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THE PSYCHROMETRIC CHART AND ITS USE

Psychrometry is an impressive word which is defined as the measurement of the moisture content of air. In broader terms it is the science and practices associated with atmospheric air mixtures, their control, and the effect on materials and human comfort. This can be accomplished through use of psychrometric tables or a psychrometric chart. The tables are somewhat more accurate, but the chart is accurate enough for all practical purposes and is much easier to use.

Before we start to explain the psychrometric chart, let us review a few of the principles on which it is based.

First, atmospheric air is a mixture consisting of dry air and water vapor in varying relative amounts. It is best to think of dry air and water vapor separately, for they behave independently of one another. In this chapter, when dry air is referred to, it will mean the air part; when water vapor or moisture is referred to, it will mean the water part; and when referring to just air, it will mean the mixture of dry air and water vapor.

Dry air is itself a mixture of several gases, mainly nitrogen and oxygen, but these do not change in their proportions, and they behave so much the same, that we can consider dry air as if it were a single gas.

We know dry air only as a gas. It can be liquefied, but at very low temperatures, about 300°F below zero. In our atmosphere it is superheated three or four hundred degrees, or far above its condensing or boiling temperature.

Thus, when we warm or cool dry air, we add or remove sensible heat only, and are not concerned with latent heat. Dry air obeys the gas laws of Charles, and Boyle, which state that if a gas, such as air, is heated, it expands and its density or weight per cubic foot becomes less, provided the pressure remains constant. Conversely, if we cool dry air, it becomes heavier per cubic foot, again provided that its pressure remains constant. Moreover, the temperatures, densities, volumes, and pressures all vary proportionately.

Therefore, problems connected with cooling or heating dry air are rather simple, for when we cool it we remove its sensible heat of about 1/4 of a Btu per pound of dry air per degree, and when we heat dry air, we add that same amount of sensible heat.

Picture a room at normal temperatures. It is filled with a mixture of dry air and water vapor. Each fill the room, so their volumes are equal. However, their densities are far different because the water vapor is a very light gas. At ordinary room temperatures, it constitutes only about 1/2 to 1–1/2 per cent of the total weight of the mixture which we call air.

Air, in addition to having weight, also exerts a pressure, called barometric pressure, which is usually measured in inches of mercury above a perfect vacuum. Standard barometric pressure at sea level is 29.921 inches of mercury, which is equivalent to 14.696 pounds per square inch absolute.

The pressure exerted by the air, or barometric pressure, is caused by both the dry air and the water vapor. Most of the total pressure is from the air, but some is from the water vapor. As is true of the weight, or density, the pressure from the water vapor is usually only about 1/2 to 1–1/2 per cent of the total pressure.

It is essential to always keep in mind that air consists of two separate gases, dry air and water vapor, which act independently, each according to its individual properties, just as if the other were not there. The total density of air is the sum of the densities of the dry air and water vapor; and the total pressure is the sum of the partial pressure of the dry air and the partial pressure of the water vapor.

Although the water vapor is a very small part of air, as far as density and pressure are concerned, it is the part which complicates the psychrometric processes and calculations. For example, the amount of water vapor in the air varies widely from time to time according to temperature, and the proportion of water vapor in the air must be regulated in order to provide comfortable conditions.

It would not be necessary to regulate the amount of water vapor in the air if it remained as a superheated gas, and providing comfortable conditions would not be difficult, for we would only have to control the dry bulb temperature of the air. It's not that simple, however, because at the temperatures at which we normally work, the water vapor sometimes condenses into water. When this occurs, a large amount of latent heat must be removed, and as you know, the latent heat per pound of water is extremely large compared to the sensible heat. Consequently, when water vapor is condensed out of the air, a large amount of cooling capacity is required.

In cooling air, such as in summer air conditioning, we are chiefly concerned with:

1. Cooling the dry air, which is comparatively simple and involves sensible cooling only. (The water vapor is such a small part of the air that we usually ignore its sensible heat.)

2. Controlling the amount of water vapor in the air. This involves condensing out some of the water vapor, thus bringing its large latent heat into the problem.

3. Circulating and distributing the air, which involves changes in density and volume.

4. Mixing together air from two different sources, at different temperatures, and different percents of water vapor.

These four processes, along with others, involved in the controlling of atmospheric conditions for human comfort, are commonly referred to as psychrometric processes. In order to arrive at the proper amounts of humidification or dehumidification and heating or cooling necessary to effect these changes, we make use of the psychrometric tables or charts.

THE USE OF THE PSYCHROMETRIC TABLE

The psychrometric properties in Table 54T03 have been compiled through countless laboratory experiments and mathematical calculations and are the basis for what we know as the psychrometric chart.

	DRY AIR		MIXTURES OF DRY AIR AND SATURATED WATER VAPOR						
	Specific Volume	Density	Specific Volume	Density	Moisture Content (per lb. of dry air)		Heat Content (Enthalpy) Btu/lb of air and moisture		
°F	Cu.ft/lb	lbs/cu.ft.	Cu.ft/lb	lbs/cu.ft	Pounds	Grains	Sensible	Latent	Total
0	11.58	.0864	11.59	.0863	.000781	5.47	0.0	.85	.85
1	11.60	.0862	11.63	.0860	.000825	5.77	.24	.90	1.14
2	11.63	.0860	11.65	.0858	.000869	6.08	.48	.95	1.43
3	11.65	.0858	11.67	.0857	.000916	6.41	.72	1.00	1.72
4	11.68	.0856	11.70	.0855	.000963	6.74	.96	1.05	2.01
5	11.70	.0854	11.72	.0853	.001015	7.10	1.21	1.11	2.31
6	11.73	.0852	11.75	.0851	.001067	7.47	1.45	1.16	2.61
7	11.75	.0851	11.77	.0849	.001125	7.87	1.69	1.22	2.91
8	11.78	.0849	11.80	.0847	.001183	8.28	1.93	1.29	3.21
9	11.80	.0847	11.83	.0845	.001246	8.72	2.17	1.35	3.52
10	11.83	.0845	11.86	.0843	.001309	9.16	2.41	1.42	3.83
11	11.85	.0843	11.88	.0841	.001378	9.64	2.65	1.51	4.16
12	11.88	.0841	11.91	.0840	.001447	10.13	2.89	1.57	4.46
13	11.91	.0839	11.94	.0838	.001523	10.66	3.13	1.65	4.78
14	11.94	.0837	11.97	.0836	.001599	11.19	3.38	1.73	5.11
15	11.96	.0836	12.00	.0834	.001681	11.77	3.62	1.82	5.44
16	11.99	.0834	12.02	.0832	.001764	12.35	3.86	1.91	5.77
17	12.02	.0832	12.05	.0830	.001855	12.98	4.10	2.01	6.11
18	12.04	.0830	12.06	.0828	.001946	13.62	4.34	2.10	6.44
19	12.06	.0829	12.10	.0826	.002045	14.31	4.58	2.21	6.79
20	12.09	.0827	12.13	.0824	.002144	15.0	4.82	2.31	7.14
21	12.11	.0825	12.16	.0822	.002252	15.8	5.06	2.43	7.49
22	12.14	.0823	12.19	.0820	.002360	16.5	5.31	2.55	7.85
23	12.16	.0822	12.22	.0819	.002478	17.4	5.55	2.67	8.22
24	12.19	.0820	12.24	.0817	.002596	18.2	5.79	2.80	8.58
25	12.21	.0819	12.27	.0815	.002725	19.1	6.03	2.93	8.96
26	12.24	.0817	12.30	.0813	.002854	20.0	6.27	3.07	9.34
27	12.26	.0815	12.33	.0812	.002994	20.9	6.51	3.22	9.73
28	12.29	.0813	12.35	.0810	.003134	21.9	6.75	3.37	10.12
29	12.31	.0812	12.39	.0808	.003289	23.0	6.99	3.54	10.53
30	12.34	.0810	12.41	.0806	.003444	24.1	7.23	3.70	10.93
31	12.37	.0809	12.44	.0804	.003613	25.3	7.48	3.88	11.36
32	12.39	.0807	12.47	.0802	.003782	26.5	7.72	4.06	11.78
33	12.41	.0806	12.49	.0801	.003938	27.6	7.96	4.22	12.18
34	12.44	.0804	12.52	.0799	.004100	28.7	8.20	4.40	12.60
35	12.47	.0802	12.55	.0797	.004268	29.9	8.44	4.57	13.02
36	12.49	.0800	12.58	.0795	.004442	31.1	8.68	4.76	13.44
37	12.52	.0799	12.61	.0793	.004622	32.4	8.93	4.95	13.87
38	12.54	.0797	12.64	.0791	.004809	33.7	9.17	5.14	14.31
39	12.57	.0795	12.67	.0789	.005002	35.0	9.41	5.35	14.76
40	12.59	.0794	12.70	.0787	.005202	36.4	9.65	5.56	15.21

TABLE 54T03 Thermodynamic Properties of Air based on one pound of air at a total pressure of 29.921 in.hg. (Atmospheric pressure)

	DRY AIR		MIXTURES OF DRY AIR AND SATURATED WATER VAPOR						
	Specific Volume	Density	Specific Volume	Density	Moisture Content (per lb. of dry air)		Heat Content (Enthalpy) Btu/lb of air and moisture		
°F	Cu.ft/lb	lbs/cu.ft.	Cu.ft/lb	lbs/cu.ft	Pounds	Grains	Sensible	Latent	Total
41	12.62	.0792	12.73	.0785	.005410	37.9	9.89	5.78	15.67
42	12.64	.0791	12.76	.0784	.005625	39.4	10.14	6.01	16.14
43	12.67	.0789	12.79	.0782	.005848	40.9	10.38	6.24	16.62
44	12.69	.0788	12.82	.0780	.006078	42.5	10.62	6.48	17.10
45	12.72	.0786	12.85	.0778	.006322	44.2	10.86	6.73	17.59
46	12.74	.0785	12.88	.0776	.006564	45.9	11.10	6.99	18.09
47	12.77	.0783	12.91	.0775	.006823	47.7	11.34	7.26	18.60
48	12.79	.0782	12.94	.0773	.007086	49.6	11.58	7.54	19.12
49	12.82	.0780	12.97	.0771	.007364	51.5	11.83	7.83	19.65
50	12.84	.0779	13.00	.0769	.007643	53.5	12.07	8.12	20.19
51	12.87	.0777	13.03	.0767	.007932	55.5	12.31	8.43	20.74
52	12.89	.0776	13.07	.0765	.008234	57.6	12.55	8.75	21.30
53	12.92	.0774	13.10	.0763	.008556	59.8	12.79	9.08	21.87
54	12.95	.0772	13.13	.0762	.008872	62.1	13.03	9.41	22.45
55	12.97	.0771	13.16	.0760	.009201	64.4	13.28	9.76	23.04
56	13.00	.0769	13.20	.0758	.009550	66.8	13.52	10.13	23.64
57	13.02	.0768	13.23	.0756	.009912	69.3	13.76	10.50	24.25
58	13.05	.0766	13.26	.0754	.01028	71.9	14.00	10.89	24.88
59	13.07	.0765	13.30	.0752	.01066	74.6	14.24	11.28	25.52
60	13.10	.0763	13.33	.0750	.01105	77.3	14.48	11.69	26.18
61	13.12	.0762	13.36	.0748	.01146	80.2	14.72	12.12	26.84
62	13.15	.0760	13.40	.0746	.01188	83.2	14.97	12.56	27.52
63	13.17	.0759	13.43	.0745	.01231	86.2	15.21	13.01	28.22
64	13.20	.0758	13.47	.0743	.01276	89.3	15.45	13.48	28.93
65	13.22	.0756	13.50	.0741	.01323	92.6	15.69	13.96	29.65
66	13.25	.0755	13.54	.0739	.01370	95.9	15.93	14.46	30.39
67	13.27	.0754	13.58	.0737	.01420	99.4	16.18	14.97	31.15
68	13.30	.0752	13.61	.0735	.01471	103.0	16.42	15.50	31.92
69	13.32	.0751	13.65	.0733	.01524	106.6	16.66	16.05	32.71
70	13.35	.0749	13.69	.0731	.01578	110.5	16.90	16.61	33.51
71	13.38	.0747	13.73	.0729	.01634	114.4	17.14	17.19	34.33
72	13.40	.0746	13.76	.0727	.01692	118.4	17.38	17.79	35.17
73	13.43	.0745	13.80	.0725	.01751	122.6	17.63	18.41	36.03
74	13.45	.0743	13.84	.0723	.01813	126.9	17.87	19.05	36.91
75	13.48	.0742	13.88	.0720	.01877	131.4	18.11	19.71	37.81
76	13.50	.0741	13.92	.0718	.01942	135.9	18.35	20.38	38.73
77	13.53	.0739	13.96	.0716	.02010	140.7	18.59	21.08	39.67
78	13.55	.0738	14.00	.0714	.02080	145.6	18.84	21.80	40.64
79	13.58	.0736	14.05	.0712	.02152	150.6	19.08	22.55	41.63
80	13.60	.0735	14.09	.0710	.02226	155.8	19.32	23.31	42.64

TABLE 54T03 (cont) Thermodynamic Properties of Air based on one pound of air at a total pressure of 29.921 in.hg. (Atmospheric pressure)

	DRY AIR		MIXTURES OF DRY AIR AND SATURATED WATER VAPOR						
	Specific Volume	Density	Specific Volume	Density	Moisture Content (per lb. of dry air)		Heat Content (Enthalpy) Btu/lf of air and moisture		
°F	Cu.ft/lb	lbs/cu.ft.	Cu.ft/lb	lbs/cu.ft	Pounds	Grains	Sensible	Latent	Total
81	13.63	.0734	14.13	.0708	.02303	161.2	19.56	24.11	43.67
82	13.65	.0733	14.17	.0706	.02381	166.7	19.80	24.92	44.72
83	13.68	.0731	14.22	.0703	.02463	172.4	20.04	25.76	45.80
84	13.70	.0729	14.26	.0701	.02547	178.3	20.29	26.62	46.91
85	13.73	.0728	14.31	.0699	.02634	184.4	20.53	27.51	48.04
86	13.75	.0727	14.35	.0697	.02723	190.6	20.77	28.43	49.20
87	13.78	.0726	14.40	.0694	.02815	197.0	21.01	29.38	50.39
88	13.80	.0725	14.45	.0692	.02910	203.7	21.25	30.35	51.61
89	13.83	.0723	14.50	.0690	.03008	210.6	21.50	31.36	52.86
90	13.86	.0721	14.55	.0687	.03109	217.6	21.74	32.39	54.13
91	13.88	.0720	14.60	.0685	.03213	224.9	21.98	33.46	55.44
92	13.91	.0719	14.65	.0683	.03320	232.4	22.22	34.59	56.78
93	13.93	.0718	14.70	.0680	.03430	240.1	22.46	35.69	58.15
94	13.96	.0716	14.75	.0678	.03544	247.1	22.71	36.86	59.56
95	13.98	.0715	14.80	.0676	.03662	256.3	22.96	38.06	61.01
96	14.01	.0714	14.86	.0673	.03783	264.8	23.19	39.30	62.48
97	14.03	.0713	14.91	.0671	.03908	273.6	23.43	40.57	64.00
98	14.06	.0711	14.97	.0668	.04036	282.5	23.67	41.88	65.55
99	14.08	.0710	15.02	.0666	.04169	291.8	23.91	43.24	67.15
100	14.11	.0709	15.08	.0663	.04305	301.3	24.16	44.63	68.79
101	14.14	.0707	15.14	.0660	.04446	311.2	24.40	46.07	70.47
102	14.16	.0706	15.20	.0658	.04591	321.4	24.64	47.54	72.18
103	14.19	.0705	15.26	.0655	.04741	331.9	24.88	49.07	73.95
104	14.21	.0704	15.33	.0652	.04895	342.7	25.13	50.64	75.77
105	14.24	.0702	15.39	.0650	.0505	354.0	25.37	52.26	77.63
106	14.26	.0701	15.46	.0647	.0522	365	25.61	53.92	79.53
107	14.29	.0700	15.52	.0644	.0539	377	25.85	55.64	81.49
108	14.31	.0699	15.59	.0641	.0556	389	26.09	57.41	83,50
109	14.34	.0697	15.66	.0639	.0574	402	26.33	59.23	85.57
110	14.36	.0696	15.73	.0636	.0593	415	26.58	61.11	87.69
111	14.39	.0695	15.80	.0633	.0612	428	26.82	63.04	89.86
112	14.41	.0994	15.87	.0630	.0631	442	27.06	65.04	92.10
113	14.44	.0693	15.95	.0627	.0652	456	27.30	67.10	94.40
114	14.46	.0692	16.02	.0624	.0673	471	27.55	69.22	96.77
115	14.49	.0690	16.10	.0621	.0694	486	27.79	71.40	99.10
116	14.52	.0689	16.18	.0618	.0717	502	28.03	73.65	101.68
117	14:54	.0688	16.26	.0615	.0739	518	28.27	75.97	104.24
118	14.57	.0686	16.35	.0612	.0763	534	28.51	78.36	106.87
119	14.59	.0685	16.43	.0609	.0788	551	28.76	80.80	109.56
120	14.62	.0684	16.52	.0605	.0813	569	29.00	83.37	112.37

TABLE 54T03 (cont) Thermodynamic Properties of Air based on one pound of air at a total pressure of 29.921 in.hg. (Atmospheric pressure)

This table, and in fact the whole psychrometric chart, is based on standard atmospheric pressure, also called barometric pressure, of 29.921 inches of mercury. This is the total pressure of the mixture of dry air and water vapor, each of which has its own partial pressure, and these pressures must be added together to get the total pressure. If we were to add water vapor to a cubic foot of dry air at standard atmospheric pressure, the total pressure would then be greater than 29.921 inches of mercury. To balance this and keep the same total pressure of 29.921 inches of mercury, we would have to let a little air out before we add the water vapor.

Since water vapor is very light compared to air, if we let the heavier air out and replace it with the lighter water vapor, the density of the mixture will be less than that of the dry air alone. If we look at the density of air at 60°F on Table 54T03, we find that its density at saturation is .0750 lbs/cu ft, while the density of dry air at 60°F is .0763 lbs/cu ft, or .0013 lbs more than the saturated air.

VAPOR PRESSURE

Water, like any other liquid such as ammonia, sulfur dioxide, or other refrigerants, has different boiling points at different pressures. These pressures are very low, however, compared to most refrigerants. Table 54T06 shows the pressure-temperature relationship for water. The pressures are in inches of mercury absolute (not vacuum) and the temperatures are in degrees Fahrenheit.

Saturation Temperature °F	Vapor Pressure In.hg.abs.	Specific Volume cu.ft/lb.	Weight of Vapor		Heat Content (Enthalpy) Btu/lb		
			Density lbs/cu.ft	Abs. Humidity Grains/cu.ft	Sensible	Latent	Total
32	.1803	3305	.0003026	2.1	0.0	1075.2	1075.2
33	.1878	3180	.0003145	2.2	1.0	1074.6	1075.6
34	.1955	3062	.0003274	2.3	2.0	1074.0	1076.0
35	.2034	2948	.0003394	2.4	3.0	1073.5	1076.5
36	.2117	2839	.0003523	2.5	4.0	1072.9	1076.9
37	.2202	2734	.0003662	2.6	5.0	1072.4	1077.4
38	.2290	2634	.0003803	2.7	6.0	1071.8	1077.8
39	.2382	2538	.0003942	2.8	7.0	1071.2	1078.2
40	.2477	2455	.0004090	2.9	8.0	1070.7	1078.7
41	.2575	2357	.0004243	3.0	9.0	1070.1	1079.1
42	.2676	2272	.0004402	3.1	10.1	1069.4	1079.5
43	.2781	2190	.0004566	3.2	11.1	1068.9	1080.0
44	.2890	2112	.0004735	3.3	12.1	1068.3	1080.4
45	.3002	2037	.0004910	3.4	13.1	1067.8	1080.9
46	.3119	1965	.0005090	3.5	14.1	1067.2	1081.3
47	.3239	1896	.0005274	3.7	15.1	1066.6	1081.7
48	.3363	1829	.0005467	3.8	16.1	1066.1	1082.2
49	.3491	1766	.0005662	4.0	17.1	1065.5	1082.6
50	.3624	1704	.0005869	4.1	18.1	1065.0	1083.1
51	.3761	1645	.0006079	4.2	19.1	1064.4	1083.5
52	.3903	1589	.0006293	4.4	20.1	1063.8	1083.9
53	.4049	1534	.0006520	4.6	21.1	1063.3	1084.4
54	.4200	1482	.0006748	4.7	22.1	1062.7	1084.8
55	.4356	1431	.0006988	4.9	23.1	1062.1	1085.2
56	.4518	1383	.0007231	5.0	24.1	1061.6	1085.7
57	.4684	1336	.0007485	5.2	25.1	1061.0	1086.1
58	.4856	1292	.0007740	5.4	26.1	1060.4	1086.5
59	.5033	1249	.0008006	5.6	27.1	1059.9	1087.0
60	.5216	1207	.0008285	5.8	28.1	1059.3	1087.4
61	.5405	1167	.0008570	6.0	29.1	1058.8	1087.9
62	.5599	1129	.0008858	6.2	30.1	1058.2	1088.3
63	.5800	1092	.0009158	6.4	31.1	1057.6	1088.7
64	.6007	1056	.0009469	6.6	32.1	1057.1	1089.2
65	.6221	1022	.0009785	6.8	33.1	1056.5	1089.6
66	.6441	988.6	.001092	7.1	34.1	1055.9	1090.0
67	.6668	956.8	.001045	7.3	35.1	1055.4	1090.5
68	.6902	926.1	.001080	7.6	36.1	1054.8	1090.9
69	.7143	896.5	.001115	7.8	37.1	1054.2	1091.3
70	.7392	868.0	.001152	8.1	38.1	1053.7	1091.8
71	.7648	840.5	.001190	8.3	39.1	1053.1	1092.2
72	.7911	814.0	.001228	8.6	40.1	1052.5	1092.6
73	.8183	788.4	.001268	8.8	41.1	1052.0	1093.1
74	.8463	763.8	.001309	9.2	42.1	1051.4	1093.5
75	.8751	740.0	.001351	9.4	43.1	1050.8	1093.9

TABLE 54T06 Saturated Water Vapor

Saturation Temperature °F	Vapor Pressure In.hg.abs.	Specific Volume cu.ft/lb.	Weight of Vapor		Heat Content (Enthalpy) Btu/lb		
			Density lbs/cu.ft	Abs. Humidity Grains/cu.ft	Sensible	Latent	Total
76	.9047	717.0	.001395	9.7	44.1	1050.3	1094.4
77	.9352	694.9	.001439	10.0	45.1	1049.7	1094.8
78	.9667	673.5	.001485	10.4	46.1	1049.1	1095.2
79	.9990	652.9	.001532	10.7	47.1	1048.6	1095.7
80	1.0323	633.0	.001580	11.0	48.1	1048.0	1096.1
81	1.0665	613.8	.001629	11.4	49.1	1047.5	1096.6
82	1.1017	595.3	.001680	11.7	50.1	1046.9	1097.0
83	1.1380	577.4	.001732	12.1	51.1	1046.3	1097.4
84	1.1752	560.1	.001785	12.5	52.1	1045.7	1097.8
85	1.2136	543.3	.001841	12.9	53.1	1045.2	1098.3
86	1.2530	527.2	.001897	13.3	54.0	1044.7	1098.7
87	1.2935	511.6	.001955	13.6	55.0	1044.1	1099.1
88	1.3351	496.5	.002014	14.1	56.0	1043.6	1099.6
89	1.3779	482.0	.002075	14.5	57.0	1043.0	1100.0
90	1.4219	467.9	.002138	14.9	58.0	1042.4	1100.4
91	1.4671	454.3	.002201	15.4	59.0	1041.9	1100.9
92	1.5136	441.1	.002267	15.9	60.0	1041.3	1101.3
93	1.5613	428.4	.002334	16.3	61.0	1040.7	1101.7
94	1.6103	416.1	.002403	16.8	62.0	1040.2	1102.2
95	1.6607	404.2	.002474	17.3	63.0	1039.6	1102.6
96	1.7124	392.7	.002546	17.8	64.0	1039.0	1103.0
97	1.7655	381.5	.002621	18.4	65.0	1038.4	1103.4
98	1.8200	370.7	.002698	18.9	66.0	1037.9	1103.9
99	1.8759	360.3	.002775	19.4	67.0	1037.3	1104.3
100	1.9334	350.2	.002855	19.9	68.0	1036.7	1104.7
101	1.9923	340.4	.002938	20.6	69.0	1036.2	1105.2
102	2.0529	331.0	.003021	21.1	70.0	1035.6	1105.6
103	2.1149	321.8	.003108	21.8	71.0	1035.0	1106.0
104	2.1786	313.0	.003195	22.4	72.0	1034.4	1106.4
105	2.2440	304.4	.003285	23.0	73.0	1033.9	1106.9
106	2.3110	296.0	.003378	23.6	74.0	1033.3	1107.3
107	2.3798	288.0	.003472	24.3	75.0	1032.7	1107.7
108	2.4503	280.2	.003569	25.0	76.0	1032.2	1108.2
109	2.5226	272.6	.003668	25.7	77.0	1031.6	1108.6
110	2.5968	265.3	.003769	26.4	78.0	1031.0	1109.0
111	2.6728	258.2	.003873	27.1	79.0	1030.4	1109.4
112	2.7507	251.3	.003979	27.9	80.0	1029.9	1109.9
113	2.8306	244.6	.004088	28.6	81.0	1029.3	1110.3
114	2.9125	238.1	.004200	29.4	82.0	1028.7	1110.7
115	2.9963	231.8	.004314	30.2	83.0	1028.1	1111.1
116	3.0823	225.8	.004429	31.0	84.0	1027.6	1111.6
117	3.1703	219.9	.004548	31.8	85.0	1027.0	1112.0
118	3.2606	214.1	.004671	32.7	86.0	1026.4	1112.4
119	3.3530	208.6	.004794	33.6	87.0	1025.8	1112.8
120	3.4477	203.2	.004921	34.5	88.0	1025.3	1113.3

TABLE 54T06 (Cont.) Saturated Water Vapor

The water vapor in ordinary air is very light and has a very low pressure. If we start to cool the mixture, the water vapor gets denser (specific volume decreases) until it finally gets down to the temperature at which moisture starts to condense out of the air. The temperature at which this happens is called the dew point. We could also call this the condensing temperature or the boiling temperature, for it is the temperature at which water vapor condenses or water evaporates at its corresponding pressure.

What is commonly called the saturation pressure or condensing pressure with refrigerants is called vapor pressure when referring to water vapor. Thus, we can define the vapor pressure of water as the pressure exerted by the water vapor in a space at its corresponding dew point temperature.

Dalton's Law of partial pressures states that the total pressure exerted by a mixture of gases is the sum of the partial pressures of the gases in the mixture.

The total pressure of air, or standard atmospheric pressure, is 29.921 inches of mercury, so the partial pressure of the dry air only is 29.921 inches of mercury minus the vapor pressure at the corresponding dew point temperature.

RELATIVE HUMIDITY

These vapor pressure tables are very useful in calculating the relative humidity of air at partially saturated conditions. The term "relative humidity" refers to the ratio of the actual partial pressure of the water vapor at a given condition to its saturation pressure at the same temperature.

This can be expressed in the equation:

$RH = P_1/P_s$

Where:

P_1 = Partial pressure of the water vapor at the dew point temperature of the mixture of dry air and water vapor,

P_s = Saturation pressure of the water vapor corresponding to the dry bulb temperature.

Relative Humidity should not be confused with Specific Humidity and Absolute Humidity, which are based on the weight of water vapor per pound of dry air and per cubic foot of dry air, respectively. In other words, the term "Specific Humidity" is used in referring to the weight of water vapor in pounds or grains contained in the air at a given condition, for each pound of dry air at that same condition. The term "Absolute Humidity" refers to the pounds or grains of water vapor contained in one cubic foot of a mixture of dry air and water vapor at a given condition.

SPECIFIC HUMIDITY

The two columns in Table 54T03 headed "Moisture Content (per pound of dry air)" refer to the amount of moisture by weight that is required to saturate one pound of dry air at the given dew point temperature. This weight of water is expressed either in pounds of water per pound of dry air or in grains of water per pound of dry air, and is called the Specific Humidity, or Humidity Ratio. A grain of water is approximately one drop, and there are 7,000 grains of water to one pound of water. Therefore, to convert grains of water to pounds of water, we simply divide the number of grains by 7,000. The term "Grains of Moisture" is used in psychrometric calculations because the larger numbers are handled more easily than the six place decimals necessary to evaluate pounds of moisture per pound of dry air. A table giving pounds of moisture per 100 pounds of air could be used to simplify these calculations. To do this we would merely move the decimal point two places to the right, and make all calculations on the basis of 100 pounds of dry air rather than one pound of dry air.

PER CENT OF SATURATION

Another term which is sometimes confused with Relative Humidity is the term "Percent of Saturation" (or percentage humidity) which is 100 times the ratio of the weight of water vapor actually held, to the weight of the water vapor necessary to saturate one pound of dry air at that dry bulb temperature. This can be expressed in an equation:

Percent of Saturation = w_1/w_s

Where:

w_1 = Specific Humidity at dew point temperature of mixture of dry air and water vapor,

w_s = Specific Humidity at Saturation (the dew point equals the dry bulb).

If we want to calculate both the relative humidity and percent of saturation at a dry bulb temperature of 95°F and a dew point temperature of 60°F, we take the vapor pressures from Table 54T06 for the dew point temperature and dry bulb temperature to obtain the relative humidity and we take the specific humidities from Table 54T06, or the psychrometric chart, for these same temperatures to obtain the Percent of Saturation, as follows:

R.H. = P_1/P_s or .5216/1.6607 = 31.4%

% Saturation = w_1/w_s or .01105/.03662 = 30.2%

From this we can see there is a difference in values between these two humidities. Since the percent of saturation is based on the weight which is not affected by temperature changes, and the relative humidity is based on volume (pressure and volume are dependent upon each other) which is affected by temperature change, the percent of saturation is the more accurate of the two. However, the difference is slight, so the use of relative humidity is permissible where extreme accuracy is not of prime importance.

SPECIFIC VOLUME

To find the specific volume and density of this mixture is a little more complicated. In order to do this, we must first go back to the fundamental gas law which states that the Absolute Pressure in pounds per square foot times the Volume in cubic feet equals the Weight in pounds times the Temperature in absolute degrees Fahrenheit times the Perfect Gas Constant for that particular gas. This can be expressed as:

PV = WRT

Since air is considered a perfect gas, and the specific volume is expressed in cu ft per one pound of air, we can transform this equation to fit all conditions at 29.921 inches Hg. The resulting equation is then:

$$V_a = 53.3 \times \frac{(460 + t)}{70.7(29.921 - P_1)}$$

Where:

v_a = Specific volume of the air,

t = Dry bulb temperature of the mixture,

P_1 = Partial pressure of the water vapor at the dew point of a mixture,

53.3 = R or Gas constant for air,

(460 + t) = Absolute temperature of Air,

70.7 = Factor to convert inches Hg to pounds per square foot. Thus, for our air conditions of 95°F Dry bulb, and 60°F dew point temperature, the specific volume is:

$$V_a = 53.3 \times \frac{(460 + 95)}{70.7(29.921 - 0.5216} \text{ or } V_a = 53.3 \times \frac{555}{(2{,}078.5)} \text{ or } V_a = 14.22 \text{ cu ft / lb}$$

The density of the mixture is then:

d_a = 1/14.22 or .0702 lbs/cu ft

HEAT CONTENT (ENTHALPY)

The energy contained in air is commonly called its heat content. However, the term enthalpy has replaced heat content in engineering terminology. The enthalpy of air is the sum of the enthalpies of the dry air and its contained moisture. For dry air, throughout the entire range of psychrometric tables and charts, only sensible heat is involved. For water vapor, its enthalpy contains two sensible components—enthalpy of superheat and the heat of the liquid—plus the latent heat of vaporization. However, by common usage, since the latent heat is by far the greatest, and exact usage is extremely complicated, all energy transfer of the water is considered as latent heat transfer.

The first column in Table 54T03 under Heat Content or Enthalpy is based on a 0°F reference point, that is, the heat content or the sensible heat of dry air at 0°F is considered 0. Some tables use other base temperatures as the zero point for the enthalpy or total heat content, but since all air conditioning calculations are based on a change in heat, the end result is the same.

The sensible heat (Btu/lb of dry air) increases from 0°F at the rate of approximately .24 Btu per lb. per degree F. This is held fairly constant throughout the air conditioning temperature range, so that we can approximate the sensible heat of air by merely multiplying the dry bulb temperature in degrees Fahrenheit by .24. For example, at 75°F, the approximate sensible heat of the air is .24 x 75 = 18 Btu per pound of dry air. This compares with the more accurate sensible heat content of 18.11 Btu per pound of dry air at 75°F as read from the tables. While this method is not accurate enough for laboratory calculations, it is useful for field calculations using the psychrometric chart, where extreme accuracy is not of prime importance.

On most psychrometric charts space is limited so that only the total heat content of the air is shown. If the sensible heat or the latent heat of the air is required, it is a simple matter to find either, using these formulas:

Sensible heat = .24×°F (dry bulb temperature).

Latent heat = Total heat – (.24×°F dry bulb temperature).

For example, in calculating the desired properties of air from the tables for air with a 95° Dry Bulb temperature, a 71.5° wet bulb temperature, and a 60° dew point, the tables can only be used for the values which refer to a saturated condition. The specific humidity will not change as long as the dew point remains unchanged, so we can get this directly from the table because the term “dew point” refers to saturation. The specific humidity at a 60°F dew point is therefore 77.3 grains per pound of dry air, or .01105 lbs per pound of dry air. The total heat of the air is dependent upon the wet bulb temperature of the air, so we can pick this from Table 54T03 for the wet bulb temperature of 71.5°F or 34.75 Btu/lb. The sensible heat of this air is dependent upon the dry bulb temperature, so from the sensible heat column we obtain 22.96 Btu/lb at the 95°F dry bulb temperature. The latent heat of the mixture is then 34.75 – 22.96, or 11.79 Btu/lb.

From these examples, we can see the large amount of calculating necessary to arrive at the properties of air if we have only two or three of these properties given. While these calculations will give a better accuracy, using the tables, the use of the psychrometric chart, which is made from these tables, will greatly simplify our calculations and still give us enough accuracy in our figures for field testing purposes.

In using these tables it must be remembered that the properties of air are for a saturated condition only, and any condition other than saturation or 100% relative humility will require correction. Bear in mind that the enthalpy, the specific volume and moisture content of the air are based on one pound of dry air, while the density is based on one cubic foot of dry air. All calculations on air conditioning processes in this section will be based upon the weight of air rather than the volume.

Now, let us study a psychrometric chart. There are many types of these, each with its own advantages. Some are made for low temperatures, some for the medium temperature range, and others for the high temperature range. Some of the psychrometric charts are stretched in length and shortened in height, while others are higher than they are wide, and others are shaped in the form of a triangle. They all serve basically the same function and the chart to be used should be chosen for the temperature range and the type of application. In this case, we will use a psychrometric chart which covers a dry bulb range from 20°F to 120°F and a wet bulb range from 20°F to 95°F.

Figure 54F10A shows a skeleton psychrometric chart with the dry bulb temperature scale along the bottom of the chart. The vertical lines extending from the bottom of the chart to the top are constant dry bulb lines, that is, any and all points on one given line have the same dry bulb temperature as indicated on the bottom dry bulb scale. If we were to plot only a dry bulb temperature of 95°F on the chart it could fall on any point on the vertical line extending upward from the 95° on the dry bulb scale at the bottom of the chart.

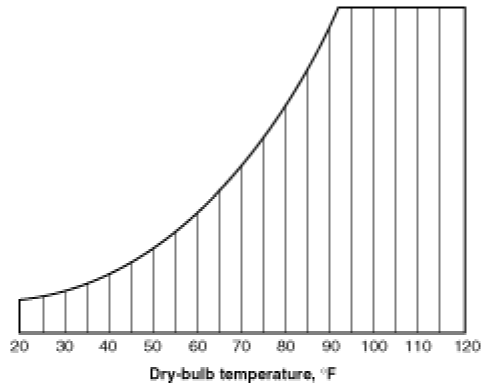

Figure 54F10A This diagram illustrates the constant dry bulb lines of a typical psychrometric chart.

Figure 54F10B shows a skeleton psychrometric chart with the wet bulb temperature scale along the outer curved line at the left of the chart. Wet bulb temperature is defined as that shown by a thermometer whose bulb is covered with a wetted wick, and with air passing over it at approximately 1000 fpm. The constant wet bulb lines run downward at an angle of approximately 30° from the Horizontal. They extend from the wet bulb scale at the left, to the right margin of the chart. All points on one given wet bulb line are at the same wet bulb temperature. If we were to plot a 75°F wet bulb temperature on the chart it could fall anywhere on the wet bulb line corresponding to the point marked 75°F on the wet bulb scale.

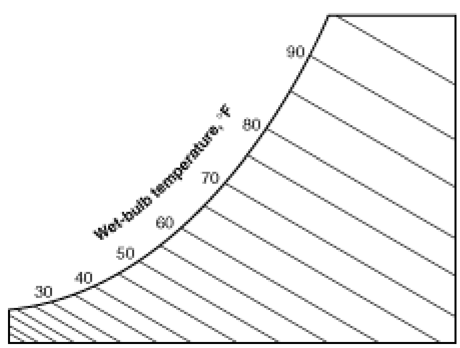

Figure 54F10B This diagram illustrates the constant wet bulb lines of a typical psychrometric chart.

Figure 54F10C shows the skeleton psychrometric chart with the dew point scale along the outer curved line at the left of the chart. You will notice that both the wet bulb and the dew point scales are the same on this chart. The constant dew point lines, however, run horizontally from the left curved line to the right. Any point on one given constant dew point line will correspond to the dew point temperature on the curved line scale. On the right hand margin of the chart is a vertical scale which is called the specific humidity scale giving the weight in grains of water vapor in one pound of dry air. The constant specific humidity lines also are horizontal and coincide with the constant dew point lines. Thus we can see that the amount of water vapor in the air is dependent upon the dew point of the air.

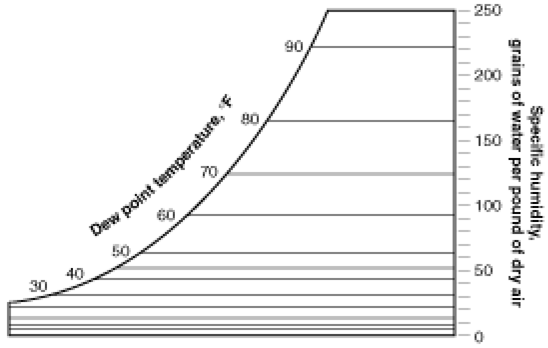

Figure 54F10C This diagram illustrates the dew point temperatures and specific humidity lines of a typical psychrometric chart.

Figure 54F11A shows the skeleton psychrometric chart with the curved constant relative humidity lines extending upward and to the right of the chart. We noted previously that the wet bulb temperatures and the dew point temperatures share the same scale along the outer curved line to the left of the chart. Since the only condition where the wet bulb temperature and the dew point are the same is at saturation, this outer curved line represents a saturation or 100% relative humidity condition. The constant relative humidity lines decrease in value as we move away from the saturation line to the right and downward. The values for these relative humidity lines are given in percentages and are shown near the right end of the constant lines.

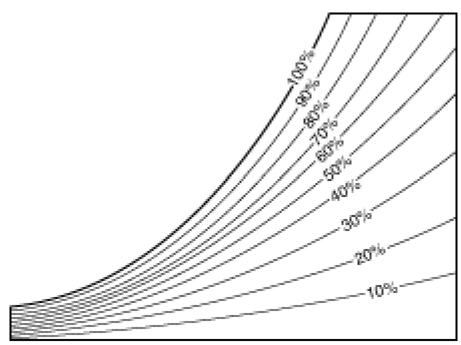

Figure 54F11A This diagram illustrates the constant relative humidity lines of a typical psychrometric chart.

Figure 54F11B shows the skeleton psychrometric chart with the constant specific volume lines. These lines are at an angle of approximately 60° from the horizontal and will increase in value as we move from left to right. On the chart we will be using, the spacing between each line represents a change in specific volume of .1 cu ft per lb. The values for these constant specific volume lines are given every fifth line and any point falling between these lines must naturally be an estimated figure. If the density of the air at any condition is desired, one merely has to divide the specific volume into (1.0) to arrive at the answer. Because most of the calculations in air conditioning work are based on the weight of air rather than on the volume of air, the use of specific volume (cu ft per one lb of air) rather than density (lbs per one cu ft of air) is advisable.

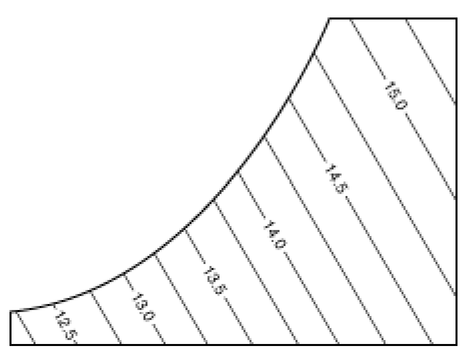

Figure 54F11B This diagram illustrates the constant specific volume lines of a typical psychrometric chart.

Figure 54F11C is a skeleton psychrometric chart showing the constant enthalpy lines. It will be noted that these lines are merely extensions of the wet bulb lines, since the total heat of the air is dependent upon the wet bulb temperature. The scale at the far left of the chart gives the total heat of the air in Btu/lb of dry air and increases from approximately 7.2 Btu/lb at 20°F wet bulb temperature up to approximately 58.6 Btu/lb at 92°F wet bulb.

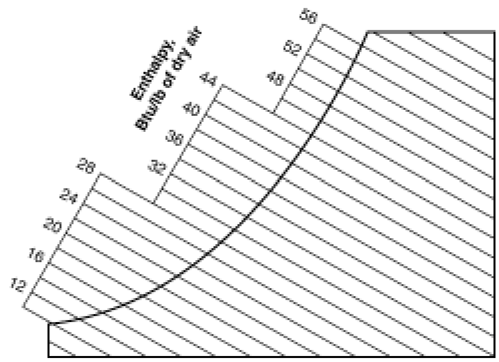

Figure 54F11C This diagram illustrates the constant enthalpy lines of a typical psychrometric chart.

Now let us take a look at the psychrometric chart which we will use for our calculations. Figure 54F13 shows the form of psychrometric chart which we will use. It was devised by the General Electric Company and is reproduced by permission. Its construction consists of all of the skeleton charts in Figures 54F10A through 54F11C, which have been superimposed on one chart. That is, all the constant lines which appear in the six skeleton psychrometric charts appear in this one chart and have the same relative positions on this chart.

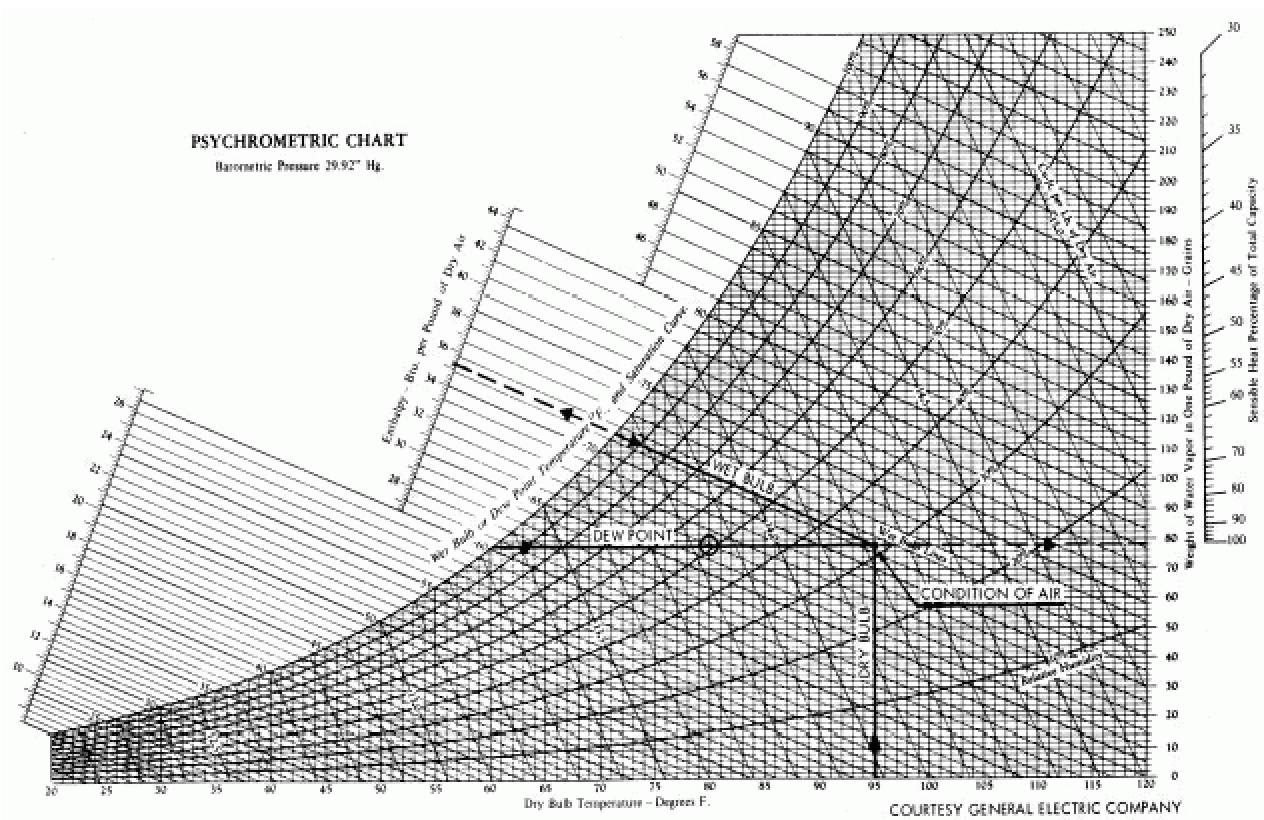

Figure 54F13

This chart shows the condition of air plotted from 95°F dry bulb, 71.5°F wet bulb, and 60°F dew point. Other conditions indicated at this point are 32% relative humidity, a specific volume of 14.22 cu ft per lb, 78 grains of moisture per lb and a total heat content of 35.25 Btu per lb.

In describing the construction of the psychrometric chart previously, we defined the constant line as a line which can contain an infinite number of points, each at the same condition; that is, if we were to plot a single condition of air, such as the dry bulb temperature on a psychrometric chart, it could fall at any point on the constant line corresponding to that dry bulb temperature.

We now have on the composite psychrometric chart a number of lines which cross each other, so if we plot a point on a constant dry bulb line, that point will also correspond to different values on the constant lines for wet bulb temperature, dew point, relative humidity, specific volume, and total heat. Since any given two of these constant lines will cross at only one point on the chart, we can plot this point exactly, if we know any two of the properties of the air. From this point, we can then move along the respective constant lines for the other properties of air and we can read them from their respective scales without going to the trouble of calculating them as before. While this method will not be quite as accurate as the calculation method, it is much faster and our degree of accuracy will be close enough for our purposes.

FINDING THE PROPERTIES OF AIR

In our previous example we calculated the properties of air at a 95°F dry bulb temperature, 71.5° wet bulb temperature and 60°F dew point. We will now apply this same problem to the psychrometric chart.

Figure 54F13 shows the composite chart with the dry bulb temperature of 95°F plotted vertically on the constant dry bulb temperature line extending from the 95°F on the dry bulb scale. The wet bulb line is then plotted from 71.5°F and runs parallel to the constant wet bulb lines. The point where these two lines cross is the only point on the chart at which both of these conditions exist. We need only two air conditions to plot this point, although on the chart the dew point temperature is also shown. If we did not know the dew point of the air, we could find it by merely moving from the point which represents our air condition to the left parallel to the constant dew point lines. The point at which this intersects the dew point scale is the dew point temperature.

We can see by a close examination of this point that it will fall about 1/5 of the distance between the 30% and the 40% relative humidity lines. We can estimate the relative humidity to be approximately 32%. If we look back at our calculations for this condition, we find that the relative humidity is actually 31.4%. This is close enough for normal purposes.

The relationship of this point with respect to the constant volume lines indicates that the point lies approximately one-fifth of the distance between the second and third lines after the 14.0 constant volume line. Since each of these unmarked lines indicates a change of .1 cu ft per pound, we can again estimate that the specific volume is 14.0, plus 2 times 1, plus about one-fifth of .1, or 14.22 cu ft per pound. Again checking with our previous calculations, we find that our estimate coincides exactly with the calculations.

Since our specific humidity constant lines are parallel with the dew point constant lines, we merely extend the constant dew point line for 60°F to the specific humidity scale at the right of the chart. This is indicated by the dotted line extending to the right on the chart. This line intersects the scale at a point which represents approximately 78 grains per pound. If we wish to convert this to pounds of moisture per pound of dry air, we merely divide the 78 grains of moisture per pound of dry air by 7,000. We then have a figure of .01112 pounds of moisture per pound of dry air. This, compared to the corresponding value in Table 54T03, .01105, shows a discrepancy of only .00007 lbs of moisture per lb of dry air.

By extending the constant wet bulb line of 71.5 straight out to the total heat scale, as indicated by the dotted line, we can read the total heat content of the air to be 35.25 Btu per pound of dry air. Again, the comparison with the figure from the tables for that wet bulb temperature is 34.75 Btu per pound, or a discrepancy of .50 Btu per pound. This indicates approximately the manner in which the wet bulb line departs from being a constant enthalpy line. While this would seem to be considerably less accurate than the other comparisons with the calculations and the tables, it should be remembered that in the air conditioning processes we are interested in a change of heat rather than the absolute values of total heat. This difference between the tables and the chart is consistent throughout the range of temperatures with which we will be working, so that the changes in total heat values from the chart will be almost identical to those from the tables.

SENSIBLE AND LATENT HEAT

In heating and cooling air from conditions which are undesirable to conditions which are suitable for human comfort, we must take into consideration the addition or removal of two types of heat: Sensible Heat and Latent Heat.

The term “Sensible Heat Change” refers to a change in heat which will affect the temperature of the air. Quite often, the cooling of hot dry desert air or the heating of cold air will require only a change in the sensible heat of the air. Since a change in sensible heat only will not affect the amount of moisture in the air, this change may be plotted on the psychrometric chart, parallel to the constant dew point lines and coinciding with the constant dew point line for the air at its original condition. This means that the dew point of the air will not change as long as sensible heat only is removed or added.

SENSIBLE COOLING AND HEATING

Figure 54F15 shows the sensible cooling of air from 110°F dry bulb temperature and 70°F wet bulb temperature to 67°F dry bulb and 55°F wet bulb. In comparing properties of the initial air condition with those of the final condition, we can see that we have increased the relative humidity of the air from approximately 12% to approximately 47%, although we have not changed the moisture content of the air. This is because, in cooling the air, we have reduced the amount of water which it could hold at saturation, and consequently have increased the ratio of the moisture in the air, to the maximum which could be held at that dry bulb temperature.

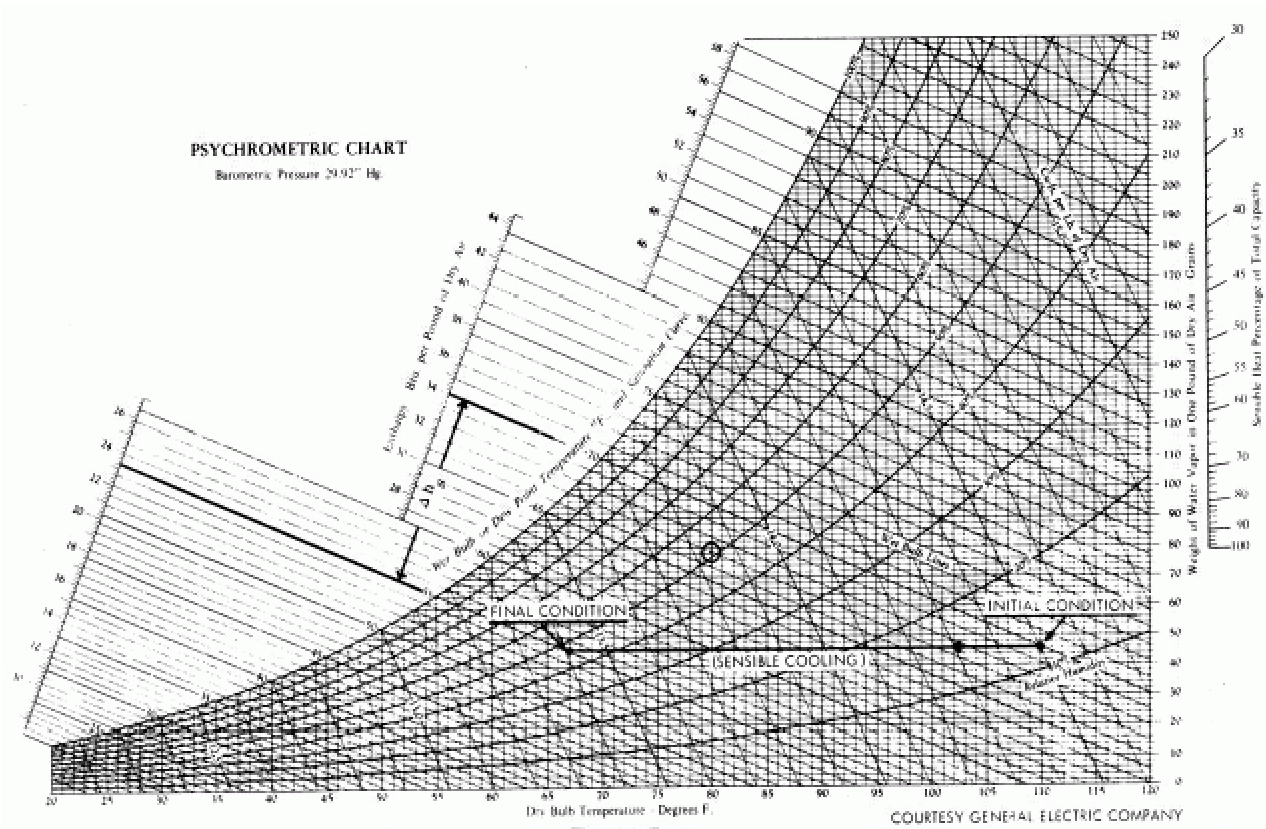

Figure 54F15

This chart shows the sensible cooling of air plotted from 110°F dry bulb and 70°F wet bulb temperature to 67°F dry bulb and 55°F wet bulb. Note increase in relative humidity although moisture content remains constant. Arrows show loss of heat content of the air. As the condition changes, the specific volume decreases.

We can also see that, in cooling the air, we have decreased the specific volume of the air and increased its density. At a given moisture content, the colder the air, the greater the density.

In plotting the change in enthalpy for this sensible cooling effect, we can see that the initial condition contains 34.0 Btu/lb, while the final condition contains approximately 23.3 Btu/lb. If we subtract the heat content at the final condition from the heat content at the initial condition, we arrive at a total change in enthalpy of 10.7 Btu/lb. Therefore, from each pound of air that we cool from the initial condition to the final condition we must remove 10.7 Btu. This change in sensible heat is shown in Figure 54F15 by the symbol Δh_s.

Although the example used in Figure 54F15 shows a sensible cooling process only, the calculations for the exact opposite of this process, as in winter heating systems, are the same. That is, each pound of air which is heated from 67°F dry bulb and 55°F wet bulb to 110°F dry bulb and 70°F wet bulb will require the addition of 10.7 Btu. Any source of dry heat such as a furnace heat exchanger, hot water coil, or electric heater, will produce a change in sensible heat only. In the cooling process, however, the average outside surface of the cooling coil must be above the dew point temperature of the air, or moisture will condense, resulting in a transfer of its latent heat.

HUMIDIFICATION OF AIR

Figure 54F16 shows a humidification process, which is a change in latent heat only, plotted on a psychrometric chart. If we take air at the initial condition of 90°F dry bulb and 60°F wet bulb and change the conditions to 90°F dry bulb and 90% relative humidity, we must add latent heat to the air without changing the sensible heat content (dry bulb temperature). From the specific humidity scale on the right, we can determine that we must add 167 grains (196-29), or .0238 pounds of water to each pound of air which we want to humidify.

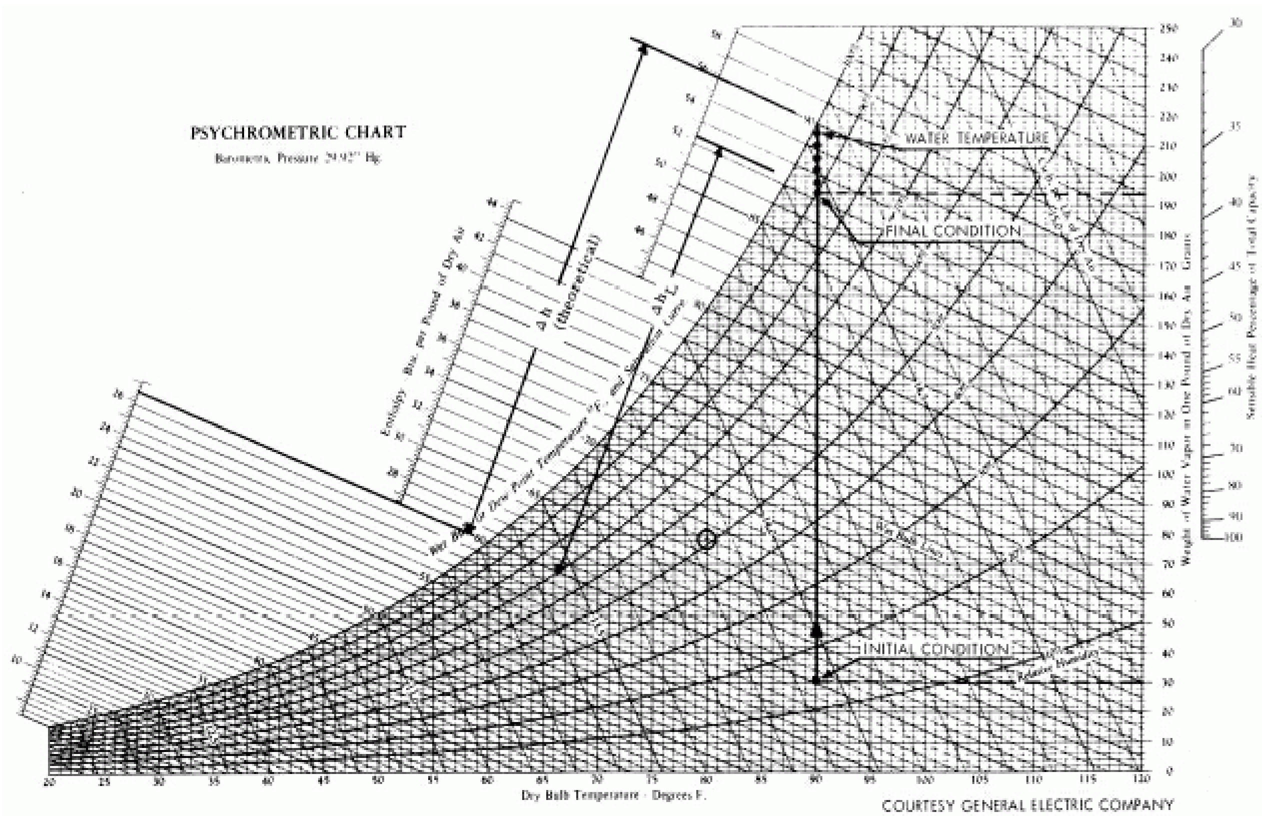

Figure 54F16

This chart shows the humidification process. The condition of the air changes from 90°F dry bulb and 60°F wet bulb to 90°F dry bulb at 90% relative humidity. The dotted line represents the by-pass factor. Since the water spray is saturated, its condition falls on the saturation curve or 100% humidity line.

In order to effect this change of properties in the air, we must introduce spray water into the air at the same temperature as the dry bulb temperature of the air. We can plot the water temperature on the psychrometric chart on the saturation line at 90°F dew point. A straight line drawn from the initial condition to the water temperature on the saturation curve will approximate the changing conditions of the air as it is humidified. Heat must be added to the water to maintain it at 90°F, at the rate of approximately 1076 Btu per lb of water evaporated to make up for the latent heat of vaporization.

The final condition of the air is determined by the efficiency of the spray, or as it is more commonly called, its HUMIDIFYING EFFICIENCY, or HUMIDIFYING EFFECTIVENESS. The humidifying effectiveness is expressed as a percentage and may be found by dividing the Actual total heat change (Δh_L) by the Maximum possible total heat change (Δh_{Max}). In this example:

$$Humidifying\ Eff. = \frac{\Delta h_L}{\Delta h_{max}} = \frac{(52.2-26.4)}{(55.7-26.4)} = .88 \text{ or } 88\%$$

While the humidifying effectiveness is very useful in the understanding of the spray process, the term “By-pass Factor”, which is merely the Humidifying Effectiveness expressed as a decimal, subtracted from one (1.0), is of much more use. This can be expressed as an equation:

By-pass Factor = 1.0 – Humidifying Effectiveness.

Quite often the equipment manufacturer will rate his equipment at various air volumes or air velocities, and will also give the by-pass factor for the different air velocities.

In general, the higher the air velocity through the equipment, the higher the by-pass factor will be. The use of the by-pass factor is also very useful for shortcuts in some of the calculations. For example:

A given piece of spray apparatus is to be installed in a city which has design conditions of 105°F dry bulb and 85°F wet bulb. With an available water supply of a constant 75°F and an apparatus by-pass factor of 20%, what will be the leaving air conditions?

Figure 54F18 shows the initial conditions and the water temperature plotted with the straight line between them approximating the air condition as it moves through the spray. Since the constant total heat lines practically coincide with the wet bulb lines, we can multiply the maximum difference in wet bulb temperatures by the by-pass factor and add this to the saturation temperature to arrive at the final condition wet bulb temperature. Thus, (85-75)×.2 = 2°F by-pass in wet bulb temperature. Then, 75°F + 2°F = 77°F final wet bulb temperature.

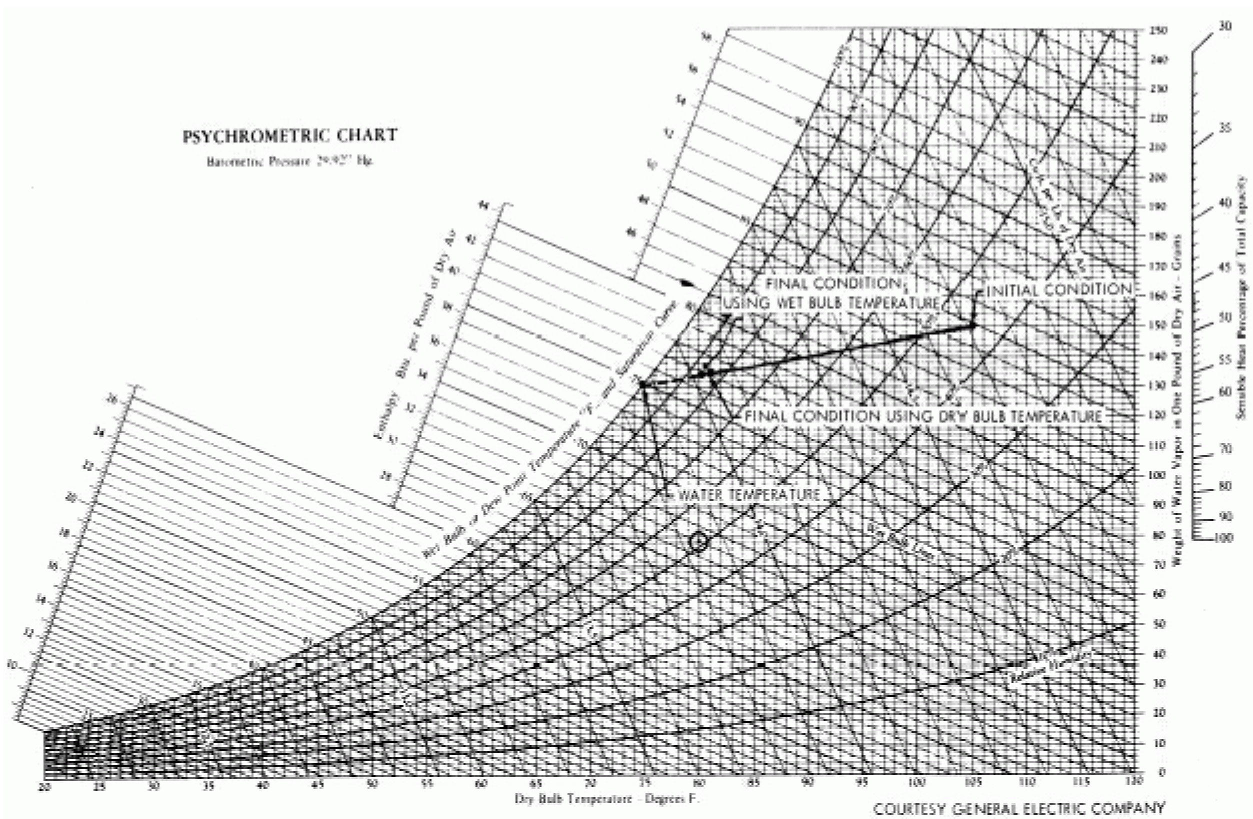

Figure 54F18

This chart shows the condition change of the air as it moves through a cold water spray at 75°F. The dotted line represents the difference in using the wet bulb and dry bulb changes in calculating the percentage, or humidifying efficiency.

If we then find where the 77°F constant wet bulb line crosses the line drawn for the conditioned air change, this is our final condition at 80.25°F dry bulb and 77°F wet bulb. It is interesting to note that if we were to use the dry bulb temperature as a basis for by-pass factor instead of wet bulb temperature, the resultant final condition would be 81°F dry bulb and 77.25°F wet bulb. Since this is very close to our calculations with wet bulb temperature, the use of dry bulb temperature in these calculations is sometimes used. In fact, if our condition line were to run parallel, or follow the constant wet bulb lines, we would have to use the dry bulb temperatures for calculating the by-pass. On the other hand, the example in Figure 54F16 shows the condition line following the constant dry bulb line for 90°F, so we must use the wet bulb temperature to calculate the final conditions.

Again, our method is the simplified approximate way to find these conditions, and we will lose some accuracy. In determining which scale is best for these by-pass calculations, it would be well to keep in mind that the maximum accuracy will occur when our condition line is parallel to the scale, so we can choose our scale accordingly.

COOLING AND DEHUMIDIFICATION

The cooling and dehumidification of air for human comfort is commonly called air conditioning. This is not entirely correct, for the term “Air Conditioning” refers to any and all phases of cooling, heating, ventilating, filtering, distribution, etc. of air to meet the requirements of the conditioned space. The cooling and dehumidification of air is the phase of air conditioning which concerns the refrigeration service engineer, for it normally requires the use of mechanical refrigeration equipment. In order to produce the required cooling and dehumidification for the conditioned space, the refrigeration equipment must be working properly and have the correct capacity for the application.

How many times have we heard the unhappy customer say, “Sure, the equipment runs all right, but it just doesn’t seem to be doing enough cooling.” The calculations for determining the heat load on the space and air distribution requirements are covered elsewhere in the manual, so we will not attempt to cover those in this chapter, but we can, with the help of the psychrometric chart, make a quick check to see how much capacity our refrigeration equipment has, under actual working conditions.

The tools needed for these calculations consist of a sling psychrometer; a device for measuring air velocities (such as an anemometer or a velometer); a ruler; and a psychrometric chart.

For example, we will check the capacity of an air conditioning system which is rated by the manufacturer at 120,000 Btu/hr at ASRE Test Conditions. While we will normally not use these test conditions in our capacity test, we can approximate them by using 100% return air entering the evaporator on a hot summer day, and adjusting the fan speed to give an air velocity of approximately 500 ft per minute over the face area of the coil.

The first thing we do is to take the wet and dry bulb temperatures of the air entering and leaving the evaporator coil. In this example, we will use an entering air or initial condition of 80°F dry bulb and a 67°F wet bulb, and a leaving or final condition of 50°F dry bulb and a 48.5°F wet bulb. The air velocity over the coil is taken by mentally dividing the coil face into approximately 2 to 3 inch squares, taking a reading at each square, recording the reading, and averaging these readings by simply adding them up and dividing by the total number of readings. The more readings that are taken, the greater will be the accuracy. In our example we will use an average air velocity of 517 fpm and a coil which is 36 inches long and 18 inches high.

The first step in calculating the capacity of the system is to plot the air conditions entering and leaving the coil on the psychrometric chart as in Figure 54F19. Point 1 represents the entering air condition; Point 2 represents the leaving air condition; and Point 3 represents the approximate effective coil temperature. Point 3 is found by merely extending the straight line connecting Points 1 and 2 to the saturation curve. This point, also commonly called the apparatus dew point, is the average temperature of the water which is condensed on the coil surface.

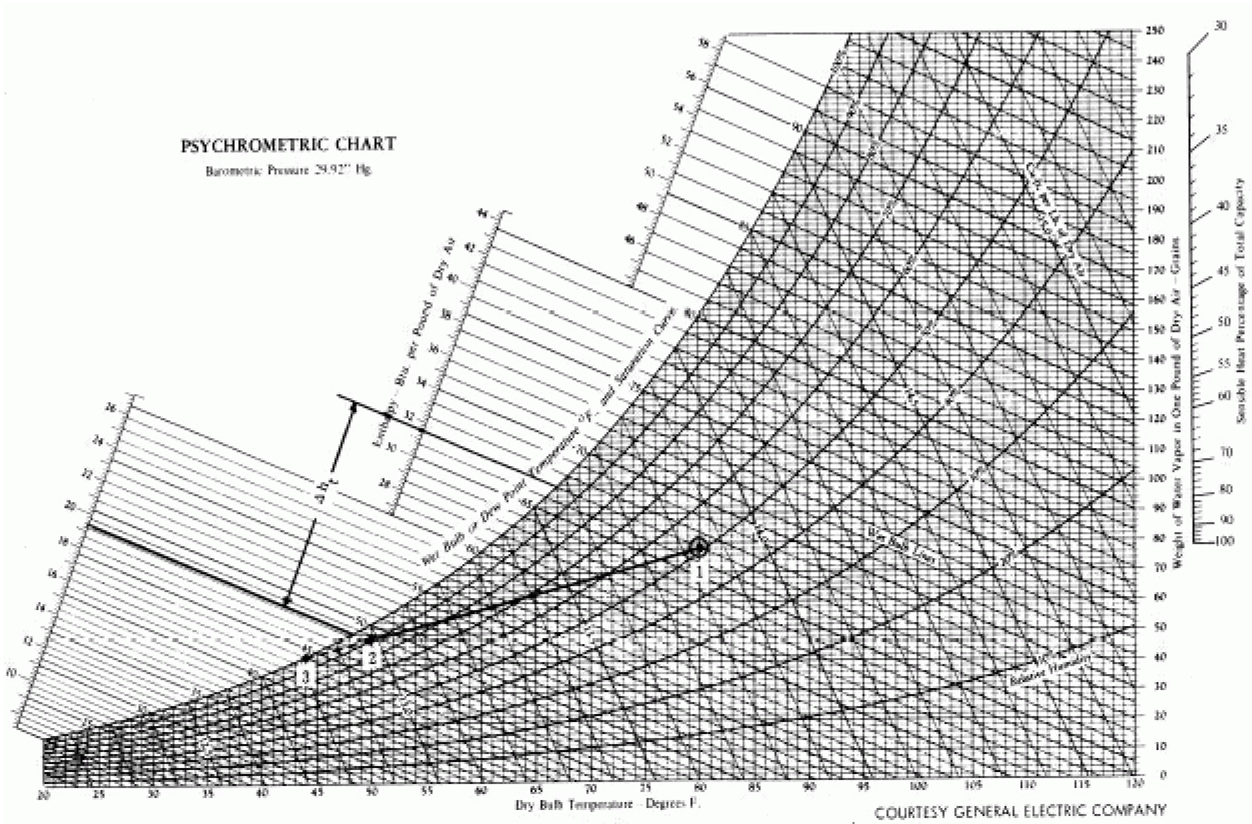

Figure 54F19

This chart indicates both sensible and latent heat removal. As the condition moves toward the left on the chart, both heat and moisture are removed from the air. The heavy line indicates the condition of the air, and Point 3 represents the effective coil surface temperature.

Next, we find the total heat change in one pound of air by subtracting the enthalpy of the leaving condition from the enthalpy of the entering condition:

Δh_t = 31.5-19.4 = 12.1 Btu/lb.

Since the volume of air over the coil is controlled by the fan, and since this air will change in density and specific volume as the temperature is changed through the system, our next step will be to determine the total weight of air circulated by the fan. The weight of air will not change since matter cannot be created or destroyed.

The coil face area is 36 inches by 18 inches, or 36×18/144 = 4.5 sq ft. If we multiply this by the face velocity of the air over the coil in ft per minute, we then have a figure of 4.5 × 517 = 2,325 cu ft per minute. Now in order to convert this volume of air to weight, we will divide the cfm by the specific volume of the air at the conditions entering the coil, for we must always make our calculations for the point at which our air velocity measurements were taken. A look at the chart shows that our entering condition lies a little less than halfway between the 13.8 and the 13.9 constant volume line. We can estimate this at 13.84. Thus, we have a total weight of air being circulated of 2,325/13.84 or 168 pounds per minute.

Now, from our previous calculations from the psychrometric chart, we arrived at a change in enthalpy of 12.1 Btu per pound of air and we have 168 pounds of air being circulated per minute. Multiplying these two figures together will give us the enthalpy change in the air per minute or 168 × 12.1 = 2,032.8 Btu/min. If we multiply this figure by the number of minutes in one hour (2,032.8 × 60) we will have the total capacity of the equipment under actual conditions or 121,968 Btu per hour, or approximately 10.15 tons of refrigeration (121,968)/(12,000).

In order to simplify the calculations, the following formula can be used:

$$Q_t = \frac{A \times V \times \Delta h_t \times 60}{v} \, Btu/hr$$

Where:

A = Face Area of coil in sq ft.

Y = Air Velocity entering coil in ft. per min.

Δh_t = Change in heat content from psychrometric chart (Btu/lb)

v = Specific volume of air entering the coil (cu ft/lb)

SENSIBLE AND LATENT HEAT CHANGES

Sometimes it is desirable to calculate the sensible heat and latent heat changes in the air conditions as shown in Figure 54F19. In the previous examples using Figures 54F15 and 54F16, it was shown that when plotting a pure sensible heat change on the psychrometric chart, the result was a horizontal condition line, and the pure latent heat change resulted in a vertical condition line. The condition line in Figure 54F19 is neither vertical nor horizontal, but falls somewhere between these two. If we drop a vertical line, parallel to the constant dry bulb lines, from point one, and draw a horizontal line parallel to the constant dew point lines, the three lines will then form a right triangle. The lengths of the horizontal and vertical lines will represent the two components of the total heat; sensible heat and latent heat. If we then draw a line parallel to the constant wet bulb lines from the intersection of the horizontal and vertical components, to the total heat scale, it can then be seen that the total heat is broken down into two components. The lower component on the scale is the sensible heat change and the upper part is the latent heat change. For example, Figure 54F21 again shows the condition line for entering air temperatures of 80°F dry bulb and 67°F wet bulb and leaving air condition of 50°F dry bulb and 48.5°F wet bulb. The enthalpy change is 31.5 - 19.4 or 12.1 Btu per pound. If we draw the horizontal and vertical components of the condition line, and then, from their intersection, draw the line parallel to the constant wet bulb lines to the total heat scale, we find that it intersects at 27.0 Btu per pound. The sensible heat change of the air is then found by subtracting the heat content at Point 2, 19.4 Btu per pound, from the 27.0 Btu per pound, which gives 7.6 Btu/pound. The latent heat change is then found by subtracting the 27.0 Btu/lb from the total heat at Point one, 31.5 Btu/lb, which gives 4.5 Btu/lb.

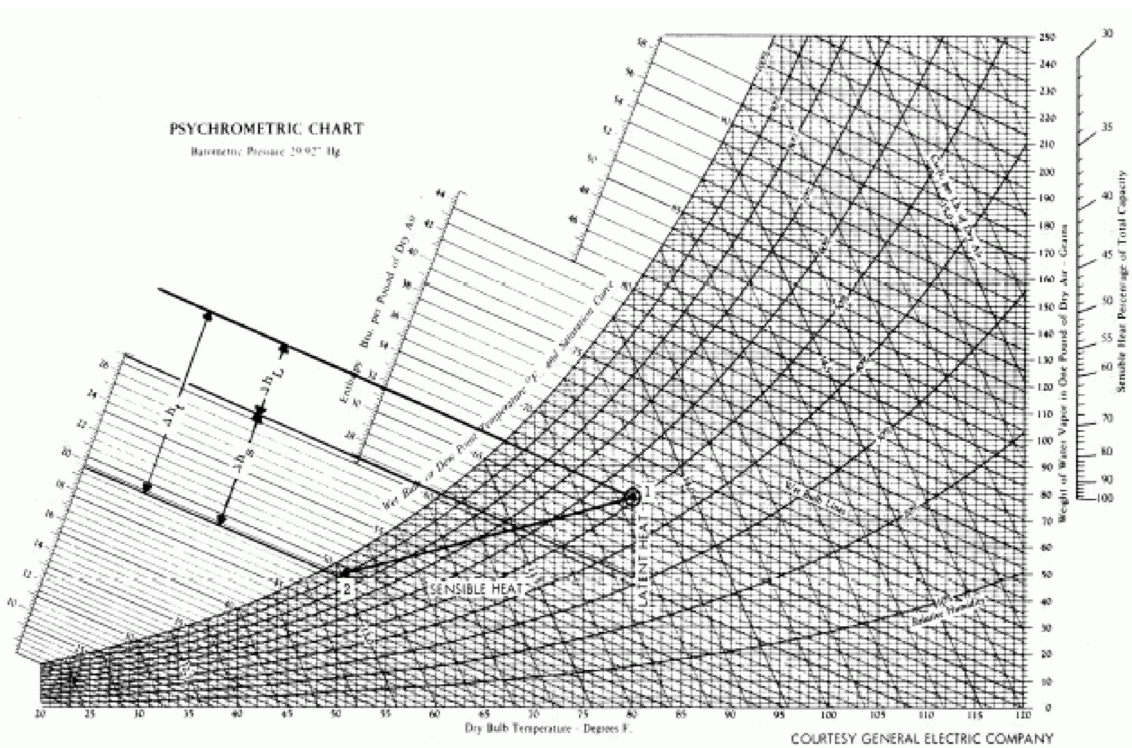

Figure 54F21

This chart indicates the same conditions as Figure 54F19, with the total heat removed being broken down into its two components, sensible and latent heat. Sensible heat changes are all shown by a horizontal line, latent changes by a vertical line.

MOISTURE REMOVAL

Quite often the service engineer will require information on the amount of moisture removed from the air by an air conditioning system. The consumer or customer who very seldom is familiar with the meaning of technical terms, such as Btu per hour or tons of refrigeration, will have a better understanding of the equipment if he knows that the equipment can remove so many gallons of water from the air while it is cooling the air so many degrees Fahrenheit.

To find this amount of water removal, the total weight of air circulated must be calculated, as in the capacity calculations. The constant dew point lines for both the initial and final air conditions are then traced to the specific humidity scale at the right margin. For example, the conditions in Figure 54F21 will give a change in specific humidity of 30 grains per pound of air (78-48). Multiplying the 30 grains per pound by the total weight of air circulated per hour, will give the total moisture removed in grains. To convert the number of grains to pounds we merely divide by 7,000, or if we want to convert grains directly to gallons, we divide the number of grains by 58,310 (7,000 x 8.33). In this case, 168 pounds of air per minute times 60 minutes per hour equals 10,080 pounds per hour times 30 grains of moisture equals 302,400 grains per hour, or 43.2 pounds per hour, which equals nearly 5.2 gallons of water per hour.

MIXING AIR AT DIFFERENT CONDITIONS

In air conditioning it is quite often necessary to mix air at different wet bulb and dry bulb temperatures to achieve a given final air condition. Most of the commercial air conditioning applications require a certain volume of fresh, outdoor air to be introduced into the occupied space. Most state and local codes require anywhere from 7.5 cfm per person to 15 cfm per person to prevent odors, etc. from being recirculated in the occupied space. Since the introduction of 100% outside air is not usually practical from a cost of operation standpoint, we therefore will mix the required outdoor air with a percentage of recirculated air prior to cooling or heating it.

The mixing of two quantities of air at different temperatures and moisture contents is also used extensively in air conditioning where constant supply air conditions are required regardless of inlet air conditions. In this method, a portion of the incoming air is bypassed around a cooling coil or heating coil and then re-mixed with the treated air to provide the desired conditions.

In both of these mixing processes, the final or resultant condition will depend upon both the weight and the temperature of each individual air quantity. Again, for absolute accuracy the weights of the mixtures should be used, although for small differences in temperatures the ratios of the individual cfm to the total cfm can be used for field calculations.

For example, if 25 cfm of outside air at 95°F dry bulb and 75°F wet bulb is mixed with 75 cfm at 80°F dry bulb and 67°F wet bulb of recirculated air, the total cfm of air is 100 cfm. This will give a mixture ratio of 25% outside air and 75% re-circulated air on the basis of volume.

If the weights of these air quantities are calculated, the weight of the outdoor air is (1/14.29) × 25, or 1.75 pounds per minute, the weight of the re-circulated air is (1/13.84) × 75, or 5.41 pounds per minute, and the total weight of air is 1.75 + 5.41, or 7.16 pounds per minute. This results in percentages of 24.42% of outside air and 75.58% of re-circulated air on the basis of weight.

The difference in our percentages, using cfm as a basis, is less than 1/2 of one percent, and at a 15° temperature difference this results in an error of only .075 degrees. Since we cannot read the field thermometer with this accuracy, we are safe to use the cfm as a basis for mixture calculations for a relatively small temperature difference.

Figure 54F23 shows a psychrometric chart with air conditions of 95°F dry bulb, 75°F wet bulb, and 80°F dry bulb, 67°F wet bulb connected by a straight line. This line represents the temperature path of the mixture of these two air conditions in any proportion. Each end point on the line represents 100% of the mixture at that temperature. In other words, if our mixture contains 99% of air at 95°F dry bulb, 75°F wet bulb, and 1% air at 80°F dry bulb, 67°F wet bulb, the mixture conditions will fall on the line near 95°F dry bulb and 75°F wet bulb and 99% of the distance between the two points above the lower condition. If the mixture contains 50% of each of the two conditions, the resultant mixture condition will fall on the line at a point of 50% or 1/2 of the distance between the two.

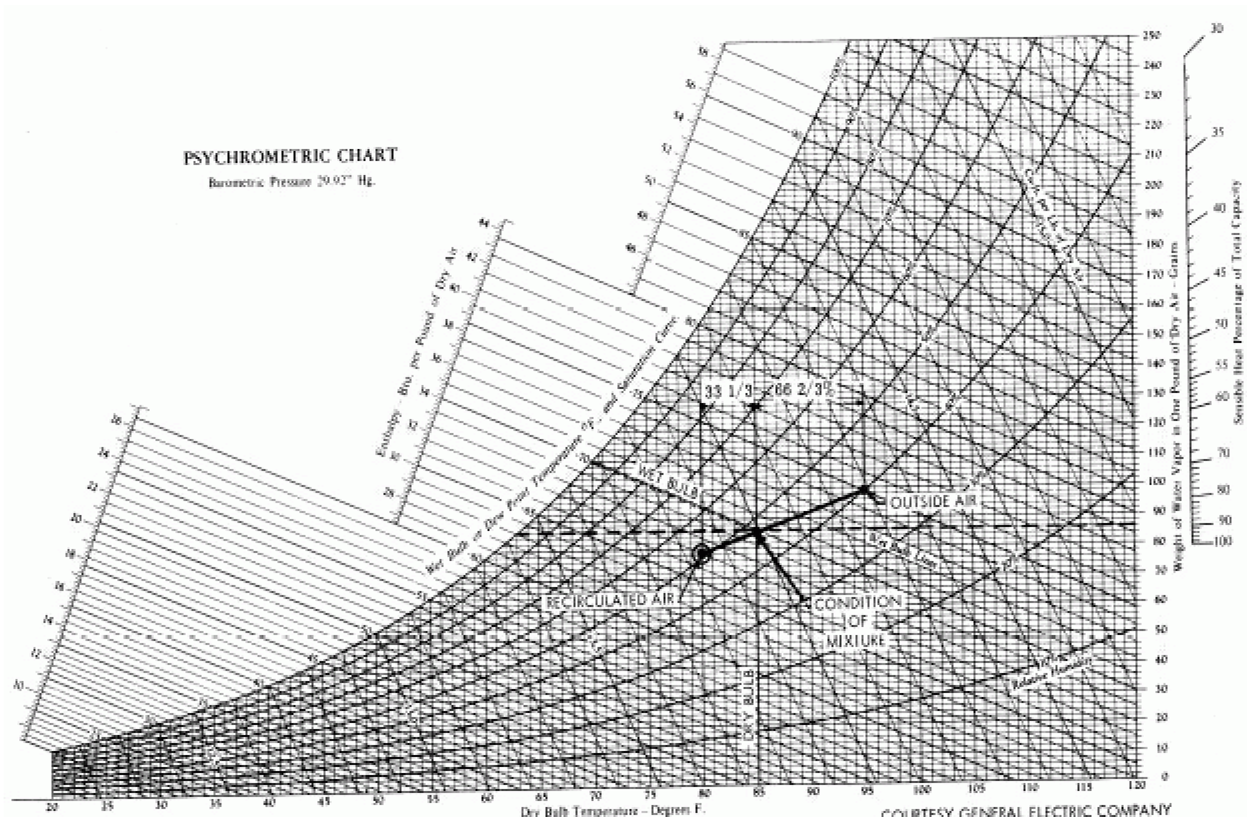

Figure 54F23

This chart shows the result of mixing air at two different conditions. In this case, the final condition is a mixture of 33-1/3% air at 95°F dry bulb and 75°F wet bulb, and 66-2/3% air at 80°F dry bulb and 67°F wet bulb. The result is air at 85°F dry bulb and 69.5°F wet bulb. The dry bulb temperature was used to calculate the percentages.

If, for example, 130 cfm of outside air at 95°F dry bulb, 75°F wet bulb, and 260 cfm of re-circulated air at 80°F dry bulb, 67°F wet bulb are mixed prior to cooling, the resultant condition prior to entering the cooling apparatus will be on the line connecting the two condition points on the psychrometric chart and 130/390, or 33-1/3% of the total distance between the two above the lower condition.

Since this distance between the two is also the difference in dry bulb temperatures, the final dry bulb temperature will be 33-1/3% of (95-80), plus the lower dry bulb temperature, or 5°F plus 80°F, giving 85°F as the mixture dry bulb temperature. Since it is easier to add than subtract, we always use the percentage of the higher temperature air quantity to be mixed, multiplied by the total dry bulb temperature difference, and add this to the lower temperature value.

To find the resultant wet bulb temperature of the mixture, we merely find where the mixture condition line crosses the 85°F constant dry bulb line. This point is the condition of our mixture, and we can then follow the other constant lines to their respective scales to find the wet bulb temperature, dew point, etc.

In this chapter, the basic fundamentals of psychrometry as well as several examples of its use to the Service Engineer have been covered. There are many more uses of the psychrometric chart other than the examples cited. If the Service Engineer will take the time and trouble to learn to use this chart, many of the complexities of air conditioning will be greatly simplified.

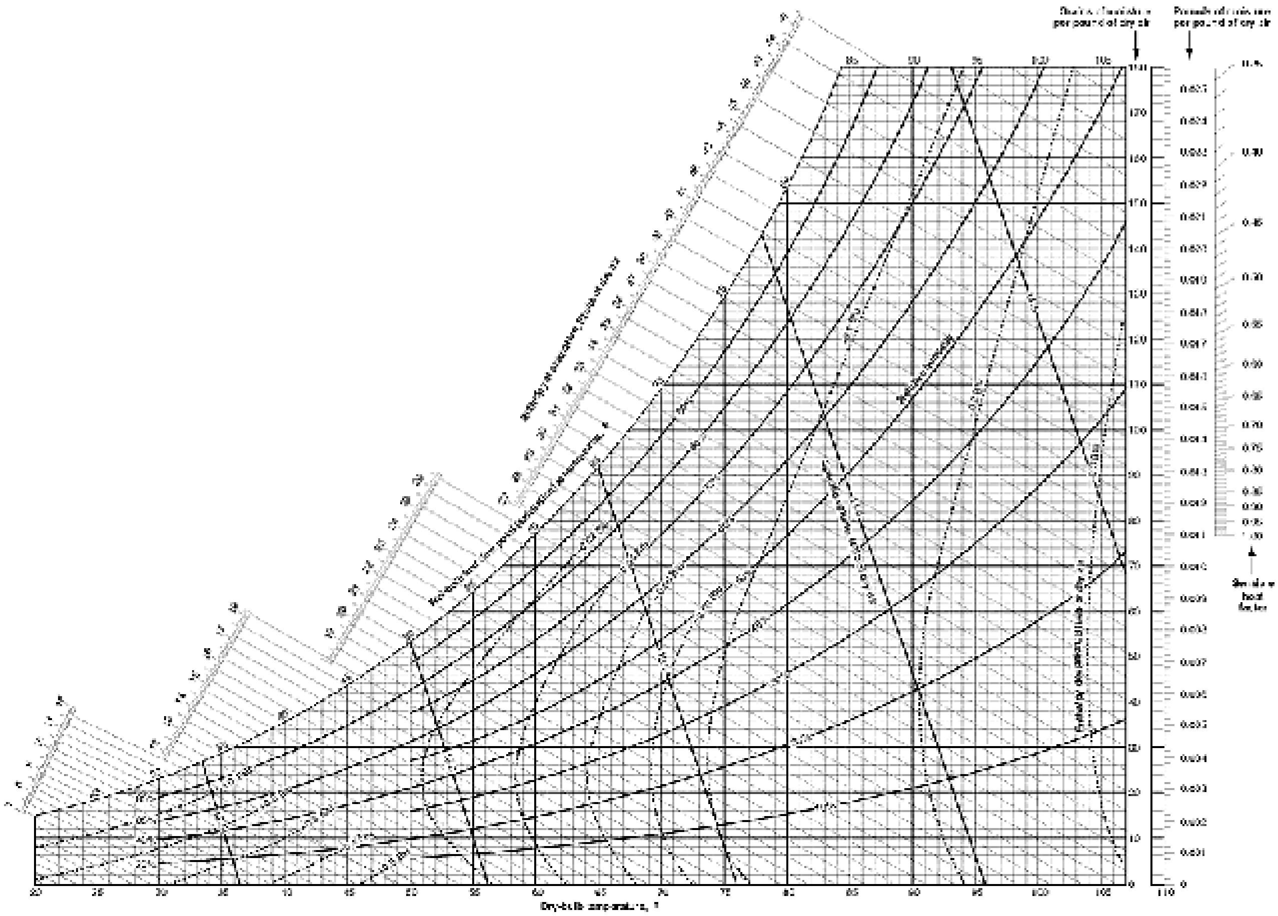

Figure 54F25