when there is no smoking in a room, 1 m^3/min per person of outside air will take care of all the conditions. But when smoking takes place in a room, 1.5 m^3/min per person of outside air is necessary. For general application, a minimum of 0.3 m^3/min of outside air per person, mixed with 0.6 m^3/min of recirculate air is good.

Table 4.6 lists the ventilation standards while Table 4.7 gives the recommended volume of ventilation air for various buildings.

Table 4.7 Ventilation Standards

Application	Smoking	m^3/min per person		m^3 per m^2 of floor area
		recommended	minimum	minimum
Apartment	Some	0.57	0.43	----
Banking space	Occasional	0.29	0.21	----
Barber shops	Considerable	0.43	0.28	----
Broker's board rooms	Very heavy	1.42	0.85	----
Corridors	----	----	----	0.007
Departmental stores	None	0.21	0.14	0.001
Director's rooms	Extreme	1.42	0.84	----
Drug stores	Considerable	0.29	0.21	----
Factories	None	0.29	0.21	0.003
Garage	----	----	----	0.028
Hospital operation	None	----	----	0.056
Room	None	0.85	0.71	0.009
Hospital private room	None	0.57	0.43	----
Hospital wards	Heavy	0.85	0.71	0.009
Hotel rooms	----	----	----	0.112
Kitchen restaurant	----	----	----	0.056
Kitchen residence	Heavy	1.42	----	0.056
Meeting rooms	None	----	----	----
School room	Some	0.43	0.28	----
Office general	None	0.71	0.43	0.007
Office private	----	----	----	----

Table 4.8 Recommended volume of ventilation air for various buildings

Type of building	No. of changes per hour		m^3/h per occupant		m^3/m^2 floor area per hour	
	min	Mix	min	Mix	min	mix
Commercial						
Garages	6	12	----	----	----	----
Offices	1.5	12	----	----	----	----
Waiting rooms	4	6	----	----	----	----
Restaurant (dining)	4	20	----	----	----	----
Restaurant (kitchen)	4	60	----	----	80	80
Stores	6	12	----	----	----	----
Farm Building						
Cow	----	----	100	100	----	----
Horse	----	----	120	140	----	----
Sheeps	----	----	200	300	----	----
Hospitals						
Dining rooms	6	12	----	----	30	30
Kitchens	20	60	----	----	80	80
Operating room	----	----	100	100	----	----
Toilets	75	30	----	----	40	40
Wards	----	----	70	150	20	20
Hostels						
Cafes	7.5	7.5	----	----	----	----
Dining rooms	4	20	----	----	30	30
Guest rooms	3	5	----	----	----	----
Kitchens	4	60	----	----	80	80
Lobbies	3	4	----	----	----	----
Toilets	10	12	----	----	----	----
Residences						
Bathrooms	1	5	----	----	----	----
Halls	1	3	----	----	----	----
Kitchen	1	40	----	----	----	----
Sleeping room	----	1	----	----	----	----
Various public spaces						
Auditorium¸ church¸ dance	4	30	20	140	30	40
Hall	----	----	60	80	40	40
Classrooms	12	12	----	----	30	30
Gymnasium	6	20	----	----	----	----
Laboratories	2	10	----	----	40	40
Locker rooms	30	30	----	----	40	40
Projection booths	3	5	----	----	----	----
Reading rooms	3	12	----	----	----	----
Engine and boiler room	----	----	----	----	----	----

4.8 Outside Design Conditions

It is observed that there is a kind of sinusoidal relationship between the air dry bulb temperature and the sun time. For example, in the month of June in a certain locality where the sunrise is at about 5 a.m. and sunset at about 7 p.m. the time minimum temperature falls at about 4 a.m. and that of maximum temperature at about 4 p.m. i.e., with a lapse of about 12 hours. As regards relative humidity, it is seen that it reaches a minimum value in the afternoon. Since the mean daily maximum dry bulb temperature occurs between 1 p.m. and 4 p.m., it is reasonable to assume that the minimum relative humidity would occur during the same period.

The air conditioning load is estimated to become a base for selecting the conditioning equipment. It must take into account the entering into the space from outdoors on a design day, as well as the heat being generated within the space. The outside design data is available in a tabular form. Many guide and data books like ASHRAE give outside design conditions for different places this can be observed in Table 4.9.

Heating Load Calculation

Heating loads are the thermal energy that must be supplied to the interior of a building in order to maintain the desired comfort conditions. There are two kinds of heat losses; (1) the heat transmitted through the walls, ceiling, floor, glass, or other surfaces, and (2) the heat required to warm outdoor air entering the space.

The actual heat loss problem is transient because the outdoor temperature, wind velocity, and sunlight are constantly changing, but during the coldest months, however sustained periods of very cold, cloudy, and stormy weather with relatively small variation in outdoor temperature may occur. In this situation heat loss from the space will be relatively constant and maximum. Therefor for design purposes the heat loss is usually estimated for steady- state heat transfer for some reasonable design temperature.

Heat loss (gain) calculations by the (BTU) or (kW) method is an accurate process of determining the heat transmission through building materials. The established (U) factors of combinations of building materials gives a simple corrected method of determining the loss of heat from a building.

With the cost of material continually rising, accurate heat loss is a must. Systems can be designed to do the job efficiently without the long used *safety factor*. The safety factor is only a "cost more" factor.

Conductivity for Plan Wall and Steady State

Fourier equation;

$q = - KA\, dT/dx \rightarrow q = - dT/(dx/KA)$ ……. (5.1)

Where:

q: heat flow (kW)

dT: thermal potential difference (^oC)

dx: wall thickness (m)

K: thermal conductivity (W/m.^oC)

A: perpendicular area (m^2)

We can say that; $R_{th} = dx/KA$

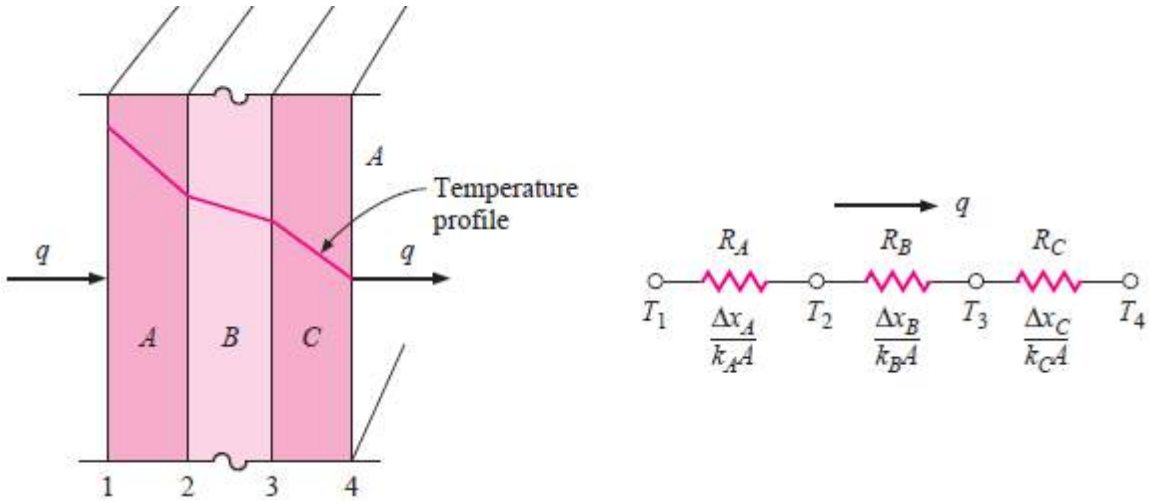


Figure 5.1 One-dimensional heat transfer through a composite wall and electrical analogy.

The temperature gradients in the three materials are shown in the Fig.5.1, and the heat flow may be written

$$q = - K_A A (T_2 - T_1)/ \Delta x_A = - K_B A (T_3 - T_2)/ \Delta x_B = - K_C A (T_4 - T_3)/\Delta x_C \quad \ldots\ldots (5.2)$$

$$q = (T_1 - T_4)/(\Delta x_A/ K_A A + \Delta x_B/ K_B A + \Delta x_C/ K_C A)$$

$$= \Delta T_{overall} / \Sigma R_{th} \quad \ldots\ldots (5.3)$$

<u>Convection heat transfer</u>

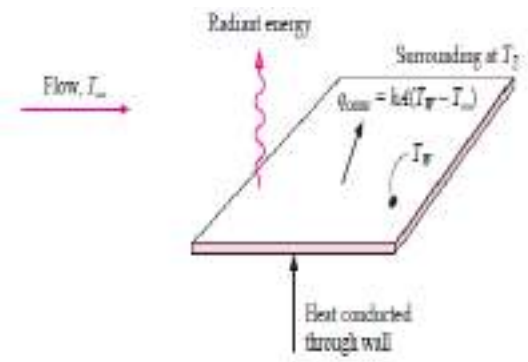


Figure 5.2 Convection heat transfer.

$$q = hA(T_W - T_\infty) \quad \ldots\ldots (5.4)$$

where; A is the surface area

$q = \Delta T/(1/h A)$, where $(1/h A)$ is the convection thermal resistance.

Overall heat transfer coefficient

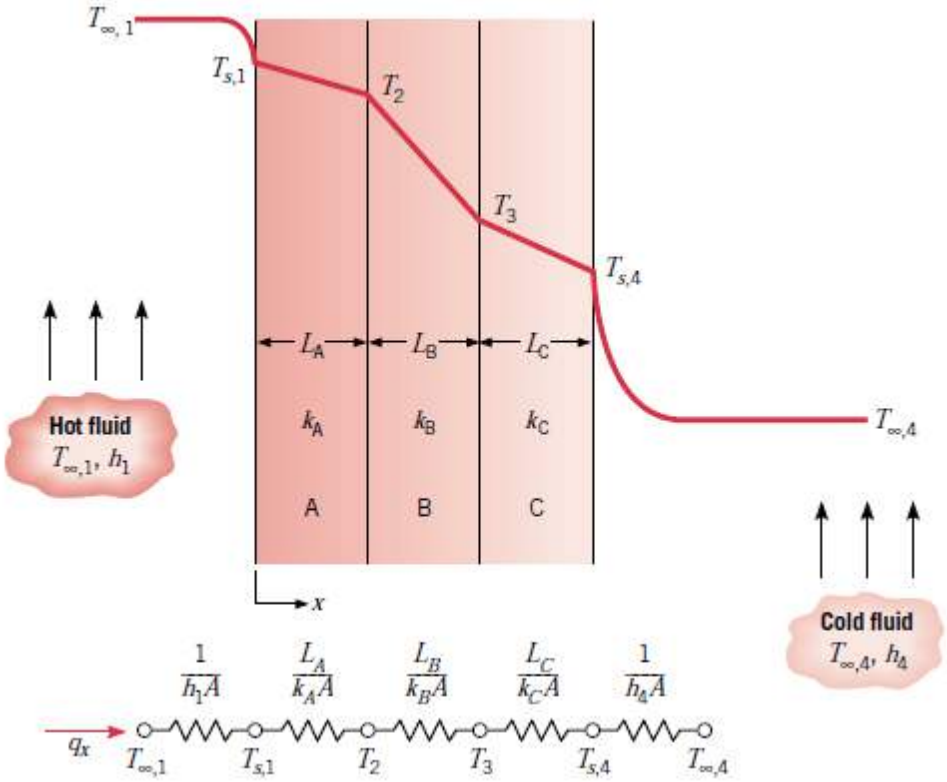


Figure 5.3 Equivalent thermal circuit for series composite wall.

$q = \Delta T_{overall} / \Sigma R_{th}$

$\Delta T_{overall} = T_i - T_o$

$\Sigma R_{th} = R_i + R_A + R_B + R_C + R_O$

Equation of heat transfer can be written as;

$q = U\ A\ \Delta T_{overall}$ …….. (5.5)

Where

U is the overall heat transfer coefficient = $1/\ \Sigma R_{th}$

To determine how much heat is lost through the walls, we have "U" factors for different types of construction. A "U" factor tells us how many (kW's) are transmitted to the colder (hotter) outside for one square meter of area with a difference of one degree between the outside surface and inside surface of the walls, windows, roofs or ceilings.

Chapter Five

Definitions

1. "U" – Coefficient of heat transmission (overall): The amount of heat transmitted from air to air per square meter of the wall, roof, or ceiling for a difference in temperature of one degree centigrade between the air on the inside and outside (winter) of the wall, floor, roof or ceiling.

2. "K" – Conductivity: The amount of heat (W) transmitted through one square meter of a homogeneous material one meter thick for a difference in temperature of one degree centigrade between two surfaces of the material.

3. "C" – Conductance: The amount of heat (W) transmitted from surface to surface through one square meter of material or construction for a difference in temperature of one degree centigrade between the two surfaces. This is not per meter or centimeter of thickness, but for thickness shown.

4. "f_o" – Outside film coefficient: The outside combined surface loss due to radiation and convection, with a 5 or 10 km/h (or more) wind velocity. The amount of heat (W) for one square meter of surface for a temperature difference of one degree centigrade($W/m^2.^oC$).

5. "f_i" – Inside film coefficient: The combined inside surface loss due to radiation and convection, with still air ($W/m^2.^oC$).

6. "a"– Thermal conductance of an air space: The amount of heat (W) transmitted through one square meter of surface for a difference in temperature of one degree centigrade ($W/m^2.^oC$).

7. "R" – This is the reciprocal of conductivity and conductance, or the overall heat transfer coefficient "U".

The total resistance "R" to heat flow through a wall is equal numerically to the sum of the resistance in series.

$$U_{Tot} = 1/R_{Tot}$$

$$R = 1/C \,, \quad R = 1/a \,, \quad R = 1/f \,, \quad R = X/K$$

$$U_{Tot} = 1/\Sigma R = 1/(1/f_i + 1/a + X_1/K_1 + X_2/K_2 + \text{----------} + 1/f_o)$$

Example:

A wall consist of 10 cm face brick with 15 cm concrete and 1.3 cm cement plaster. The outdoor temperature is -7°C, and the wind velocity is 24 km/h, while the indoor space temperature is 25°C. Calculate:

(a) Thermal resistance of the wall (b) Overall heat transfer coefficient

(c) Rate of heat transfer through the wall (d) Rate of heat transfer when neglect the thermal resistance of the air layers.

Solution:

Use the table of thermal conductivity of building materials to find the following:

For face brick $K_b = 1.30$ W/m.°C

For concrete $K_C = 1.73$ W/m.°C

For cement plaster $K_p = 0.721$ W/m.°C

$f_i = 8.29$ W/m^2.°C, $f_o = 34.1$ W/m^2.°C

(a) The total thermal resistance (ΣR_{th})

$$\Sigma R_{th} = 1/f_o + X_b/K_b + X_c/K_c + X_p/K_p + 1/f_i$$

$$= 1/34.1 + 0.1/1.30 + 0.15/1.73 + 0.013/0.721 + 1/8.29$$

$$= 0.332 \text{ m}^2.^oC/W$$

(b) The overall heat transfer coefficient (U)

$$U = 1/\Sigma R_{th} = 1/0.332 = 3.012 \text{ W/m}^2.^oC$$

(c) The rate of heat transfer through the wall (q/A)

$$q/A = U (T_i - T_o) = (3.012)(25 - (-7)) = 96.384 \text{ W/m}^2$$

$$\Sigma R_{th} = X_b/K_b + X_c/K_c + X_p/K_p$$

$$= 0.1/1.30 + 0.15/1.73 + 0.013/0.721 = 0.182 \text{ m}^2.^oC/W$$

$$q/A = (1/0.182)(25 + 7) = 175.82 \text{ W/m}^2$$

Example:

Door made of wood sheet and contains a glass in the middle, which represent 80% of the door area. The thickness of the door is 40 mm and the room temperature is 22°C, while the outdoor temperature is 5°C. Determine the rate of heat loss from the room, if the dimensions of the door are 2 m x 1 m.

Solution:

Door area = 2 x 1 = 2 m^2

Wood area = 0.2 x 2 = 0.4 m^2

Glass area = 0.8 x 2 = 1.6 m^2

K_W = 0.159 W/m.°C , C^*_g = 6.25 W/m^2.°C

Rate of heat transfer through the wood of the door:

$\Sigma R_{th} = 1/f_o + X_W/K_W + 1/f_i$

$= 0.121 + (0.04/0.159) + 0.029 = 0.402\ m^{2o}C/W$

$q_w = (1/0.402) \times 0.4 \times (22 - 5) = 16.9$ W

Rate of heat transfer through the glass:

$q_g = U.A.(t_i - t_o) = 6.25 \times 1.6 \times (22 - 5) = 170$ W

Then the total heat loss through the door:

$q_d = q_w + q_g = 16.9 + 170 = 186.9$ W.

The Temperature of the Surface of the Wall

It is important to know that the inside and outside air temperature is not equal to wall surface temperature. So in case when the temperature of inside wall surface reaches or below the dew point of indoor space temperature, it will causes to condensate the water vapor and may be defect this wall (especially in the winter). It is important to make a chick about this point.

Example:

An external wall consist of three layers, layer A (X= 5 cm, K= 0.4 W/m.K), layer B (X= 24 cm, K= 0.6 W/m.K), layer C (X= 5 cm, K= 0.8 W/m.K). If the indoor space condition is 20°C DBT and 14°C WBT, and the outdoor temperature is -15°C. Chick is the vapor condensate on the inner surface of the wall or not?

Solution:

We must find the temperature of the inner surface (T_w) and compare it with the dew point of the air space, which is 10.5°C

$U = 1/\Sigma R_{th} = 1/(1/f_i + X_A/K_A + X_B/K_B + X_C/K_C + 1/f_o)$

$= 1/(1/8.29 + 0.05/0.4 + 0.24/0.6 + 0.05/0.8 + 1/34.1) = 1.36\ W/m^2.^oC$

$q = U.A.(T_i - T_o) = 1.36 \times 1 \times (20 + 15) = 47.6\ W/m^2$

Also

$q = f_i (T_i - T_s)$ where; T_s is the inner surface temperature

$47.6 = 8.29 \times (20 - T_s) \rightarrow T_s = 14.26^oC$

The inner surface temperature is greater than the dew point, so the vapor will not condenses.

Example:

A wall was constructed of hollow concrete blocks (C = 5.11 $W/m^2.^oC$). The indoor condition is 20°C DBT and 14°C WBT, when the outdoor air temperature is -15°C and the wind velocity is 24 km/h in winter. Determine the inner surface temperature, and show if the vapor condenses on it or no.

Solution:

The dew point for the indoor air from the psychromertic chart is = 9.3°C

$\Sigma R_{th} = 1/f_i + 1/C + 1/f_o = (1/8.29) + (1/5.11) + (1/34.1) = 0.346\ m^{2o}C/W$

$R_f/ R_t = \Delta T_f/ \Delta T_t$

$0.121/0.346 = (20 - T_s)/(20 + 15)$

$T_s = 7.8^oC$

The inner surface temperature is lower than the dew point, so the vapor will condenses.

Infiltration

All structures have some air leakage or infiltration. This means a heat loss because the cold dry outdoor air must be heated to the inside design temperature and moisture must be added to increase the humidity to the design value. The heat required to increase the temperature is given by

$$q_s = \dot{m}\, C_p\, (T_i - T_o) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5.6)$$

Where

q_s: sensible heat loss

$\dot{m}$: mass flow rate of the infiltration air

C_p: specific heat of moist air

Infiltration is usually estimated on the basis of volume flow rate at outdoor conditions.

Then equation 5.6 become:

$$q_s = (\dot{V}/v)\; C_p\, (T_i - T_o) = 1.22\, \dot{V}\, (T_i - T_o) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5.7)$$

Where

$\dot{V}$: infiltration volume flow rate (m^3/s)

v : specific volume (m^3/kg)

The latent heat required to humidify the air is given by

$$q_L = \dot{m}\, (W_i - W_o)\, h_{fg} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5.8)$$

$$= (\dot{V}/v)\, (W_i - W_o)\, h_{fg} = 3010\, \dot{V}\, (W_i - W_o) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5.9)$$

Where

$(W_i - W_o)$: the difference in design moisture ratio ($kg/kg_{d.a}$)

h_{fg}: the latent heat of vaporization at indoor condition (J/kg) or (kJ/kg)

More than one method is used in estimating air infiltration in building structures.

1. Air Change Method

Experience and judgment are required to obtain satisfactory results with this method. Experienced engineers will often simple make an assumption of the number of air changes per hour (ACH) that a building will experience based on their appraisal of the building type,

construction, and use. The range will usually be from 0.5 ACH (very low) to 2.0 ACH (very high). This approach is usually satisfactory for design load calculation but not recommended for the beginner.

In practice, the following values of air changes per hour can be used with reasonable precision for rooms with the extent of windows and external doors given.

No windows or exterior doors	0.5
Exterior doors or windows on one side	1
Exterior doors or windows on two sides	1.5
Exterior doors or windows on three sides	2
Entrance halls	2

2. Crack Method

The flow (leak) through an opening is proportional to the area of the cracks, the type of the cracks, and the pressure difference across the crack.

$\dot{V} = A.C.\Delta P^n$ ……. (5.10)

Where

A: effective leak area of the cracks

C: flow coefficient, which depends on the type of crack and the nature of the flow in the crack.

ΔP: outside – inside pressure difference ($P_o - P_i$)

N: Exponent that depends on the nature of the flow in the crack $0.4 < n < 1.0$.

The following table gives the leakage rates through cracks in doors on the windward side for different wind velocities and different door constructions.

Type of door	m^3 per linear meter of crack					
	Wind velocity, km/h					
	8	16	24	32	40	48
Glass door, good installation 3.2 mm crack	0.3	0.6	0.9	1.21	1.49	1.77
Average installation 4.76 mm crack	0.45	0.93	1.3	1.86	2.23	2.7
Poor installation 6.4 mm crack	0.6	1.21	1.77	2.42	2.42	3.53
Ordinary wood or metal door well fitted W-stripped	0.04	0.06	0.08	0.12	0.16	0.2
well fitted now W-stripped	0.08	0.11	0.17	0.24	0.31	0.39
Poorly fitted Now W-stripped	0.08	0.21	0.34	0.48	0.61	0.78
Factory door 3.2 mm crack	0.3	0.6	0.9	1.21	1.49	1.77

Example:

Find the sensible heat loss due to infiltration outdoor air at -6°C into a heated house through an aluminum window (2 m x 1.2 m), which has double swinging parts the width of each is 0.5 m, and a fixed glass in the middle. The temperature of the air inside the house is 22°C.

Solution:

Crack length = 2[(1.2 + 0.5) x2] = 6.8 m

ASHRAE Handbook, 1981 gives value of 0.77 (l/s. m) infiltration air through aluminum windows when the air velocity is 40 km/h.

$$\dot{V} = (0.77/1000) \times 6.8 = 0.00524 \ m^3/s$$

And from equation 5.7

$$q_s = 1.22 \ \dot{V} (T_i - T_o) = 1.22 (0.00524)(22 + 6) = 0.179 \ kW.$$

Ventilation

The introduction of outdoor air for ventilation of conditional spaces is necessary to dilute the odors given off by people, smoking and other internal air contaminates.

The amount of ventilation required varies primarily with the total number of people, the ceiling height and the number of people smoking. People give off body odors which required a minimum of 5 cfm (2.36 l/s) per person. When people smoke, the additional odors given off by cigarettes or cigars requires a minimum of 15 to 25 cfm (7 – 12 l/s) per person. In special gathering rooms with heavy smoking, 30 to 50 cfm (15 – 24 l/s) per person is recommended.

Tables 4.7 and 4.8 are used to determine the minimum and recommended ventilation air quantity for several applications.

The sensible and latent heating loads from ventilation air can be estimated by the following equations

$$q_s = (\dot{V}/v)\ C_p\ (T_i - T_o) = 1.22\ \dot{V}\ (T_i - T_o) \quad \text{..... (5.10)}$$

$$q_L = (\dot{V}/v)\ (W_i - W_o)\ h_{fg} = 3010\ \dot{V}\ (W_i - W_o) \quad \text{..... (5.11)}$$

$\dot{V}$: ventilation volume flow rate (m^3/s)

Example:

An auditorium seats 1000 people. The space design conditions are 21°C and 40% RH, and outdoor design conditions 5°C DBT and 60% RH. What is the heating load due to ventilation.

Solution:

From table 4.8 the minimum ventilation air per person to be 5.5 l/s

Total ventilation air = 5.5 x 1000 = 5500 l/s = 5.5 m^3/s

$$q_s = (\dot{V}/v)\ C_p\ (T_i - T_o) = 1.22\ \dot{V}\ (T_i - T_o) = 1.22\ (5.5)(21 - 5)$$

W_i = 0.0061 kg/kg dry air , W_o= 0.0034 kg/kg dry air [from the psychrometric chart at the inner and outer design conditions]

$q_L = (V^{\cdot}/v)(W_i - W_o)\, h_{fg} = 3010\, V^{\cdot}(W_i - W_o)$

$= 3010\,(5.5)(0.0061 - 0.0034) = 44.7$ kW

Then the total heating load due to ventilation

$Q_v = 107.36 + 44.7 = 152.06$ kW

Heat Losses from Air Ducts

The losses of a duct system can be considerable when the ducts are not in the conditioned space. Proper insulation will reduce these losses but cannot completely eliminate them. The losses may be estimated using the following relation

$q_D = U.A_s.\Delta T_m$ …… (5.12)

where

U: overall heat transfer coefficient (W/m^{2}ºC)

A_s: outside duct surface area (m^2)

ΔT_m: mean temperature difference between the air in the duct and the environment.

Example:

Estimate the heat losses from 0.5 m^3/s of air at 50ºC round duct 8 m in length. The duct has 25 mm (1 in) of fibrous glass insulation and the overall heat transfer coefficient is 1.1357 W/m^2.ºC. The environment temperature is -10ºC and the duct diameter is 400 mm.

Solution:

Equation 5.12 will be used to estimate the heat losses

$\Delta T_m = T_s - T_o = 50 - (-10) = 60$ºC

The surface area of the duct is

$A_s = \pi.D.L = \pi\,(0.4 + 0.05)(8) = 11.3$ m^2

Then, $q_D = 1.1357\,(11.3)(60) = 770$ W $= 0.77$ kW

the temperature of air leaving the duct may be computed from

$q = m^{\cdot} C_p (T_2 - T_1) = V^{\cdot} \rho\, C_p (T_2 - T_1)$

$T_2 = T_1 + q/(V^{\cdot} \rho\, C_p) = 50 + (-0.77/0.5 \times 1.2 \times 1.017) = 48.7$ºC

Minimum insulation of supply and return duct is presently specified by ASHRAE standards as follows:

All duct system shall be insulated to provide a thermal resistance, excluding film resistance, of

$R = \Delta T/47.3$ ($m^2 {}^oC/W$)

Where, $\Delta T = T_{duct} - T_{surrounding}$

Heat losses from the supply ducts become part of the space heating load and should be summed with transmission and infiltration heat losses.

Heat losses from the return ducts are not a part of the space heat loss, but should added to the heating equipment load.

Air Required for Space Heating

The air quantity is computed from

$q_s = \dot{m} C_p (T_s - T_r) = 1.22 \dot{V} (T_s - T_r)$ ……. (5.13)

$\dot{V} = q_s/[1.22(T_s - T_r)]$

Where

$\dot{V}$: volume flow rate of supplied air, m^3/s

v: specific volume of supplied air, m^3/kg

T_s: temperature of supplied air, ^oC

T_r: temperature of room (conditioned space), ^oC

The temperature difference ($T_s - T_r$) is normally less than 10^oC. It can be considered as a temperature difference about 6^oC is suitable for most of the comfort air conditioning applications in Arab region.

After the total air flow rate required for the complete structure has been determine. The next step is to allocate the correct portion of the air to each room or space. Air quantity for each room should be apportioned according to the heating load for the space; therefore

$\dot{V}_{rn} = \dot{V}(q_{rn}/q_t)$ ……. (5.14)

Where; q_{rn}: total heat loss of room(n), W

$\dot{V}_{rn}$: volume flow rate of air supply to room(n), m^3/s

Exercises:

1. The dimensions of the outer wall of a room are 6 m x 3 m, and it contains a glass window of dimensions 2 m x 1.5 m. The wall is made up of 24 cm brick with 10 cm face brick and 2 cm of a gypsum layer from the inside. The indoor air temperature is 20°C, while the outdoor air temperature and wind velocity are 5°C and 24 km/h respectively. Calculate:

(a) The total heat loss from the wall and the window together.

(b) The temperature of the inner surface of the wall.

2. An external wall is made of 100 mm common brick, with 40 mm gypsum plaster. What is the thickness of rock wool insulation (K= 0.04 W/m.^{o}C) to reduce the rate of heat loss by 80%.

3. The room conditions in winter are 22°C DBT and 40% RH. Infiltration air at 1°C DBT and 50% RH enters the room at a rate of 0.006 m^3/s.

(a) Calculate the rate of water that must be evaporating to maintain the room conditions.

(b) Calculate the amount of heat required to vaporize the water at the above rate.

4. A building wall consists of 25 cm concrete (K= 1.75 W/m.K) and 1.9 cm plaster (K= 87 W/m.K) on the inside surface. The outside and inside surface heat-transfer coefficients can be taken as 34 and 9.4 W/m^2K respectively. The outside temperature is -18°C , while the room is held at 23.5^{o}C DBT and 16.8^{o}C WBT.

(a) What is the temperature on the inside wall surface.

(b) Will the moisture condense on the wall.

(c) How many layers of 1.25 cm thick fiber-board insulation (K= 0.048 W/m.K) should be applied on the inside wall surface to prevent moisture to condense on it.

5. A residential house contains a reception room and a family room and two-bedroom. The heating load of the listed rooms are; 5.5 kW, 3.5 kW and 4.2 kW for each bedroom respectively. The inside required temperature is 21°C. Find the rates of hot air which required for every room.

Chapter Six

Cooling Load Calculation

A large number of variables are considered in making cooling load calculations than heating load calculations. In both situations the actual heat loss or gain is a transient one. In design for cooling, however, transient analysis must be used if satisfactory results are to be obtained. This is because the instantaneous heat gain into a conditioned space is quite variable with time, primarily because of the strong transient effect created by the hourly variation in solar radiation.

6.1 Heating Gain and Cooling Load

It is important to differentiating between heat gains and cooling load. Heat gain is the rate at which energy is transferred to or generated within a space. Heat gains usually occur in the following forms:

a. Solar radiation through openings.

b. Heat conduction through boundaries with convection and radiation from the inner surface into the space.

c. Sensible heat convection and radiation from internal objects.

d. Ventilation (outside) and infiltration air.

e. Latent heat gain generated within the space.

Figure 6.1 shows the heat gain sources in summer.

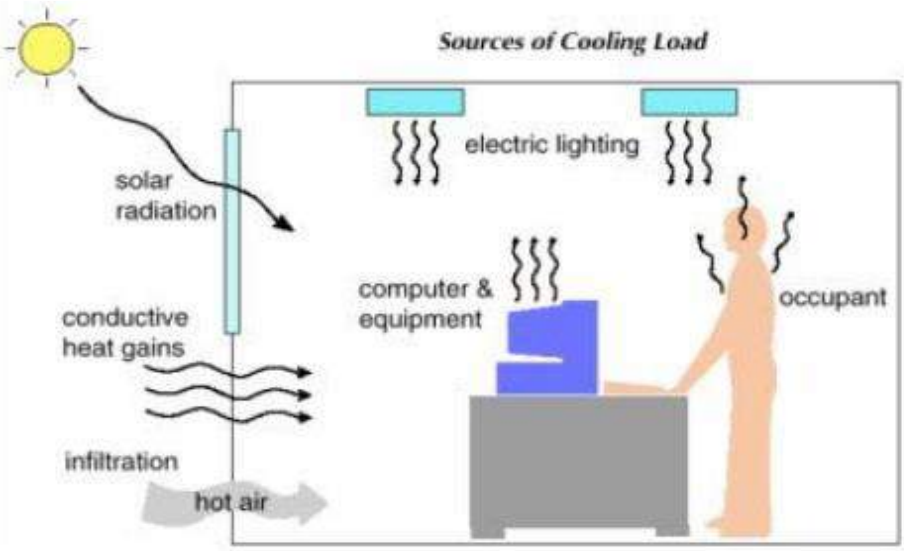


Fig. 6.1 Sources of heat gain.

The space cooling load is the rate at which heat must remove from a space to maintain the temperature and humidity at the design values. The space cooling load will generally differ from the space heat gain at any instant of time.

The heat storage characteristics of the structure and interior objects determine the thermal lag and therefor the relationship between heat gain and cooling load.

Figure 6.2 shows the relation between the heat gain and cooling load and the effect of the mass of the structure. The attenuation and delay of the peak load gain is very evident especially for heavy construction.

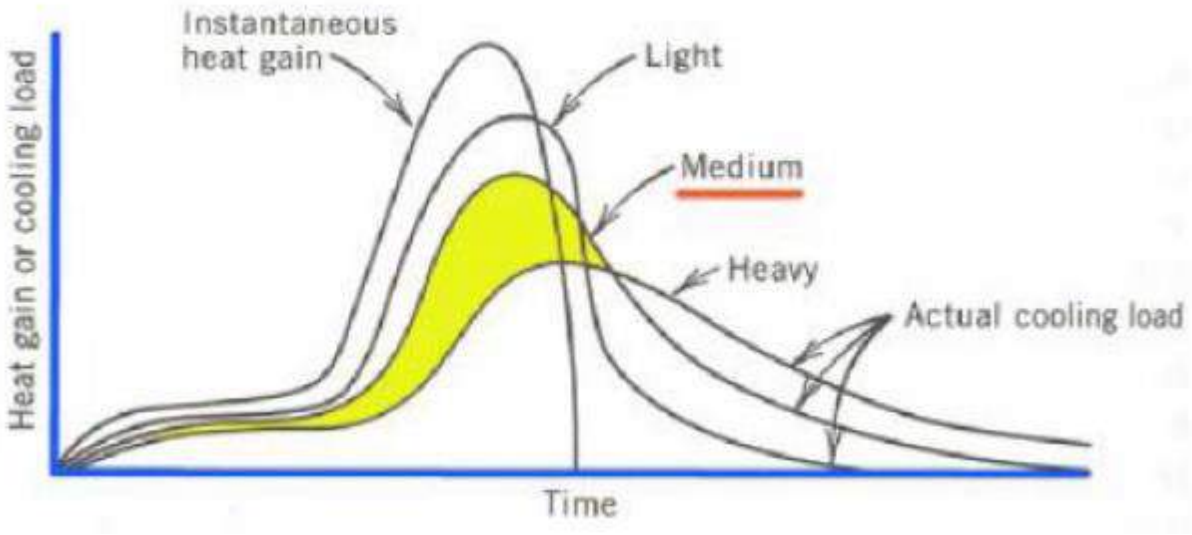


Figure 6.2 Relation between heat gain and cooling load.

6.2 Solar Air Temperature

In the first instance, heat transfer through a wall depends on the rate at which heat enters its outer surface. The concept of 'Sol-air temperature' has been in use for some time as an aid to the determination of the initial rate of entry of heat. It is defined as the value of the outside air temperature which would, in the absence of all radiation exchanges, give the same rate of heat flow into the outer surface of the wall as the actual combination of temperature differences and radiation exchanges really does.

The sol-air temperature, t_{eo}, otherwise termed the outside environmental temperature, is used in the following equation

$$Q' = h_{so}(t_{eo} - t_{so}) \quad \text{---------- (6.1)}$$

where Q' = rate of heat entry into the outer surface, in W m^{-2}
h_{so} = outside surface heat transfer coefficient, in W m^{-2} K^{-1}
t_{eo} = sol-air temperature, in °C
t_{so} = outside surface temperature, in °C

Q' can be expressed in another way which does not involve the use of the sol-air temperature:

$$Q' = \alpha I_\delta + \alpha' I_s + h_{so}(t_o - t_{so}) + R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6.2)$$

In this basic heat entry equation, α and α' are the absorption coefficients (usually about the same value) for direct, I_δ, and scattered, I_s, radiation which is normally incident on the wall surface. R is a remainder term which covers the complicated long wavelength heat exchanges by radiation between the wall and nearby surfaces. The value of R is difficult to assess; in all probability it is quite small and, if neglected, results in little error.

Equations (6.1) and (6.2) can be combined to yield an expression for $\mathbf{t_{eo}}$ in useful form:

$$t_{eo} = t_o + \frac{\alpha I_\delta + \alpha' I_s + R}{h_{so}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6.3)$$

If α' is made equal to α, and R is ignored, this expression becomes

$$t_{eo} = t_o + \frac{\alpha(I_\delta + I_s)}{h_{so}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6.4)$$

6.3 Cooling Load Calculation Methods

For a thorough calculation of the zones and whole-building loads, one of the following three methods should be employed:

a. Transfer Function Method (TFM): This is the most complex of the methods proposed by ASHRAE and requires the use of a computer program or advanced spreadsheet.

b. Cooling Load Temperature Differential/Cooling Load Factors (CLTD/CLF): This method is derived from the TFM method and uses tabulated data to simplify the calculation process. The method can be fairly easily transferred into simple spreadsheet programs but has some limitations due to the use of tabulated data.

c. Total Equivalent Temperature Differential/Time-Averaging (TETD/TA): This was the preferred method for hand or simple spreadsheet calculation before the introduction of the CLTD/CLF method.

These three methods are well documented in ASHRAE Handbook Fundamentals, 2001.

The CLTD Method

The CLTD method accounts for the thermal response in the heat transfer through the wall or roof, as well as the response due to radiation of part of the energy from the interior surface of the wall to objects and surfaces within the space. The CLTD method makes use of (a) the temperature difference in the case of walls and roofs and (b) the cooling load factors (CLF) in the case of solar heat gain through windows and internal heat sources, that is,

$$Q = U \times A \times CLTD_c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6.5)$$

Where

Q: is the net room conduction heat gain through roof, wall or glass (W)

A: is the area of roof, wall or glass (m^2)

U: is the overall heat transfer coefficient ($W/m^2.K$)

$CLTD_c$: is the cooling load temperature difference (^oC)

CLTD/CLF calculation

- **Walls and roofs**

To account for the temperature and the solar variations, the concept of cooling load temperature difference (CLTD) is introduced. The CLTD is a steady-state representation of the complex heat transfer involving actual temperature difference between indoors and outdoors, mass and solar radiation by the building materials, and of time of day. Table 6.1 lists the types of the walls according to installation structural, while Table 6.2 gives the values of CLTDs for different groups of sunlit walls. Table 6.3 lists the CLTDs values for thirteen type of roofs for the typical cooling design day. The following relation makes corrections in the CLTDs listed in the Tables 6.2 and 6.3 for walls and roofs respectively for deviations in design and solar conditions as follows:

$$CLTD_c = [(CLTD + LM)k + (25.5 - T_r) + (T_{o.m} - 29.4)]f \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6.6)$$

Where

$CLTD_c$: is the corrected value of CLTD

LM: is latitude-month correction from Table (6.4)

K: is a color adjustment for light-coloured roof (1.0 for dark coloured roof; 0.5 if permanently light coloured)

T_r: is the design room temperature

$T_{o.m}$: is the average outdoor temperature, computed as the design temperature less half the daily range.

f: is attic fan factor (1.0 for no attic fans; 0.75 for positive attic ventilation).

Example 6.1:

Calculate the $CLTD_c$ for a wall of group D facing to the south and located at the latitude of 32°N, at 2:00 P.M. in the month of October. Assume that the wall is dark color, and the indoor design temperature is 25.5°C. The maximum outdoor temperature is 35°C with the daily range of 11.2°C.

Solution:

From Table 6.2, and for wall of group D, at solar time 14 and south direction;

The value of CLTD = 9°C

The value of corrected CLTD can be calculate from equation 6.6

$$CLTD_c = (CLTD + LM)k + (25.5 - T_r) + (T_{o.m} - 29.4)$$

$$= (9 + 6.1) \times 1.0 + (25.5 - 25.5) + (29.4 - 29.4) = 15.1^oC$$

Example 6.2:

Roof of one of the buildings consists of 102 mm high weight concrete with 50.8 mm insulation, and a suspended ceiling. The overall heat transfer coefficient of the roof is 0.511 $W/m^2.^oC$. The building is located at latitude of 40°N. The outdoor design conditions are 36°C DBT and 26°C WBT, with daily range of 12°C, and the indoor design conditions are 24°C DBT and 50% relative humidity. Compute the cooling load per square meter of the roof at 4:00 P.M. in the month of August.

Solution:

Roof No. 9, U= 0.511 $W/m^2.^oC$, 40°N

CLTD = 19°C, LM = 1.6, $T_{o.m} = 36 - (12/2) = 30^oC$

$$CLTD_c = [(19 + 1.6) \times 1 + (25.5 - 24) + (30 - 29.4)] \times 1 = 22.7^oC$$

$Q/A = U.CLTD_c = 0.511 \times 22.7 = 11.6\ W/m^2$

- **Windows and glass**

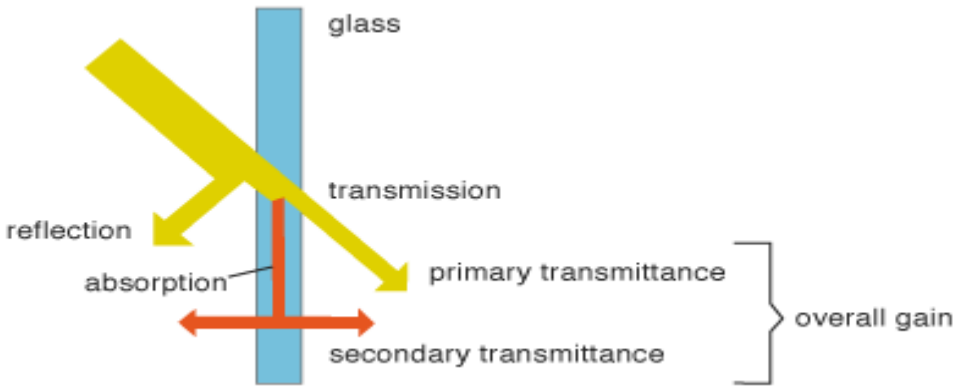


Figure 6.3 Absorption, reflection and transmission of solar radiation in glass.

When solar radiation strikes an unshaded window (Fig.6.3), about 8% of the radiant energy is typically reflected back outdoors, from 5-50% is absorbed within the glass, and the remainder is transmitted directly indoors, to become part of the cooling load.

The solar gain is the sum of the transmitted radiation and the portion of the absorbed radiation the flows inward. Because heat is also conducted through the glass wherever there is an outdoor- indoor temperature difference, the total of heat admission is

Total heat admission through glass = Radiation transmitted through glass + Inward flow of absorbed solar radiation + Conduction heat gain

We can rewrite the above equation

Total heat gain = Solar heat gain + Conduction heat gain ……. (6.7)

Conduction heat gain = $U_g.A_g.CLTD_c$ …….. (6.8)

where

U_g: overall heat transfer coefficient of the glass

A_g: Area of the glass

$CLTD_c$: Correct cooling load temperature difference for glass. Table 6.5 gives CLTD.

The correction of the CLTD is as follow;

$CLTD_c = CLTD + (25.5 – T_r) + (T_{o.m} – 29.4)$ …….. (6.9)

Solar heat gain = A_g. [SHG_{max} x SC x CLF]…… (6.10)

Where

SHG_{max}: Maximum solar heat gain in W/m^2 from Table 6.6

SC: Shading coefficient from Table 6.7, 6.8, 6.9, and 6.10

CLF: Cooling load factor for glass from Table 6.11 and 6.12

Shading coefficient (SC); is the ratio between the solar heat gain through any given type of fenestration system and the solar heat gain through unshaded clear glass (reference glass)

SC = (Solar heat gain of fenestration)/ (Solar heat gain of double-strength glass)… (6.11)

Example 6.3:

The wall of Example 6.1 has a 1.2 m x 1.5 m single glass window (U = 4.6 $W/m^2.^oC$). The window has light-colored venetian blinds. Compute the cooling load due to the window at 5:00 P.M. solar time for August, using the design condition given in Example 6.1.

Solution:

The total cooling load for the window can be determined as following;

$Q_g = A_g[U \times CLTD_c + SC \times SHG_{max} \times CLF]$

$CLTD_c = CLTD + (25.5 – T_r) + (T_{o.m} – 29.4) = 7 + (25.5 - 25.5) + (29.4 – 29.4) = 7^oC$

SC = 0.55 from Table 6.8

$SHG_{max} = 350\ W/m^2$ from Table 6.6

CLF = 0.27 from Table 6.12

$Q_g = 1.8[4.6 \times 7 + 0.55 \times 350 \times 0.27]$

= 151.5 W

- **Space cooling load through partitions**

Whenever a conditioned space in adjacent to a space with a difference temperature, transfer of heat through the separating physical section must be considered

$q_p = U \times A \times (T_a - T_r)$ …….. (6.11)

Where

U: overall heat transfer coefficient of the partition ($W/m^2.^oC$)

A: area of partition (m^2)

T_a: temperature of the adjacent space (^oC)

T_r: temperature of the conditioned space (^oC)

- **Infiltration heat gain**

The infiltration load should be considered in the space heat gain calculations. Sensible and latent heat from infiltration gains can be calculated in the same way that already explained in Chapter Five with some simple difference.

Sensible infiltration heat gain, $q_{s.i} = 1.22 \times \dot{V}_i \times (T_o - T_r)$ ……. (6.12)

Latent infiltration heat gain, $q_{l.i} = 3010 \times \dot{V}_i \times (W_o - W_i)$……. (6.13)

Where

$\dot{V}_i$: is the infiltration volume flow rate (m^3/s) or (l/s)

Heat Gain from Internal Sources

Internal heat comprises sensible and latent heat gains from occupants, lights, appliances and equipment and piping, etc.

- **Occupancy**

The people who occupy the building give off thermal energy continuously, the rate of which depends on the level of activity (Table 6.13). For the sensible portion of the heat released, a cooling load factor (Table 6.14) has been developed to account for the lag in time

between occupancy and the observed cooling load. The sensible cooling load due to people is therefore,

$$q_s = N \times G_s \times CLF_s \quad \text{..... (6.14)}$$

Where

q_s = sensible cooling load due to occupants (W)

N = number of occupants

G_s = sensible heat gain depending on activity and time from entry (W)

CLF_s = cooling load factor (dimensionless) for people.

The latent heat gain from occupants is found by

$$q_L = N \times G_L \quad \text{..... (6.15)}$$

Where

q_L: latent heat gain from occupants (W)

N: number of occupants

G_L: latent heat gain from occupants depending on a activity and time from entry (W)

Example 6.4:

An office suite is designed with 15 people. Estimate the cooling load from the occupants after 7 hours of their entering the office. Also, assumed that the occupants stay in the space are 9 hours.

Solution:

We will assume moderately active, office work, and use data from Table 6.13 and 6.14

q_L = 55 W/person, q_s = 75 W/person, CLF_s = 0.825

$q_L = 15 \times 55 = 825$ W

$q_s = 15 \times 75 \times 0.825 = 928$ W

$Q_P = q_L + q_s = 825 + 928 = 1753$ W = 1.753 kW

- **Lighting**

Lighting is often the major space cooling load component. The rate of heat gain at any instant, however, is not the same as the heat equivalent of power supplied instantaneously to these lights. Only part of the energy from lights is transferred to the room air by convection, and thus becomes the cooling load. The remaining portion is the radiant heat that affects the conditioned space only after having been absorbed by walls, floors furniture, etc. and released after a time lag. The instantaneous heat gain for lights may be expressed as

$$q_L = W \times F_u \times F_s \quad \text{.....} \quad (6.16)$$

Where

q_L: instantaneous heat gain for lights

W: summation of all installed light wattage

F_u: use factor- ratio of wattage in use to that installed

F_s: special allowance factor for lights, for fluorescent lamp $F_s = 1.2$

The cooling load is then given by

$$Q_L = q_L \times (CLF)_L \quad \text{.....} \quad (6.17)$$

The cooling load factor is a function of the building mass, air-circulation rate, type of fixture and time. Table 6.15 give the cooling load as a function of time for lights that are on for 8, 10, 12 and 14 hour. The "a" classification depends on the nature of light fixture, the return-air system, and the type of furnishings, where the "b" classification depends on the construction of the building and the type of supply and return air system. Design values of coefficients "a" and "b" are given in Tables 6.16 and 6.17 respectively.

Example 6.5:

The office suite of example 6.4 has total installed light wattage of 8400 W. The fluorescent light fixtures are recessed with 40 W lamps. Supply air is through the ceiling with air returning through the ceiling plenum. The lights are turned on at 8:00 A.M. and turned off at 6:00 P.M. Estimate the cooling load at 4:00 P.M. The floor is 75 mm concrete.

Solution:

Assuming that about 15% of the lights are off, the use factor; $F_u = 85\%$, $F_s = 1.2$

The instantaneous heat gain for lights is

$q_L = W \times F_u \times F_s = 8400 \times 0.85 \times 1.2 = 8568$ W

The cooling load is then

$Q_L = q_L \times (CLF)_L$

The $(CLF)_L$ from Table 6.15 for 10 hours on and a = 0.55 (Table 6.16), b = B (Table 6.17)

$(CLF)_L = 0.82$

$Q_L = 8568 \times 0.82 = 7026$ W

- **Miscellaneous Equipment**

Most appliances contribute both sensible and latent heats. The latent heat produced depends on the function the appliances perform, such as drying, cooking, etc. Gas appliances produce additional moisture as product of combustion.

The heat gain from Equipment Q_M is

$Q_{M.S} = q_{M.S} \times CLF_M$ ……. (6.18)

Where

$q_{M.S}$: sensible heat gain from appliances Table 6.18

CLF_M: cooling load factor for appliances Table 6.19

$Q_{M.L} = q_{M.L} \times$ No. of appliances …… (6.19)

Where

$q_{M.L}$: latent heat gain from appliances Table 6.18

- **Heat Gain from Ventilation Air**

Provision of ventilation air is mandated by local codes and ordinances. The ASHRAE standard 62 recommends minimum ventilation rates for most common applications. For general applications, such as offices, 10 l/s per person is recommended. Ventilation air is generally introduced at the air handling unit (AHU) rather than directly into the conditioned space. It thus becomes a cooling coil load component instead of a space load component.

Heat gain corresponding to a flow rate of $\dot{V}$ through an enthalpy difference of Δh (for an air density of 1.20 kg/m^3) is shown below:

Sensible heat gain corresponding to the change in dry bulb temperature ΔT for a given air flow $\dot{V}$ is

$q_s = 1.22 \times \dot{V} \times \Delta T$ ……..…… (6.20)

where

q_s: is the sensible heat gain (kW)

$\dot{V}$: is the ventilation flow rate (m^3/s)

$\Delta T = (T_o - Tr)$

Latent heat gain corresponding to the change in moisture content (ΔW) for a given air flow $\dot{V}$ is

$q_L = 3010 \times \dot{V} \times \Delta W$ ……. (6.21)

where

q_L: is the latent heat gain

$\Delta W = (W_o - W_r)$

- **<u>Required Air Quantity</u>**

The air quantity required to offset simultaneously the room sensible and latent loads may be calculated using the following equation

Air flow required $\dot{V}_R$ = room (zone) sensible heat/ 1.22 $(T_r - T_s)$ …… (6.22)

where

T_r: is the room temperature (^oC)

T_s: is the supply air temperature (^oC)

The problem is how we can calculate or determine the supply air temperature T_s. One of the two unknown ($\dot{V}_R$, T_s) is chosen according to "good practice" (such as costs and job conditions), and the remaining unknown is then calculated from the equation.

Example 6.6:

A hair salon shop has a sensible cooling load of 16 kW and latent cooling load of 6.5 kW. The room conditions are to be maintained at 25°C DBT and 50% RH. If 56 m^3/min of supply air is furnished, determine the required supply air DBT and WBT.

Solution:

$q_s = 1.22 \times \dot{V} \times (T_r - T_s)$

$16 = 1.22 \times (56/60) \times (25 - T_s)$

$T_s = 11°C$ (DBT of the supply air)

$q_L = 3010 \times \dot{V} \times (W_r - W_s)$

$6.5 = 3010 \times (56/60) \times (0.01 - W_s)$

$W_s = 0.00232$ kg/kg dry air (moisture content of the supply air)

Now from the psychrometric chart at DBT=11 °C and W= 0.00232 kg/kg, we find WBT=9°C.

- General Design Guidelines

The general procedure required to calculate the space cooling load is as follows:

(a) *Building configuration and characteristics*: Determine the building location, orientation and external shading, building materials, external surface color and shape. These details are usually obtained from building plans and specifications.

(b) *Outdoor design conditions*: Obtain the outdoor weather data for the building location and select the outdoor design conditions.

(c) *Indoor design conditions*: Specify temperature, humidity, air velocity, etc.

(d) *Operating schedules*: Obtain a schedule of lighting, occupancy, internal equipment, appliances and processes generating heat load.

(e) *Date and time*: Select the time of the day and month to estimate the cooling load.

Several different times of the day and several different months need to be analysed to determine the peak load time. The particular day and month are often dictated by peak solar conditions. A calculation form [Table 6.20] is given blow, wherein the calculated values of the components of cooling load may be entered.

Exercises

1. A window in a south wall in a building which has a light construction. The dimensions of the window are (1.5 m x 1.5 m) and the glass is of heat absorbing type (U= 5.91 W/m^2.^{o}C) and thickness of 6 mm. The building is located at latitude of 32°N. Outdoor design condition is 42°C, with daily range of 14°C, and the indoor design temperature is 25°C. Calculate the cooling load from the window at 3:00 P.M. in the month of July.

2. Roof of one of the building consists of 152.4 mm high weight concert with 50.8 mm insulation, and without suspended ceiling. The overall heat transfer coefficient of the roof is 0.664 W/m^2.^{o}C, and the building is located at latitude of 48°N. The outdoor design condition is 38°C DBT with daily range of 11°C, and the indoor design condition is 25°C DBT and 50% relative humidity. Calculate the cooling load per square meter of the roof at 2:00 P.M. in the month of July.

3. Hall meetings can accommodate for 100 person located in Mosul city

(a) Choose the appropriate outdoor and indoor design conditions for the summer.

(b) The sensible heat and the latent heat gains from the occupants of the hall.

(c) The amount of air ventilation required to this hall, and the cooling load of this amount.

4. The west wall in a building in Baghdad (32°N) has a window of the dimensions (1.0 m x 2.0 m). The glass is of heat absorbing type (U= 4.6 W/m^2.^{o}C) and thickness of 6 mm. Curtains type (III_D) was used. Outdoor design condition is 34°C DBT, with daily range of 11°C, and indoor design temperature is 24°C. Calculate the cooling load from the window at 4:00 P.M. in the month of August.

5. A window in a southern west wall in a building which has a medium construction. The dimensions of the window are (2.0 m x 1.5 m), and the glass is of the clear type (U= 5.91 W/m^2.^{o}C) and thickness of 6 mm. The glass was shaded from the outside. The building is located at latitude of 36°N. Outdoor design conditions are 35°C DBT and 50% RH, with the daily range of 11.5^{o}C, and the indoor design conditions are 26°C DBT and 18°C WBT. Compute the cooling load from the window at 5:00 P.M. in the month of September.

6. An air-conditioning system is to be design for a restaurant with the following data:

Outside design conditions: 40°C DBT, 28°C WBT

Inside design conditions: 25°C DBT, 50% RH

Solar heat gain through walls, roofs and floor: 5.87 kW

Solar heat gain through glass: 5.52 kW

Occupant: 25

Sensible heat gain per person: 58 W, Latent heat gain per person: 58 W

Internal lighting load: 15 lamps of 100 W, 10 fluorescent of 80 W

Sensible heat gain from other sources: 11.63 kW

Rate of Infiltration air: 15 m^3/min

If 25% fresh air and 75% recirculate air is mixed and passed through the conditioner coil, find:

(a) The amount of total air required in m^3/min.

(b) The apparatus dew point temperature of the coil.

(c) The condition of the supply air to the room.

(d) The capacity of the conditioning plant.

Assume the by –pass factor equal to 0.2. Draw the schematic diagram of the system and show the system on psychrometric chart and insert the temperature and enthalpy values at salient points.

Air Conditioning Systems and Equipment

10.1 INTRODUCTION

Heat always travels from a warmer to a cooler area. In summer, hot outside air continuously enters buildings that have lower temperature. To maintain the room air at a comfortable temperature, this excess heat must be continuously removed from the room. The equipment that removes this heat is called a *cooling system*.

In winter, there is continuous heat loss from room to the outdoors. If the air in the room is to be maintained at a comfortable temperature, heat must be continuously supplied to the air in the rooms. The equipment that supplies the heat required is called a *heating system*.

An air conditioning system may provide heating, cooling or both. Its size and complexity may range from a window unit for a small room to a huge system for a complex building, yet the basic principles are the same. Most heating and cooling systems have at least the following basic components:

1. A cooling source that removes heat from the fluid (air or water).
2. A heating source that adds heat to the fluid (air, water or steam).
3. Air distribution system (a network of ducts or piping) to carry the fluid to the rooms to be heated or cooled.
4. Equipment (fans or pumps) for moving the air or water.
5. Devices (e.g. radiation) for transferring heat between the fluid and the room.

Other components included are automatic controls, safety devices, valves, dampers, insulation, and sound and vibration reduction devices.

Air conditioning systems that use water as the heating or cooling fluid are called all-water or *hydronic systems*; those that use air are called all *air systems*. A system which uses both air and water is called a combination of *air and water system*.

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Air conditioning systems

An air conditioning system should satisfy the need of an occupant/user at the most economical cost. The selection of the system depends upon many factors:

Customer's objectives: It could be only a relief in temperature or complete control of environment. Fresh air and air quality requirement are not given much importance in air conditioning systems of a theatre or auditorium compared to the hospital air conditioning units.

Economics: Every effort should be made in all air conditioning systems to achieve the comfort conditions with minimum energy inputs.

Occupancy: Single purpose occupancy means all occupants have the same purpose in one or more spaces. Multipurpose occupancy may need a complex system.

Thermal load: Multiplexes and supermarket buildings, these days, are provided with air conditioning systems to meet the comfort conditions. During the planning and construction of such buildings, design engineers have to find ways to minimize heat gain through external sources, especially solar radiations. The feasibility of reducing the thermal load by choosing construction options or precooling can be decisive in the design of the building.

Internal environment: Level of cleanliness, acoustics and concentration of load within the space may affect the selection of the air conditioning system.

There are a large number of variations in the types of air conditioning systems and the way they can be used to achieve the required environment in the buildings. In every installation of an air conditioning system, the engineer/contractor has to look for the best choice.

10.2 CLASSIFICATION OF AIR CONDITIONING SYSTEMS

Air conditioning systems can be classified in a number of ways. Some of these ways are as follows.

The cooling/heating fluid used

(i) **All-air systems:** These systems use only air as a heating or cooling medium.

(ii) **All-water (hydronic) systems:** These use only water for both cooling and heating purposes.

(iii) **Air-water combination systems:** Such systems use both water and air for cooling and heating purposes.

Single zone or multiple zone systems

A single zone air conditioning system can satisfy the air conditioning needs of a single zone. A multiple zone air conditioning system can satisfy the needs of air conditioning of many zones.

Unitary or central systems

Such air conditioning systems can be broadly classified into two types.

Unitary system: These systems use packaged equipment. The units consisting of fans, coils and refrigeration equipment are assembled as one unit in a factory and fitted at site. Mass production of such units with good quality is possible. Examples of the unitary systems are window air conditioners and split air conditioners.

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Central or built-up systems: The components are manufactured separately and assembled and installed at site. The systems at higher capacities (> 60 TR) are suitable and more economical in initial cost as well as in running cost.

The year round air conditioning in all seasons is possible with the central systems. There are a large number of options available in these systems.

Let us now study the above common systems in more detail, beginning with the unitary system.

10.3 UNITARY SYSTEM

Such a unit is designed to be installed in or near the conditioned space. The components are contained in the unit. Unitary systems are standardized for certain applications but minor modifications are possible to suit an application. Heating components are rarely included.

10.3.1 Window Air Conditioner

The refrigeration system components—compressor, condenser, capillary tube and evaporator—are connected through copper tubes. The evaporator and condenser are at two ends such that the evaporator part is inside the room while the condenser is outside the room as shown in Figure 10.1. There is an insulated cabinet around the cooling coil with two compartments. A blower is fitted in this cabinet behind the evaporator coil which pumps air into the upper compartment. The blower pulls the room air through the cooling coil and through the filter fitted on the face of the coil. This air is then discharged back to the room through the upper compartment.

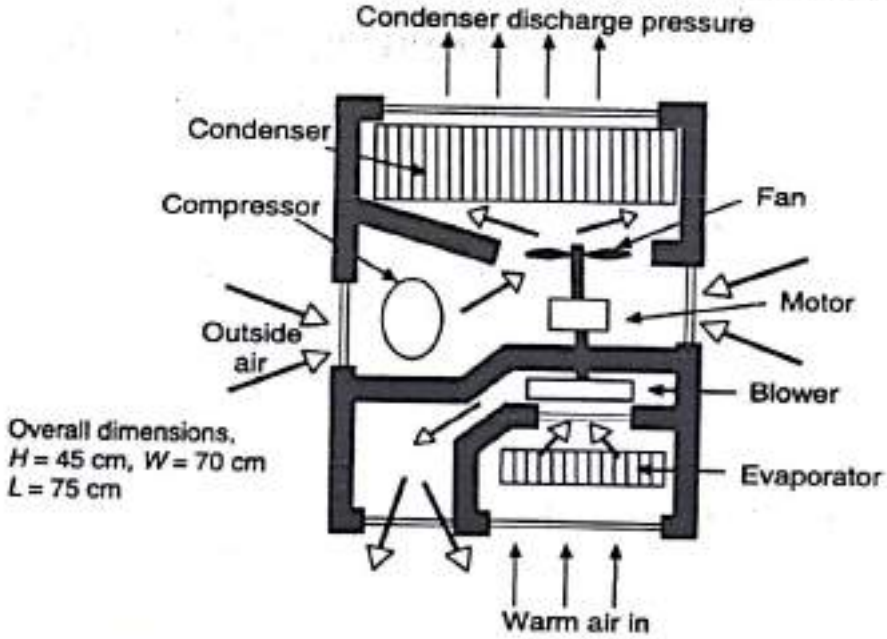

Figure 10.1 Schematic view of a typical room air conditioner.

A fan draws air from the sides and throws over the condenser coil. This helps condenser to reject heat outside the room. The control panel has three knobs. One controls the speed of the blower motor to give high cool or low cool. The second knob is of a thermostat, the bulb of which is placed at the filter to sense the temperature of the room air being sucked in by the blower. This allows the user to set the room temperature. The third knob operates a flap in the insulated cabinet to allow ventilation air supply. Refrigerant R22 is employed in this unit.

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Window air conditioners are available in capacities ranging from 0.8 TR to 5 TR. One-and-half TR window air conditioner, whose height is 45 cm, width 70 cm, and depth 75 cm, is most commonly used for commercial applications. The maximum size of the window AC is limited to 5 TR due to the available capacity of the hermetic compressor. Companies such as Blue Star, Videocon, Carrier Aircon, manufacture window AC units.

In a real sense, the window AC is not an air conditioner. To a certain extent, dehumidification is possible in it while there is no provision for humidification. It is especially suitable for use in cities like Mumbai, Kolkata and Chennai where RH is more than 50%.

Limitations of window AC

(i) No humidity control though it carries out dehumidification.
(ii) Most of the window air conditioners do not provide heating for winters.
(iii) No provision for humidification in window air conditioner.
(iv) Outside temperature above 40°C can cause derating of the conditioner.

Precautions to be taken while installing a window AC

(i) It should be fitted with a small slope (3° to 5°) downwards, towards the outside which ensures that draining of condensate at the cooling coil is outside the room.
(ii) It should be ensured that the condenser is not exposed to direct solar heat to prevent undue rise in condenser pressure.
(iii) The gaps between the wall opening and the package should be blocked by insulation.

10.3.2 Split Air Conditioner

Basically it is a package unit like window AC. It splits the window air conditioner into two parts with evaporator placed inside the conditioned room while assembly of other components is placed outside the room. So it has a fan coil unit fitted inside the room and a condensing unit with an additional fan installed outside. The two units are connected by a suction line and a liquid line. In some cases, capillary is inside the condensing unit and low-pressure liquid is supplied through an insulated line to the fan coil unit.

The noise generation in a window AC is mainly due to the compressor unit which is outside in a split unit. So the split air conditioner ensures low noise level in the room. The fan coil unit has greater air throw than that of a window air conditioner. A split AC unit consumes more energy compared to window AC of the same capacity due to two reasons:

(i) There are two motors to drive two fans, one in condensing unit and another in fan coil unit.
(ii) Refrigerant flow lines are longer, so more pressure drops resulting in higher compressor power requirement.

These units are readily available in the range of 0.8 to 4 TR.

10.4 CENTRAL AIR CONDITIONING SYSTEMS

A central air conditioning system can be used for single-zone (a zone consisting of a single room or group of rooms) or multizone applications. In this section a central AC system, all-air for a single-zone application is discussed and the system is shown in Figure 10.2.

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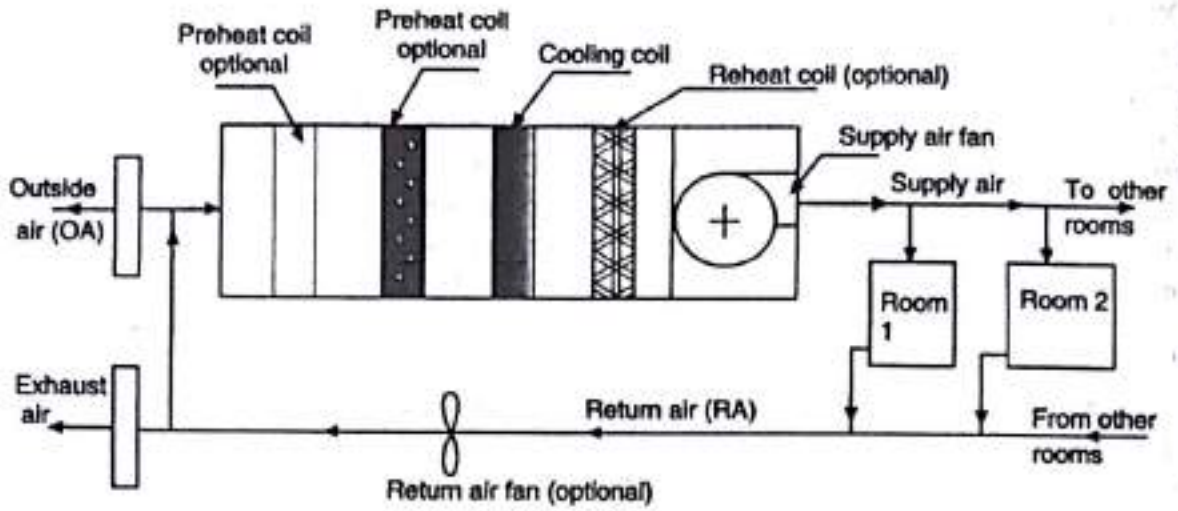

Figure 10.2 Single-zone central air conditioning system.

A single-zone air conditioning system has one thermostat that automatically controls one heating or cooling unit to maintain proper temperature in a zone comprising a single room or a group of rooms. A window air conditioner is an example of a single-zone air conditioning unit.

The system shown in Figure 10.2 is for year-round air conditioning to control both temperature and humidity. All the components shown in the figure may not be utilized in all the circumstances.

An air-handling unit (AHU) cools or heats air that is then distributed to the single zone. The supply air fan is necessary to distribute air through the ductwork to the rooms.

(i) **Cooling coil:** It cools and dehumidifies the air and provides humidity control in summer. Reheat coil is optional and is used when air temperature is to be maintained at the required level, especially in winter. In summer, it may remain idle.

(ii) **Reheating coil:** It heats the cooled air when the room heat gain is less than the maximum, thus providing humidity control in summer. The coil capacity is such that it satisfies the heating needs during winter.

(iii) **Ductwork:** It is arranged so that the system takes in some outside ventilation air (OA), the rest being return air (RA) recirculated from the rooms. The equivalent amount of outside air must then be exhausted from the building. Dampers are provided to vary the rate of ventilation air as per the requirement of fresh air in the rooms. The arrangement of dampers is shown in Figure 10.3. In some applications as in operating theatres, ventilation air can be 100%.

(iv) **Return air fan:** It takes the air from the rooms and distributes it through return air ducts back to the air conditioning unit or to the outdoors. In small systems with little or no return air ducts, the return air fan is not required because the supply fan can be used to draw in the return air.

(v) **Preheat coil:** The preheat coil may be located either in the outside air or the mixed airsteam. It is required in cold climates (below freezing) to increase the temperature of air so that the chilled water cooling coils do not freeze. It is optional in milder climates and when DX (dry expansion) cooling coils are used.

(vi) **Filters:** The filters are required to clean the air.

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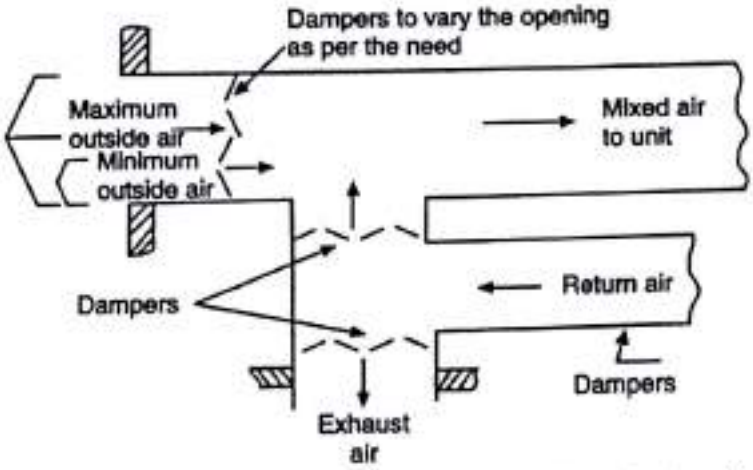

Figure 10.3 Dampers to vary the proportion of outside and return duct air.

Bypassing air around the cooling coil shown in Figure 10.4 provides another method of controlling humidity but does not give as good a humidity control in the space as with a reheat coil.

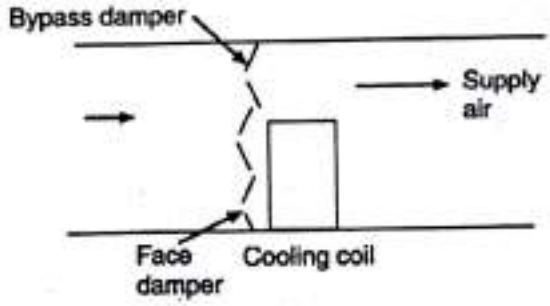

Figure 10.4 Arrangement of face and bypass dampers to provide reheat for humidity control.

A room thermostat will control the cooling coil capacity to maintain the desired room temperature. If control of room humidity is required, a room humidistat is used.

To achieve satisfactory temperature and humidity control in different zones, individual single zone units can be used for each zone. This may unacceptably increase costs and maintenance. However, there are a number of schemes that require only one air handling unit to serve a number of zones.

Four basic types of multiple-zone (all-air units and systems) systems are available:

- Reheat system
- Multizone system
- Dual duct system
- Variable air volume (VAV) system.

The reheat, multizone and dual duct systems are all constant air volume (CAV) type systems. That is, the air quantity delivered to the rooms does not vary. The temperature of this air supply is changed to maintain the appropriate room temperature. The variable air volume (VAV) varies the quantity of air delivered to the rooms.

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10.5 REHEAT SYSTEM

In this system the air conditioning system is the same as with a single-zone system (air filters, cooling/heating coils, and fans as in central air conditioning). In this reheat type, a separate single duct is laid from the AHU unit to each zone or room that is to be controlled separately as shown in Figure 10.5.

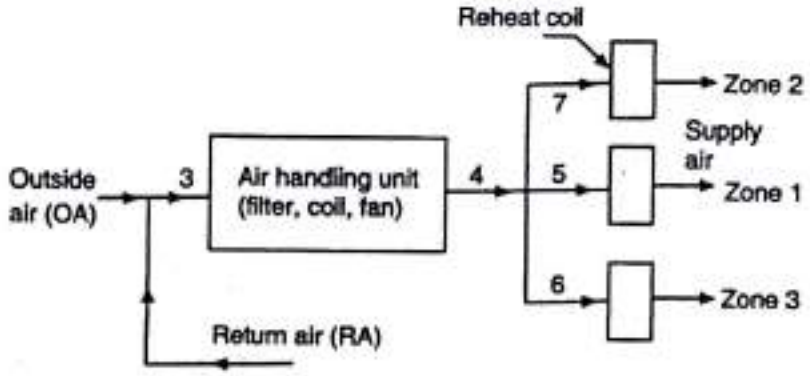

Figure 10.5 Reheat system with individual reheat coils.

A separate reheat coil is used for each zone so that one can achieve better control over both temperature and humidity. There is a wastage of energy as air is cooled in the AC system and then reheated as per the needs of each zone or room. A thermostat fitted in each room controls the temperature of the respective room or zone. The various processes of reheat system are shown on the psychrometric chart in Figure 10.6.

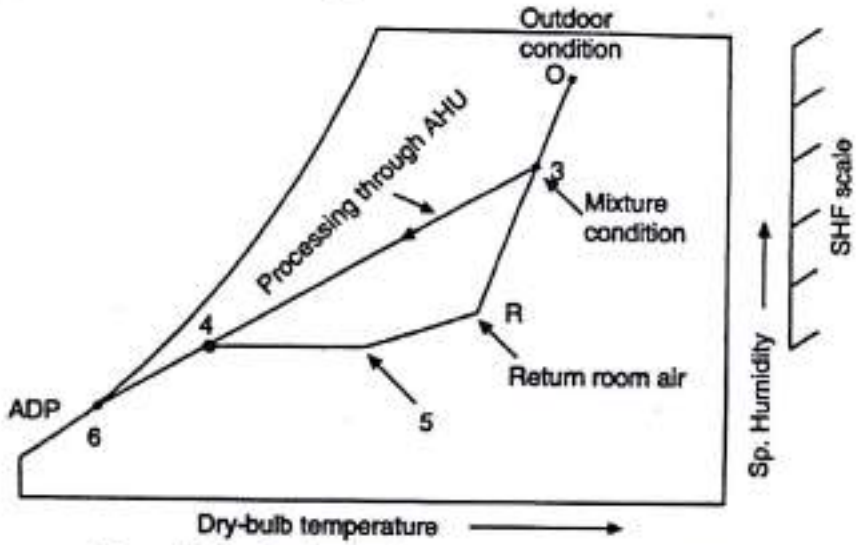

Figure 10.6 Psychrometric processes for reheat system.

10.6 MULTIZONE SYSTEM

The system uses an AHU that consists of a heating coil (hot deck) and a cooling coil (cold deck) but are placed parallely as shown in Figure 10.7. Zone dampers are installed in the unit across the hot and cold decks at the outlet of the unit. Separate ducts run for hot and cold air but are placed adjacent to each other. The hot and cold air is first mixed in a definite proportion to achieve the required temperature of an individual zone and then supplied. The duct arrangement for the multizone system is shown in Figure 10.8.

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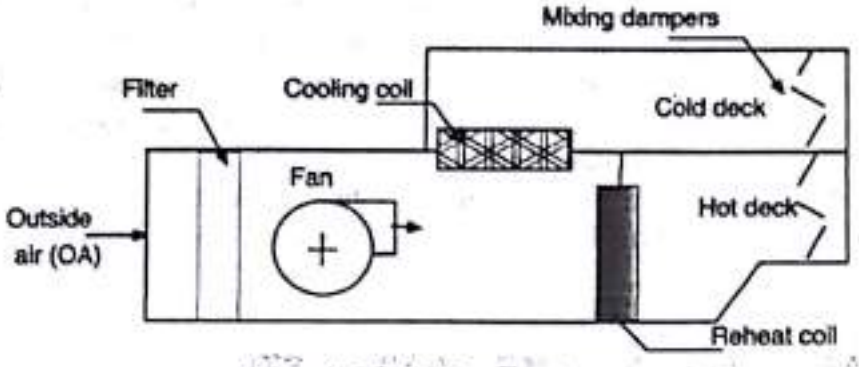

Figure 10.7 Multizone system.

The multizone system is suitable for one AHU with 12–14 zones. It is relatively inexpensive where only a few separate zones are desired and humidity conditions are not critical because one cannot control humidity accurately in this unit.

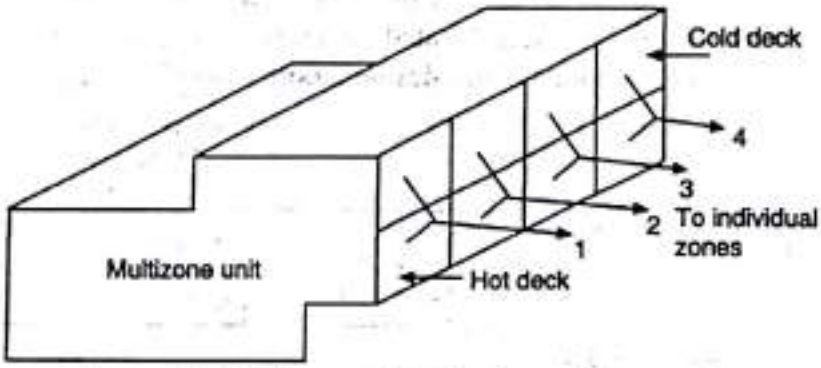

Figure 10.8 Duct arrangement for the multizone system.

10.7 DUAL DUCT SYSTEM

The system is as shown in Figure 10.9. It consists of a common filter, a common fan and individual heating and cooling coils placed in two parallel ducts. Mixing boxes are provided in the zone. The hot and cold air is mixed in the box in a definite proportion as per the needs of that zone. Placing a thermostat in each zone and sending the signal for the operation of dampers placed in the ducts leading to the box achieve this.

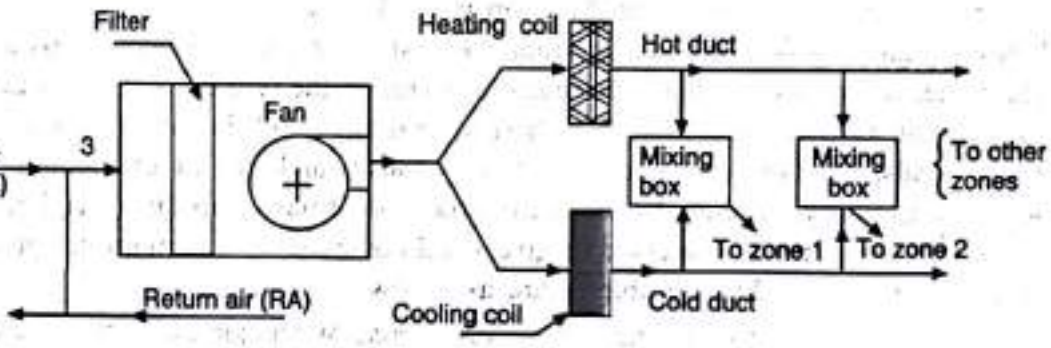

Figure 10.9 Dual duct system.

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The availability of cold and warm air at all times in any proportion gives the dual duct system a great flexibility in handling many zones with widely varying loads. Dual duct systems are usually designed as high velocity air systems in order to reduce duct sizes. Fan horsepower requirements are high because large volumes of air are moved at high pressure. So the cost of the dual duct system is usually quite high.

Both the dual duct and multizone systems are inherently energy wasteful, since during part load cooling for a zone, overcooled air is reheated by mixing warm air with it—a double waste of energy.

10.8 VARIABLE AIR VOLUME (VAV) SYSTEM

In this system the air quantity supplied to each zone or room is varied to maintain the appropriate room or zone temperature.

The basic VAV system arrangement is shown in Figure 10.10. A single main duct is run from the air handling unit. Branch ducts are run from this main duct through VAV boxes to each zone. The VAV box has an adjustable damper or valve so that the air quantity delivered to the space can be varied. Room thermostats located in each zone control the dampers in their respective zone VAV boxes to maintain the desired room set-point temperature.

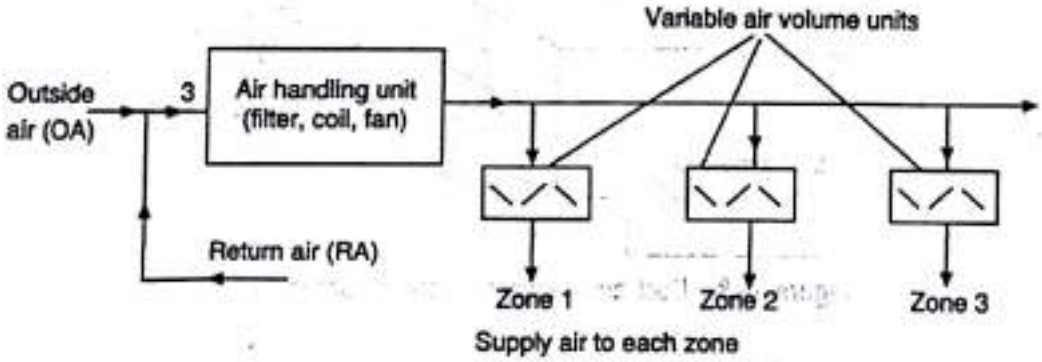

Figure 10.10 Variable air volume (VAV) system arrangement.

The psychrometric processes for a VAV system are shown in Figure 10.11 for summer cooling. The average room conditions are as indicated by point *R*. Zone Z1 is shown at part load when its sensible load has decreased, but its latent load has not. Its RSHF line is therefore steeper as shown. To maintain the design room DB temperature, the airflow rate to zone Z1 is throttled, and the room DB in zone Z1 is the same as *R*, as desired. Notice, however, that the humidity in zone Z1 is higher (point Z1) than desired.

There are many difficulties in operating the system at partial loads. At partial load of a particular zone, where cooling load continues to decrease, the reheat coil is activated. This type of VAV box can also be used to handle the problem with high latent heat loads.

At low loads, air flow rate decreases tremendously and the air circulation in the room becomes unsatisfactory, leading to uncomfortable conditions. This happens because the air supply diffusers are generally selected to give good coverage at maximum design air quantity.

The solutions to the above problems are as follows:

1. One solution is to use the reheat VAV box. When air quantity is reduced to the minimum for good air distribution, the reheat coil takes over.

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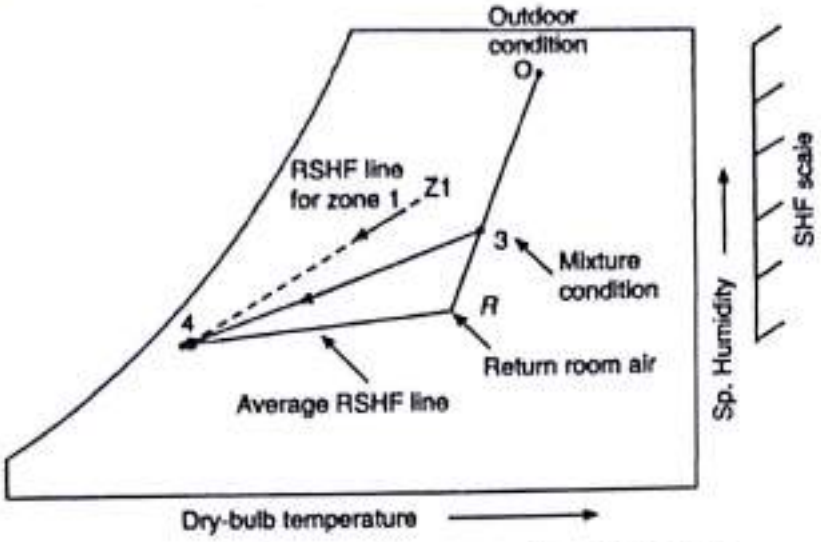

Figure 10.11 Psychrometric processes for reheat system.

2. Use variable diffusers. These diffusers have a variable sized opening. As the air flow rate decreases, the opening narrows thus resulting in better air distribution.
3. A further solution is to use fan-powered VAV boxes. This type of VAV box has a small fan. In addition to the supply air quantity, this fan draws in and recirculates some room air, thus maintaining a high total airflow rate through the diffuser.

In spite of these potential problems, VAV systems are still very popular. This is because of their significant energy saving feature when compared to other (constant air volume) multiple zone central systems. There is also another significant energy saving feature. Whenever there is part load, the air supply quantity is reduced and there is a saving of the fan power. Since a typical air conditioning system operates at part load of up to 95% of the time, this saving is considerable.

10.9 ALL-WATER SYSTEM

These are also known as *hydronic systems*. Hydronic systems distribute chilled or hot water from a central plant to each space or room. No air is distributed from the central plant. The system is schematically shown in Figure 10.12.

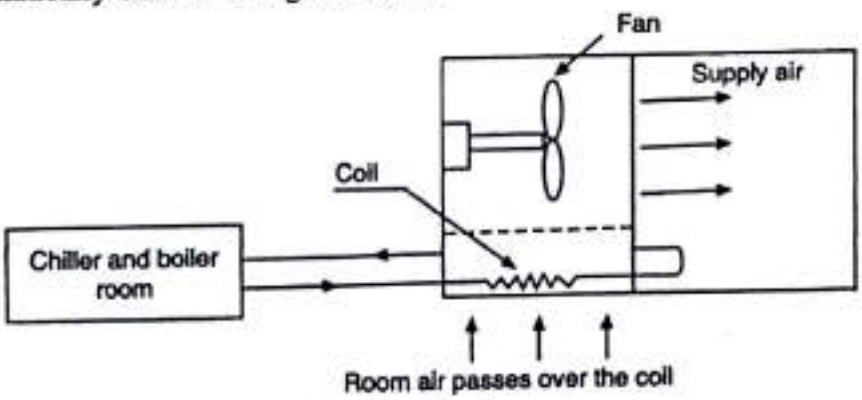

Figure 10.12 All-water system.

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It is difficult to provide fresh air intake for each fan coil unit making it a room air conditioner. The fresh air supply is by opening of door or periodical opening of windows by the occupants. In case this is not acceptable, fresh air intake can be provided to the fan coil unit at a higher cost.

It takes less space and is considerably less expensive because there is no ductwork in the central air handling unit. The specific heat of water is almost four times that of air and its density is 1000 times the density of air. Therefore, for the same amount of heat dissipation very less quantity of water needs to be circulated. It requires less coil surface area. It is useful when only limited space is available. For example, installation of an AC system in existing large buildings that were not originally designed to include AC.

The all-water system has certain drawbacks too. The multiplicity of fan coil units increases the maintenance work and cost. Control of ventilation air is not precise as there is no provision for a separate ventilation arrangement. Fresh air enters only through the door and window openings. The control of humidity is also limited.

10.10 AIR-WATER SYSTEMS

A combination of air-water systems distributes both chilled and/or hot water and conditioned air from a central system to the individual rooms.

One type of air-water system uses fan-coil units as the room terminal units. Chilled or hot water is distributed to them from the central plant. Ventilation air is distributed separately from an air-handling unit to each room.

Another type of air-water system using room terminal units is called *induction unit*. It receives chilled or hot water and ventilation air from the central plant (from a central air handling unit). The central air delivered to each unit is called *primary air*. As it flows through the unit at high velocity, it induces room air (secondary air) through the unit and across the water coil. Therefore, no fans or motors are required in this type of unit, reducing maintenance greatly. The induction unit air-water system is very popular in high-rise office buildings and similar applications. Its initial cost is relatively high.

10.11 UNITARY VS. CENTRAL SYSTEMS

As already stated earlier, the classification of air conditioning systems into unitary and central systems, is not according to how the system functions, but how the equipment is arranged.

In a unitary system, the refrigeration and air conditioning components are factory selected and assembled in a package. This includes refrigeration equipment, fan, coils, filters, dampers and controls.

A central system is one where all the components are separate. The engineer has to design and install the central plant and its suitable components are based on the air-conditioning load.

Unitary equipment is usually located in or close to the space to be conditioned whereas the central equipment is usually remote from the space, and each of the components may or may not be remote from each other, depending on the desirability.

Unitary systems are generally all-air systems limited largely to the more simple types such as single-zone units with or without reheat. This is because they are factory assembled on a volume basis.

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Central systems can be all-air, all-water or air-water systems and they are generally suitable for multizone units.

Unitary systems and equipment can be divided into the following three groups.

- Room units
- Unitary conditioners
- Rooftop units

These names are not standardized in the industry. For example, unitary conditioners are also called self-contained units or packaged units.

10.12 AIR CONDITIONING EQUIPMENT

A central air conditioning system has processing equipment such as air cleaner, cooling coil and heating coil, humidifiers and fans fitted in an AHU. All these items of equipment are required in different capacities to suit the requirement. Therefore, various such types of equipment are available for selection. These are described in the subsequent paragraphs.

10.13 COOLING COIL

Cooling coils may be either chilled water or evaporating refrigerant. The latter are called dry expansion (DX) coils.

Cooling coils are usually made of copper tubing with aluminium fins, but copper fins are sometimes used as shown in Figure 10.13.

Figure 10.13 Cooling coils.

The number of fins placed per centimetre is called *fin density*. The surface area provided by the fins and available for heat exchange is called *secondary area*. The inner surface area of the tubes through which water flows is called *primary area*. Fins are provided to make the heat exchangers compact. Normally the secondary area is 8 to 10 times the primary area depending upon the fin density. The coil may be constructed either with tubes in series or in parallel to reduce the water pressure drop.

When cooling coils have a number of rows, they are usually connected so that the flow of water and air are opposite to each other, called counterflow (Figure 10.14). In this way, the coldest water is cooling the coldest air, thus fewer rows may be needed to bring the air to a chosen temperature than if parallel flow were used, and therefore the chilled water temperature can be higher.

The water inlet connection should be made at the bottom of the coil and the outlet at the top, so that any entrapped air is carried through more easily. In addition, an air vent should be located at the outlet on top.

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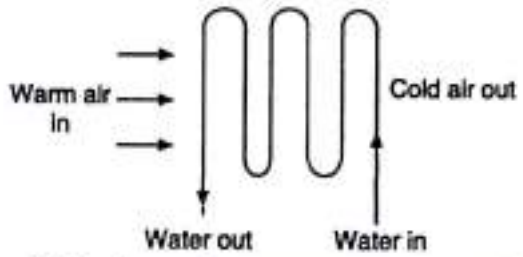

Figure 10.14 Counterflow arrangement of air and water.

10.13.1 Coil Selection

Coil selections are made from the manufacturer's tables or charts based on the required performance. The performance of a cooling coil depends on the following factors:

1. The amount of sensible and latent heat that must be transferred from the air.
2. DBT and WBT of air entering and leaving.
3. Coil construction—number and size of fins, size and spacing of tubing, number of rows.
4. Water (or refrigerant) velocity.
5. Air face velocity. The face velocity is the airflow rate in CFM divided by the projected (face) area of the coil.

Water velocities from 0.3 m/s to 2.5 m/s are used. High water velocity increases heat transfer but also results in high pressure drop and therefore requires a large pump and increased energy consumption. Velocities in the midrange of about 1–1.25 m/s are recommended.

High air velocities also result in better heat transfer and also more volume of air handled. However, if the coil is dehumidifying, the condensed water will be carried off the coil into the airstream above 125–150 cmm face velocity and eliminator baffles must be used to catch the water droplets.

The form in which manufacturers present their coil rating data varies greatly, one from another. Since using these ratings does not give much insight into how a coil performs, we will not present any rating data here.

The dehumidification effect of the cooling coil frequently results in water collecting on the coil. The water may then be carried as droplets into the moving airstream. To prevent this water from circulating into the air-conditioning ductwork, eliminators are provided downstream from the coil. This consists of vertical Z-shaped baffles that trap the droplets, which then fall into the condensate pan.

Access doors should be provided to permit maintenance. They should be located on both sides of the coils and filters. In a large-size equipment, lights should be provided inside each section.

10.14 HEATING COILS

The heating of air could be carried out in a number of ways. The heating device could be a steam coil, hot water coil, an electric heater, fuel gas furnace or a heat pump.

Steam coils

Steam generated in the boiler of a boiler room is carried through insulated pipe to the heating coils placed suitably in the conditioned space.

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The steam coils consist of copper tubes connected to a common header and mounted within a metal casing. Spiral or plate fins are mounted on the tubes at spacings of 1 to 6 fins/cm depending upon the manufacturer's specifications. The coil could be one row or two rows.

The coils are available up to steam pressure of 1.2 MN/m^2. This type of coil needs a boiler and other accessories and therefore it is used only where the extent of heating justifies the expenses.

Hot water coil

Water is heated in a separate tank or in a boiler and supplied to the heating coils. The hot water coils are like single tube steam coils available in two rows. The coils may have multicircuits to reduce pressure drop and turbulators to ensure turbulent flow. Hot water coils are available up to 120°C water temperature.

Electric heaters

In open-type heaters, the heating element is exposed directly to air. These elements operate at lower temperature and their life is more. The response is also quicker.

In finned tube type, the heating element is placed in a finned tube surrounded by refractory material. It permits use of fins and cannot be damaged even by large impurities in air.

Electrical heaters have low initial cost, low installation cost, simplicity of operation and control, fast response and clean environment. Since the cost of electrical energy is very high, the use of electric heaters is limited to small-capacity units.

Duct furnaces

Gas-fired furnaces are used in air ducts. It has a burner section, a heat exchanger, a plenum and controls. This is a direct heat producing device and hence more efficient than an electric heater. Its simplicity, easy control and clean installation makes it suitable for small applications like domestic heating. Heating coils, in general, have higher face velocities than those in the cooling coil.

10.15 AIR CLEANING DEVICES (FILTERS)

Air conditioning systems that circulate air, also generally have devices that remove dust or dirt particles which result largely from industrial pollution. Occasionally, gases that have objectionable odour are also removed from the air.

Air conditioning systems should have proper filters to clean the air for the following reasons:

- Dust particles can cause serious respiratory ailments (emphysema and asthma). So, to protect human health and for comfort, dust particles have to be removed.
- To maintain cleanliness of room surfaces and furnishings.
- To protect equipment and machinery, the working of which are affected by air pollutants like dust or dirt. Some equipment will not operate properly or will wear out faster without adequate supply of clean air.
- To protect the air-conditioning machinery, for example, lint collecting on coils will increase the coil resistance to heat transfer.

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10.15.1 Types of Filters/Cleaners

Air cleaners can be classified in a number of ways such as (1) types of media, (2) permanent or disposable, (3) stationary or removable, and (4) electronic air cleaners.

10.15.2 Types of Media

Viscous impingement filters

The dust particles in the air stream strike the filter media and are therefore stopped. The viscous impingement air filter has a media of coarse fibres (glass fibres and metal screens) that are coated with a viscous adhesive. Air velocities range from 90–180 mpm. This type of filter will remove larger dust particles satisfactorily but not the smaller particles. It is low in cost. Refer to Figure 10.15(a).

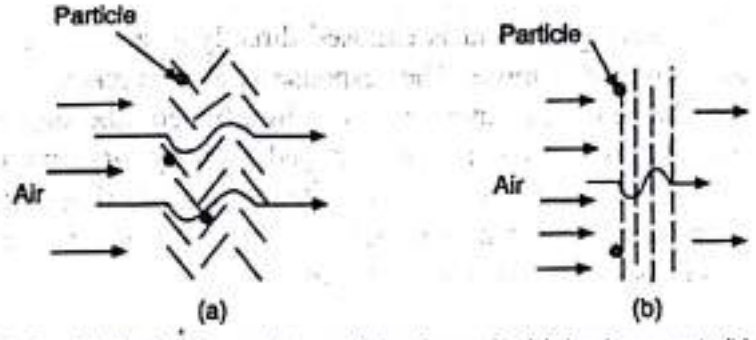

Figure 10.15 Methods of removing particles from air: (a) Impingement; (b) straining.

In straining type air filters [Figure 10.15(b)], the dust particles are larger than the space between adjacent fibres and therefore are trapped there.

Dry-type filters

These use uncoated fibre mats (glass fibres and paper) as shown in Figure 10.16. The media can be constructed of either loosely packed coarse fibres or densely packed fine fibres. By varying density, dry-type air filters are manufactured that have good efficiency only on larger particles, as with the viscous impingement type, or are also available with medium or high efficiency for removing very small particles.

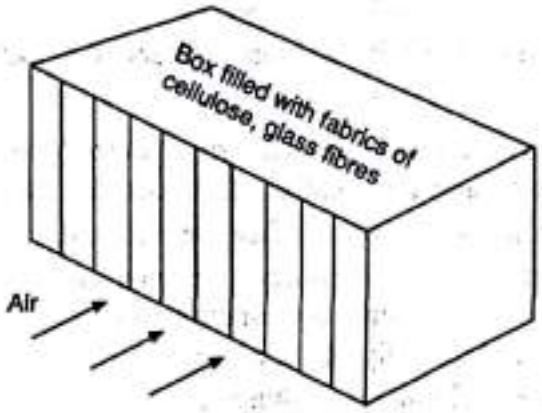

Figure 10.16 Dry filter.

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HEPA (High Efficiency Particulate Air) filter

It is a very high efficiency dry-type filter for removing extremely small particles. For example, it is the only type of filter that will effectively remove viruses as small as 0.05 micron. Air face velocities through HEPA filters are very low (about 15 mpm).

The media in air filters can be arranged in the form of random fibre mats, screens or corrugated sinuous strips.

Permanent and disposable air filters

One can develop disposable type air filters so that they can be discarded. One can have permanent type too, which when saturated with dust, can be cleaned and reused. Permanent types have metal media that will withstand repeated washings, but cost more than disposable types.

Stationary and renewable air filters

Stationary air filters are manufactured in rectangular panels. The panels can be removed and either replaced or cleaned when dirty. Renewable type air filters consist of a roll mounted on a spool that moves across the airstream as shown in Figure 10.17. The fabric is wound on a take-up spool, driven by a motor. The clean fabric is continuously brought in front of the air stream by rotating the shaft. Renewable air filters are considerably more expensive than the stationary types, but maintenance costs are greatly decreased. Either fibrous materials or metal screens are used as media.

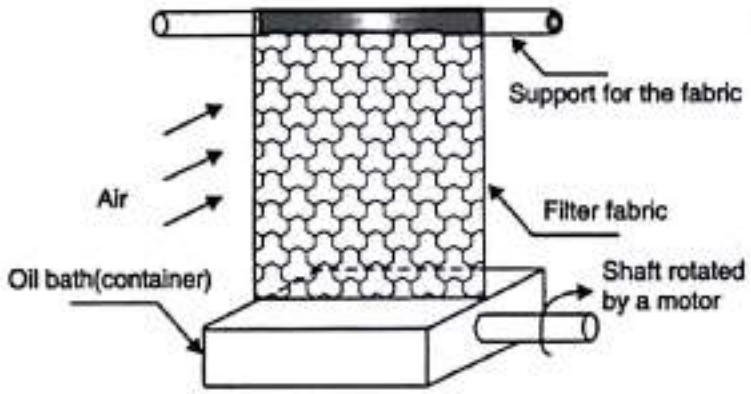

Figure 10.17 Stationary or renewable or viscous impingement filter.

10.15.3 Electronic Air Cleaners

Dust particles are given a high voltage charge by an electric grid (bank of charging plates) shown in Figure 10.18. A series of parallel plates are given the opposite electric charge. As the dust laden airstream passes between the plates, the particles are attracted to the plates. The plates may be coated with a viscous material to hold the dust. After an interval of time, the air cleaner must be removed from the service to clean the plates and remove the dirt. Electronic air cleaners are expensive, but are very efficient for removing both large and very small particles.

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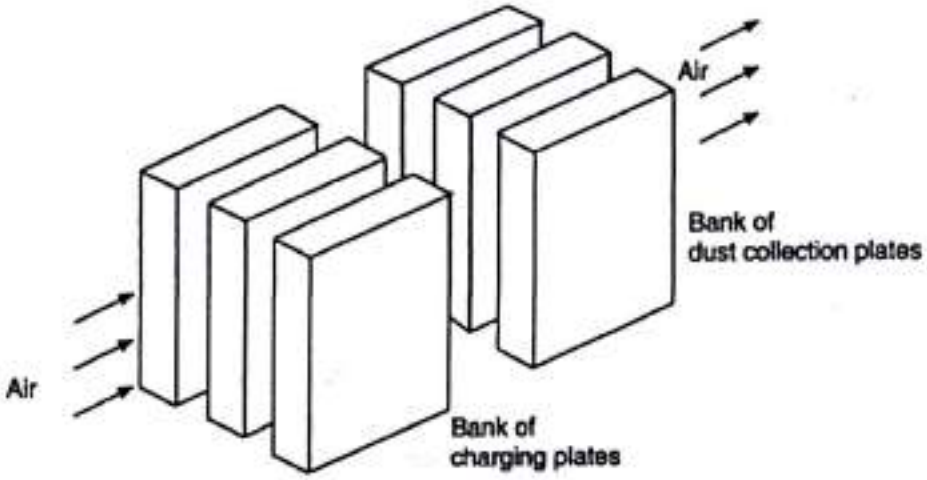

Figure 10.18 Electrostatic filters.

10.15.4 Choice of Filter

Choice of a filter depends on two factors—(1) contaminants in air and (2) performance of the filter.

Contaminants in the air

The dust particles are either solids like dust, unburnt carbon, pollens, spores, bacteria and virus or aerosols like smoke, fumes and mist. Their size, shape and concentration affect the choice of filter. The sizes of some of the contaminants are given in Table 10.1.

Table 10.1 Size of various contaminants

Tobacco smoke	0.2 micron	Bacteria	2 to 10 micron
Dust	1 to 100 micron	Pollen	10 to 150 micron
Mist	50 to 150 micron	Virus	0.05 to 2 micron

The concentration of dust particles varies from place to place. In rural areas, air may just contain mud particles while in metro cities it may contain smoke particles emitted from the vehicles.

Performance criteria

The performance of a filter is rated as per the air flow resistance and dust capacity efficiency. The filter resistance varies directly depending on the flow of air and dust content in the airflow.

The capacity of a filter is a measure of the life of the filter. The life of a filter is the period between two cleanings for cleanable filters and between replacements for disposable filters.

The efficiency of a filter is the ratio of the mass of impurities removed to the mass of total impurities present. But for fine filters, efficiency is measured by the dust spot method. The dust spot method measures the extent of blockage of light by the dust spot on the filter.

10.16 HUMIDIFIERS

These are the devices used to add moisture to the air so that the humidity increases. This becomes necessary in comfort air conditioning when the RH of atmospheric air is below 30%. The various methods of humidification are discussed in the following text.

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Pan humidifier

It is the simplest humidifier. Warm air is passed over the surface of water in the pan due to which evaporation of water takes place. The vapour mixes with the flowing air. The rate of evaporation can be increased by a small heating element in the pan. This is suitable only for a small-size application due to limited capacity of humidification available.

Wetted pack humidifier

Air is passed across the water absorbent pad which is regularly wetted in a pan. The water evaporates and the air carries away the water vapour. This is also suitable for small requirements only.

Air washer or spray chamber

An air washer consists of the water sprays, arranged as shown in Figure 10.19. The baffles (not shown in the figure) are placed at the entry of air which assure uniform airflow. As air passes, the water droplets are evaporated and mixed with the air. Water falling down is collected in a pan and recirculated. An eliminator at the outlet prevents carryover of water.

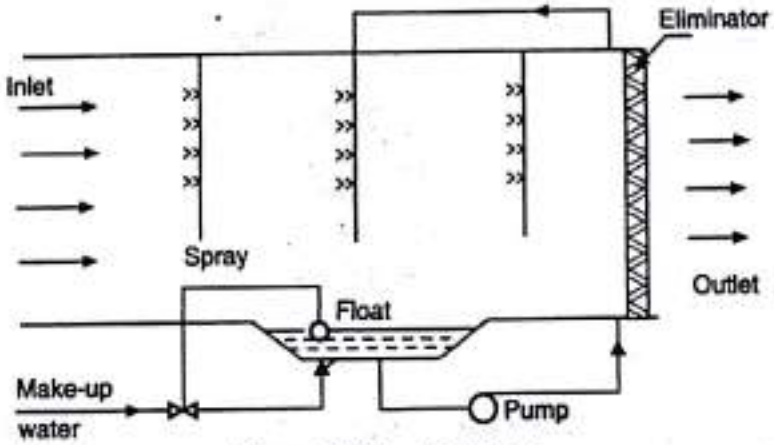

Figure 10.19 Air washer.

The air velocities through washer are 100 to 200 m/min to ensure proper functioning of the eliminator. Selection is done usually at permissible maximum velocities. A square cross-section reduces the cost of washer. Spray pressures are usually 1.5 to 2 bar. Spray density is generally 1.8 L/m^2 to 2.5 L/m^2 of face area.

Steam type humidifier

Many sections of the textile industry require RH higher than 80%. In such cases steam is introduced directly into the air for quick humidification.

10.17 FAN

Fans are necessary to distribute air through ducts to spaces that are to be conditioned. In this section, we will study the types of fans, their performance and selection, etc.

10.17.1 Types of Fans

Fans are mainly classified into two groups, namely centrifugal and axial fans.

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Centrifugal fans

In a centrifugal fan, air is pulled along the fan shaft and then blown radially outwards from the shaft. The air is usually collected by a scroll casing and concentrated in one direction. Centrifugal fans may be subclassified into forward curved, radial, backward curved and backward inclined types. These differ in the shape of their impeller blades as shown in Figure 10.20. In addition, backward curved blades with a double-thickness blade are called *airfoil blades*.

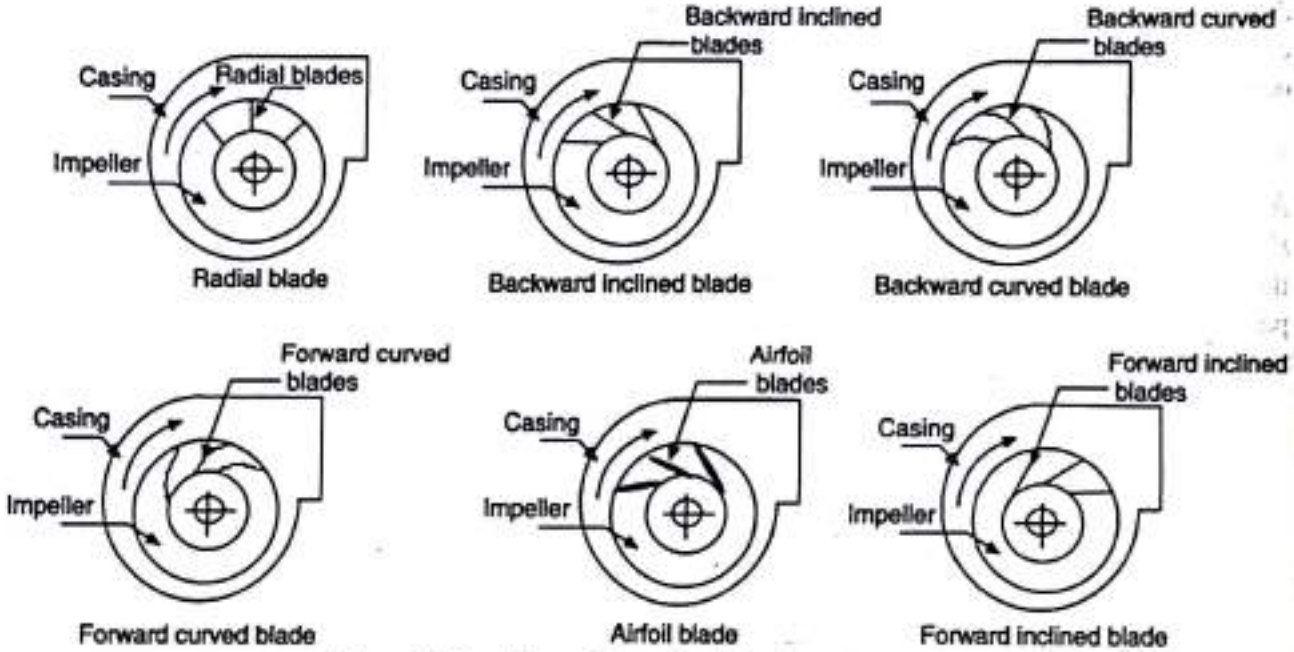

Figure 10.20 Types of centrifugal fan impeller blades.

Axial fans

In an axial flow fan, air is pulled along the fan shaft and then blown along the axis of the shaft. Propeller, tubeaxial, and vaneaxial types (Figure 10.21) are available in axial fans. The propeller fan consists of a propeller-type wheel mounted on a ring or plate. The tubeaxial fan has a vaned wheel mounted in a cylinder. The vaneaxial fan is similar to the tubeaxial types, except that it also has guide vanes behind the fan blades which improve the direction of air flow through the fan.

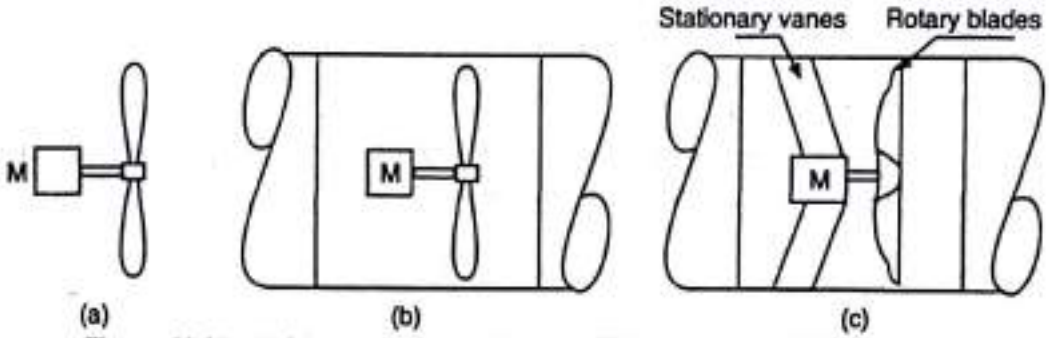

Figure 10.21 Axial type fans: (a) Propeller, (b) Tubeaxial, and (c) Vaneaxial.

10.17.2 Performance Characteristics of Fans

In general, any fluid flow through the conduit is opposed by the friction between the fluid and the conduit surface. There is a resistance to the flow of air through ducts. Therefore, a fan is

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required for two purposes: (1) To overcome this resistance to flow due to friction; (2) To supply the air to the conditioned space. Therefore, energy in the form of pressure must be supplied to the air. This is accomplished by the rotating fan impeller, which exerts a force on the air, resulting in both flow of the air and an increase in its pressure.

Knowledge of the fan performance is useful for correct fan selection and proper operating and troubleshooting procedures.

The volume flow rate of air (cmm) delivered and the pressure (H_t = total pressure, in mm of water gauge), created by the fan are called *performance characteristics*. Other performance characteristics of importance are efficiency (η) and brake power (BP). ME = mechanical efficiency = air power output/BP input. Fan performance is best understood when presented in the form of curves.

Forward curved blades centrifugal fan

Figure 10.22 shows the typical performance curves of a forward curved blades centrifugal fan.

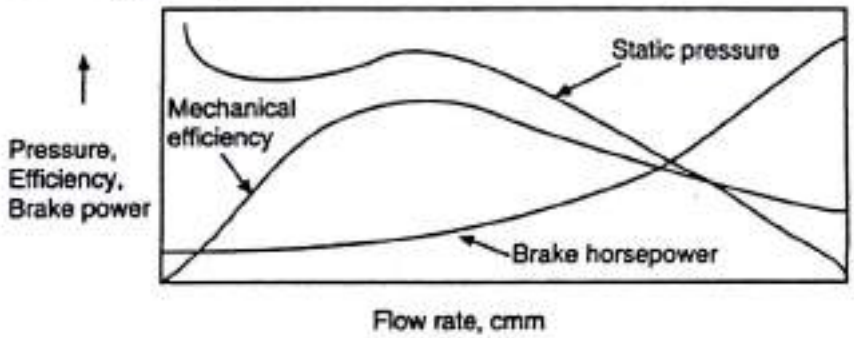

Figure 10.22 Performance curves of a forward curved blades centrifugal fan.

Some important features of this fan are follows:

1. As the flow (cubic metre per min, cmm) increases, the static pressure developed by the fan first decreases, then increases and reaches a maximum value, and then continuously decreases. The pressure developed has a slight peak in the middle range of flow, then the pressure drops off as the flow increases.
2. The BP required increases sharply with the flow.
3. Efficiency is highest in the middle ranges of flow.

Backward curved blades centrifugal fan

The performance curves of a backward curved blades centrifugal fan are shown in Figure 10.23.

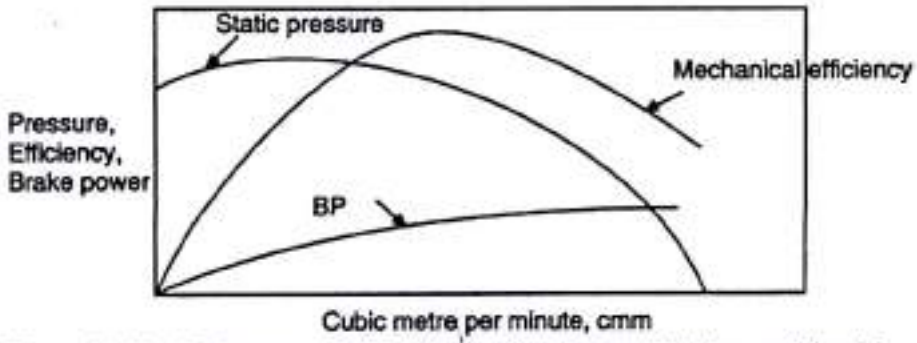

Figure 10.23 Performance curves of a backward curved blades centrifugal fan.

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Some important features of this fan are follows:

1. The pressure developed has a slight peak in the middle range of flow, then the pressure drops off as the flow increases.
2. The BP increases only gradually, reaches the maximum, and then falls off.
3. Efficiency is highest in the middle ranges of flow.
4. A higher maximum efficiency can often be achieved with a backward curved blades type of fan.

10.17.3 Fan Selection

The choice of fan for a given application depends upon the pressure developed, volume flow rate and on the performance characteristics.

Centrifugal fans are very common in ducted air conditioning systems. Forward curved blades centrifugal fans are usually lower in performance. The rising BP characteristic curve could result in overloading the motor if operated at a condition beyond the selected cmm. The operating cost will often be higher due to lower efficiency.

Backward (curved or inclined) blades centrifugal fans are generally more expensive than forward curved types. These consume comparatively less power than the forward type. There is a possibility of overloading the motor if the fan is delivering more air than it was designed for. Airfoil bladed fans have the highest efficiency of any type.

Propeller fans cannot create a high pressure and are thus used where there is no ductwork. These fans are often used in packaged air conditioning units because of low cost.

Tubeaxial fans are not suitable in ducted air conditioning systems. Vaneaxial fans are suitable for ducted air-conditioning systems. They usually produce a higher noise level than centrifugal fans and therefore may require greater sound reduction treatment. Their compact physical construction is useful when the space is limited.

10.17.4 Fan Ratings

Once the proper type of fan is selected for an application, the next step is to determine the proper size to be used. Usually the fan manufacturers provide the performance characteristics of a fan for the variation in the air flow (cmm). The specifications of a fan include cmm, speed (rpm), BP, wheel diameter, etc. Based on the performance curves of a fan supplied by the manufacturers, one can get the power consumption, pressure developed and cmm of the fan.

Performance curves help the engineer to visualize changes in static pressure, BP and efficiency easily. Note that each fan curve represents the performance at a specific fan speed and air density. Fans are usually rated with air at standard conditions: a density of 1.02 kg/m^3 at 21°C. Performance curves at different air conditions may be available from the manufacturer, but if not, they may be predicted from the fan laws.

Before selecting a fan, first the duct system static pressure resistance (duct H_s) is calculated. Manufacturer's data is then used to select a fan that will produce the required cmm against the system static pressure resistance. In effect, the fan must develop a static pressure (fan H_s) and cmm equal to the system requirements. The fan may also be selected on the basis of total pressure rather than the static pressure. Either basis is satisfactory for low-velocity systems. For high-velocity systems, it is sometimes more accurate to use total pressure.

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10.17.5 System Characteristics

It is quite essential to understand the system character, i.e. cmm versus pressure loss (H_f). The pressure loss due to frictional resistance in a given duct system varies as the cmm changes as follows:

$$H_{f2} = H_{f1}\left(\frac{\text{ccm}_2}{\text{ccm}_1}\right)^2 \tag{10.1}$$

Equation (10.1) can be used to find the changed pressure loss in a duct system for a changed cmm flow, if the pressure loss is known at some other flow rate.

By plotting a few of such H_f versus cmm points, a system characteristic curve can be determined. Note that the pressure loss rises sharply with cmm for any duct system, as shown in Figure 10.24.

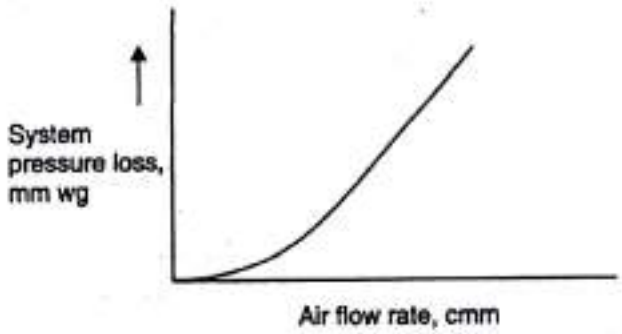

Figure 10.24 System characteristic curve.

10.17.6 Fan–system Interaction

Here we plot both the fan and system characteristic pressure versus the air flow, as shown in Figure 10.25. Then find the condition of operation of the fan and system—the intersection of the two curves gives this condition. The fan has its own performance curve and thus has its own characteristic curve.

Examining the fan and system curves is not only useful for selecting the operating condition, but aids in analysing changed conditions and in finding causes of operating difficulties. A

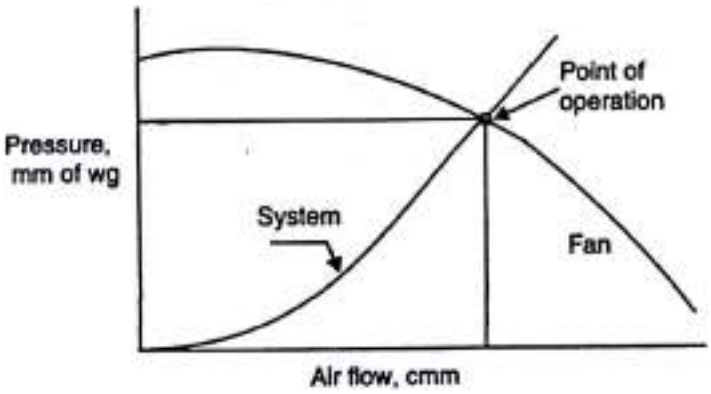

Figure 10.25 Fan and system curves plotted together.

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common occurrence in air conditioning systems is that the actual system resistance for a design CFM is different from that calculated by the designer. Some reasons due to which this may happen are as follows.

1. An error in calculating pressure loss.
2. The designer adds extra resistance as a safety measure to act as a "safety factor".
3. The contractor installs the ductwork in a manner different from that planned.
4. Filters may have a greater than expected resistance due to excess dirt.
5. An occupant may readjust damper positions.

The result of this type of condition is that the duct system has a different characteristic than was planned. An examination of the fan and system curves will aid in analysing these situations.

10.17.7 Selection of Optimum Fan Conditions

Fans of different sizes or different speeds would satisfy the pressure and cmm requirements; therefore the next step in selecting fans is to decide the criteria that should be used for selecting the "best" choice.

Some of these factors will now be examined.

- Fans should be chosen for close to maximum efficiency considering the pressure cmm curve.
- Fans should not be selected to the left of the peak pressure on the fan curve. At these conditions, the system operation may be unstable, have pressure fluctuations and the system may generate excess noise.
- When using forward curved blades centrifugal fans, it should be ensured that these do not operate at significantly greater than the design cmm. If so, the motor horsepower required will increase and a larger motor may be necessary.
- Fans may have pressure curves of varying steepness. If it is expected that there would be considerable changes in system resistance, but constant cmm is required, a fan with a steep curve is desirable.

10.17.8 Fan Laws

There are a number of relationships among fan performance characteristics for a given fan operating at changed conditions, or for different size fans of similar construction; these are called *fan laws*. These relationships are useful for predicting performance if conditions are changed. We will present some of these relationships and their possible uses:

$$\text{cmm}_2 = \text{cmm}_1 \times \frac{N_2}{N_1} \tag{10.2}$$

$$H_{t2} = H_{t3}\left(\frac{N_2}{N_1}\right)^2 \text{ and } H_{s2} = H_{s1}\left(\frac{N_2}{N_1}\right)^2 \tag{10.3}$$

$$\text{BP}_2 = \text{BP}_1\left(\frac{N_2}{N_1}\right)^3 \tag{10.4}$$

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$$H_{t2} = H_{t1} \times \frac{d_1}{d_2} \text{ and } H_{s2} = H_{s1} \times \frac{d_1}{d_2} \quad (10.5)$$

where

cmm = volume flow rate, m^3/min
H_s = static pressure, mm of water gauge (w.g.)
H_v = velocity pressure, mm of water gauge (mm of w.g.)
H_t = total pressure (mm of w.g.)
BP = brake power input
N = speed, revolutions per min (rpm)
d = air density (lb/ft^3)
ME = mechanical efficiency = air power output/BP input

EXERCISES

1. What are the factors that one has to keep in mind while selecting an air conditioning system?
2. Give the classification of air conditioning systems.
3. Explain what a packaged air conditioning system is.
4. Is a window air conditioner really an air conditioner or an air cooler? Explain.
5. Write a short note on split air conditioner.
6. Discuss the limitations of the window air conditioner.
7. Discuss the position of package air conditioner in the range of unitary and central systems.
8. Describe the operation of an AHU during different seasons of the year.
9. Describe a central air conditioning system.
10. Describe an all-air system.
11. Describe an all-water system and its application.
12. Explain the types of fans with neat sketches. Draw the performance curves of forward and backward curved blades centrifugal fans. What is system resistance? How does it help in fan selection? Explain fan ratings.
13. Compare the viscous impingement filter with the dry filter.
14. What are the advantages of chilled water coil?
15. Briefly describe the different types of heating and cooling devices.
16. Write a short note on the air-washer type humidifier.
17. Compare the performance of axial and centrifugal fans.
18. Compare the forward and backward curved blades centrifugal fans.
19. What are fan laws and explain their significance?
20. How are the air distribution system and fan balanced?
21. What is the effect of dirty filters on system balance?

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