Table 3 . . . . Properties of Trichloromonofluoromethane (CCl_3F)[a]

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Enthalpy and Entropy Taken from −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	97.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

[a] ASHRAE designation—Refrigerant 11.

of liquid leaving the condenser from the enthalpy of superheated vapor going into it, thus,

$$Q_c = 26.3\ (89.34 - 29.68) = 1569 \text{ Btu per minute.}$$

(*e*) The cooling water rate (based on a gallon as 8.34 lb) is $1569 \div (8 \times 8.34) = 23.5$ gpm.

(*f*) The compressor size is fixed by the volume of gas which must be drawn into the machine per unit time. Saturated vapor at 52.7 psia has a specific volume, from Table 1, of 0.779 cu ft per pound, hence $26.3 \times 0.779 = 20.49$ cfm of gas must be handled. Assuming a volumetric efficiency of 90 percent, the compressor must then displace $20.49 \div 0.9 = 22.8$ cfm. The speed is given as 500 rpm and, as the unit is known to be double-acting, the displacement is therefore $(22.8 \times 1728) \div (2 \times 500) = 39.4$ cu in. If the unit were designed so that bore d and stroke were the same,

$$(\pi d^2) \div 4 = 39.4 \qquad d = 3.69 \text{ in.}$$

(*g*)
$$(CP) = (h_{vs} - h_{fc}) + (h_d - h_{vs})$$
$$= (82.82 - 29.68) + (89.34 - 82.82) = 8.17$$

where h_{fc} is the specific enthalpy of liquid at discharge from the condenser.

The coefficient of performance of *Example 1* may be compared with that of an ideal system operating on the Carnot cycle between the same temperature limits. Then $T_s = 501$ F (which is 41 F + 460) and $T_c = 554$ F (which is 94 F + 460) and,

$$(CP) = \frac{501}{554 - 501} = 9.6$$

The actual cycle is therefore $8.17 \div 9.6$ or 85 percent as effective as a Carnot cycle between the same temperature limits.

Influence of Suction Pressure

Brief consideration of the analytical procedure used in discussion of the simple saturation cycle will bring out the need for maintaining the suction pressure on any refrigeration system as high as the load will permit. As the suction pressure increases, for fixed discharge pressure, the enthalpy of refrigerant entering the evaporator remains unchanged, but the leaving enthalpy increases and, hence, the refrigerating effect increases. Further, compressor energy input is reduced not merely because of the greater enthalpy of the gas at suction, but also because of a reduction in the enthalpy of the superheated gas at discharge. Since the refrigerating effect is greater and the work less, it is obvious that there will be a substantial gain in the coefficient of the performance. See Chapter 54, Fig. 1.

The actual value of suction pressure on any system is obviously determined by the required temperature which must be maintained in the conditioned space. For a direct-expansion system the evaporator can be held at a temperature not much less than that of the conditioned enclosure, except in cases where lower temperatures may be needed in order to establish a desired ratio of dehumidifying to cooling load. When dehumidification requirements dictate the use of unusually low evaporator temperatures, the increased operating cost should properly be charged against the dehumidification rather than the sensible cooling.

Table 4 . . . . Properties of Trichlorotrifluoroethane ($C_2Cl_3F_3$)[a]

Temp	Pressure		Volume		Density		Enthalpy from −40°F			Entropy from −40°F	
F	psia	psig	Liquid cu ft/lb	Vapor cu ft/lb	Liquid lb/cu ft	Vapor lb/cu ft	Liquid Btu/lb	Latent Btu/lb	Vapor Btu/lb	Liquid Btu/lb/F deg	Vapor Btu/lb/F deg
0	0.8377	28.21*	0.00966	31.31	103.56	0.03194	7.98	70.92	78.89	0.0182	0.1725
4	.9503	27.99*	.00968	27.84	103.27	.03592	8.78	70.68	79.46	.0199	.1724
8	1.075	27.73*	.00971	24.81	102.98	.04031	9.59	70.44	80.03	.0216	.1723
12	1.213	27.45*	.00974	22.17	102.69	.04511	10.41	70.20	80.61	.0234	.1722
16	1.366	27.14*	.00977	19.84	102.40	.05040	11.22	69.96	81.18	.0251	.1722
20	1.534	26.80*	0.00979	17.81	102.10	0.05616	12.03	69.72	81.75	0.0268	0.1722
24	1.719	26.42*	.00982	16.02	101.81	.06243	12.85	69.48	82.33	.0285	.1721
28	1.922	26.01*	.00985	14.43	101.51	.06929	13.67	69.24	82.91	.0302	.1722
32	2.145	25.55*	.00988	13.03	101.21	.07675	14.49	69.00	83.49	.0318	.1722
36	2.388	25.06*	.00991	11.79	100.91	.08483	15.32	68.75	84.07	.0335	.1722
40	2.655	24.52*	0.00994	10.68	100.60	0.09361	16.16	68.50	84.65	0.0352	0.1723
44	2.944	23.93*	.00997	9.703	100.30	.1031	16.99	68.25	85.24	.0368	.1723
48	3.258	23.29*	.01000	8.830	99.99	.1133	17.82	68.00	85.82	.0385	.1724
52	3.602	22.59*	.01003	8.044	99.68	.1243	18.66	67.74	86.40	.0401	.1726
56	3.973	21.83*	.01006	7.342	99.37	.1362	19.50	67.48	86.98	.0418	.1727
60	4.374	21.02*	0.01010	6.713	99.05	0.1490	20.35	67.22	87.57	0.0434	0.1728
64	4.807	20.14*	.01013	6.149	98.73	.1626	21.19	66.96	88.15	.0450	.1729
68	5.275	19.18*	.01016	5.640	98.42	.1773	22.05	66.69	88.74	.0467	.1731
72	5.780	18.16*	.01019	5.180	98.10	.1931	22.90	66.43	89.33	.0483	.1732
76	6.320	17.06*	.01023	4.769	97.77	.2097	23.76	66.16	89.92	.0499	.1734
80	6.902	15.87*	0.01026	4.392	97.45	0.2277	24.63	65.88	90.51	0.0515	0.1736
84	7.527	14.60*	.01030	4.051	97.12	.2468	25.49	65.60	91.09	.0531	.1738
88	8.194	13.24*	.01033	3.742	96.79	.2672	26.36	65.32	91.68	.0547	.1740
92	8.908	11.79*	.01037	3.463	96.46	.2888	27.24	65.04	92.28	.0563	.1742
96	9.668	10.24*	.01040	3.208	96.13	.3117	28.11	64.75	92.86	.0578	.1744
100	10.48	8.59*	0.01044	2.976	95.79	0.3360	28.99	64.46	93.45	0.0594	0.1746
104	11.35	6.82*	.01048	2.762	95.46	.3620	29.89	64.16	94.05	.0610	.1748
108	12.28	4.93*	.01051	2.567	95.12	.3896	30.78	63.86	94.64	.0626	.1751
110	12.76	3.95*	0.01053	2.477	94.95	0.4038	31.22	63.71	94.93	0.0634	0.1752

* Inches of mercury below one atmosphere.
† Standard cycle temperatures.
[a] ASHRAE designation—Refrigerant 113.

Influence of Discharge Pressure

In contrast to suction pressure, the compressor discharge pressure should be kept as low as operating conditions will allow. This pressure must be high enough to provide a saturation temperature of refrigerant within the condenser that is greater than the exit temperature of the cooling water. The discharge pressure therefore is a direct function of the temperature of the cooling fluid, and will automatically rise whenever the temperature of cooling water (or air) rises; it will also rise when the flow rate of the cooling medium is decreased.

Increase in discharge pressure (for fixed suction pressure) raises the enthalpy of the gas leaving the compressor; hence, increases the work of compression. Further, as the enthalpy of saturated liquid leaving the condenser increases with pressure, the refrigerating effect must decrease. Thus the effect of such a pressure rise is to require more work per pound of refrigerant handled, and at the same time to necessitate an increase in the refrigerant flow rate. (See Chapter 54, Fig. 1.)

Influence of Water Jacket

The preceding discussion has, in every case, assumed isentropic compression. Where exact performance data are not available, this assumption is a desirable one since it leads to a conservatively large determination of the power required. In most actual systems, the compression process departs from isentropic due to irreversible heat transfers which occur between the vapor in the cylinder and the cylinder wall, and also because of intentional heat dissipation from the outside of the cylinder walls to the surroundings, or to a cooling fluid passing through a water jacket around the