Section 19

Refrigeration, Cryogenics, Optics, and Miscellaneous

BY

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19.1 MECHANICAL REFRIGERATION

by A. J. Rydzewski and Warren W. Rice

REFERENCES: Publications of the ASRE and ASHRAE; ''Air-conditioning Refrigerating Data Books''; *Refrigerating Engineering;* ARI Equipment Standards; ASHRAE Guide and Data Book. Merkel and Bosnjakovic, ''Diagrammen und Tabellen zur Berechnung der Absorptionskältemaschinen,'' Springer, Berlin. Jordan and Priester, ''Refrigeration and Air Conditioning,'' Prentice-Hall.

REFRIGERATION MACHINES AND PROCESSES

(For general theory of refrigeration, see Sec. 4; for air-conditioning uses, see Sec. 12.4; for cryogenics, see Sec. 19.2.)

Refrigeration is a term used to describe thermal systems which maintain a process space or material at a temperature less than available from ambient conditions. Heat is transferred from the materials to be cooled into a lower-temperature substance referred to as the **refrigerant**. The refrigerant is cooled by physical processes which pump heat into the ambient environment. These apply several physical phenomena such as phase changes, sensible heating, thermoelectric effects, work extraction by mechanical devices, and endothermic chemical reactions. Stored energy in the form of ice, sublimating solid carbon dioxide, and cold liquids are included with processes described by the word **refrigeration**.

Many refrigeration systems use a continuous cycle. This requires that the refrigerant be recovered for recycling after it has absorbed heat from the product or space to be cooled. The most commonly used system has a recirculating fluid that boils at a low temperature to remove heat. The vapor is compressed to a pressure such that the condensing temperature is elevated. The vapor is cooled and condensed by a heat exchanger which rejects the heat into the environment. The condensed liquid is passed through a restricting valve or orifice. The liquid changes into a mixture of liquid and vapor that is colder than the refrigerated space. This expansion at constant enthalpy is known as the *Joule-Thompson effect.* An expansive turbine can remove heat from a high-pressure vapor by extracting energy in the form of mechanical work. When the exhaust conditions are proper, the material will be a mixture of gas and condensate. This fluid can be used as a refrigerant or stored in liquid form as product. The remaining vapor can be recycled as needed to continue the process.

The equipment used in refrigeration processes is referred to as follows: high-pressure vapor heat exchangers are called *condensers,* low-pressure heat exchangers are called *evaporators,* throttling devices are called *expansion valves* or *capillaries,* and pressure-raising devices are called *compressors.*

With continuous recirculation of the cycle fluid, the low-temperature heat pickup must be pumped, via energy input, to a higher level so as to discharge it to atmospheric temperature. In this sense the system acts as an **energy-transport** device or **heat pump** or, from another viewpoint, as a temperature transformer. This philosophy of energy transport may equally well be applied to more remote systems, such as thermoelectric. It is, of course, a manifestation of the principles of the Second Law, and the effectiveness of the system's operation may be evaluated numerically.

Efficiency ratings of refrigeration equipment are expressed several ways. Some of the more common terms include **coefficient of performance** (COP) and **energy efficiency ratio** (EER). **COP** is defined as refrigerant effect divided by net work input, where the refrigerant effect is the absolute value of the heat transferred from the lower temperature source, and the net work input is the absolute value of heat transferred to the higher temperature sink minus the refrigerant effect. COP can alternatively be defined as the ratio calculated by dividing the total heating capacity in Btu per hour provided by the refrigeration system, including circulating fan heat, but excluding supplementary resistance, by the total electric input in watts × 3.412. This definition applies primarily to heat pumps. **EER** is a ratio calculated by dividing the cooling capacity in Btu per hour by the power input in watts at any set of rating conditions, expressed in Btu/W · h.

Other rating terms used primarily for residential and commercial air-conditioning units and heat pumps include **seasonal energy efficiency ratio** (SEER) and **heating season performance factor** (HSPF). Both adjust efficiency ratings for the variations encountered over a normal heating or cooling season.

The various association and society standards have specific conditions at which equipment is rated. In efficiency comparisons, familiarity with the specific rating method is important. The factors used in the calculation of manufacturer's published data may not be representative of the specific application. Part load conditions must be considered in any evaluation when the unit will not normally be fully loaded. If the load is constant and is known, comparisons should be made at that point. Other important factors include whether the stated efficiency is just the compressor efficiency, or if it includes motor, seal, bearing, and drive losses. It is also recommended that rated operating conditions be adjusted to actual conditions when known. For example, the actual condensing temperatures may be significantly higher or lower than rated conditions depending on the geographic region.

Refrigeration capacity is defined in terms of the ''ton.''

Units of Refrigeration In the United States, a **standard ton of refrigeration** corresponds to a heat absorption at a rate of 288,000 Btu/day or 200 Btu/min (3.5168 kW). The heat absorption per day is approximately the heat of fusion of 1 ton (907.19 kg) of ice at 32°F (0°C). The **standard rating** of a refrigerating machine, using a condensable vapor, is the number of standard tons of refrigeration it can produce under the following conditions: (1) liquid only enters the expansion valve and vapor only enters the compressor or the absorber of an absorption system; (2) the liquid entering the expansion valve is subcooled 9°F (5°C) and the vapor entering the compressor or absorber is superheated 9°F (5°C), these temperatures to be measured within 10 ft (3.05 m) of the compressor cylinder or absorber; (3) the pressure at the compressor or absorber inlet corresponds to a saturation temperature of 5°F (− 15°C); and (4) the pressure at the compressor or absorber outlet corresponds to a saturation temperature of 86°F (30°C).

The **British unit of refrigeration** corresponds to a heat-absorption rate of 237.6 Btu/min (4.175 kW) with inlet and outlet pressures corresponding to saturation temperatures of 23°F (− 5°C) and 59°F (15°C), respectively. The standard practice in Europe is to use kilowatts to specify refrigeration loads. Occasionally a unit of refrigeration capacity called the **frigorie** is used. A frigorie is approximately equivalent to 50 Btu/min (0.8786 kW), or one-quarter of a standard ton of refrigeration.

PROPERTIES OF REFRIGERANTS

(For detailed properties, see Sec. 4.)

Refrigerants are the transport fluids which convey the heat energy from the low-temperature level to the high-temperature level, where it can, in terms of heat transfer, give up its heat. In the broad sense, gases involved in liquefaction processes or in gas-compression cycles go through low-temperature phases and hence may be termed ''refrigerants,'' in a way similar to the more conventional vapor-compression fluids.

Refrigerants are **designated by number**. The identifying number of the refrigerant, or the word *Refrigerant* or both, may be used in conjunction with the trade name. For example, Refrigerant 134a can also be referred to as R-134a, HFC-134a, or SUVA-134a. The number designation

Table 19.1.1 Environmental Properties of Refrigerants

Refrigerant and number	Ozone depletion potential	Global warming potential	Toxicity, ppm (v/v)	AEL,* TLV†	Flash point, lower/upper, vol. %
R-11/(CFC-11)	1.0	1.0	1000	TLV	−
R-12/(CFC-12)	1.0	2.8	1000	AEL	−
R-502	0.23	3.75	1000	TLV	−
R-114	0.7	3.9	1000	TLV	−
R-402B (SUVA HP81)	0.03	0.52	1000	AEL	None
HCFC-123	0.02	0.02	30	AEL	None
HFC-134a	0.0	0.26	1000	AEL	None
HFC-125	0.0	0.84	1000	AEL	None
HCFC-124	0.02	0.1	500	AEL	None
R-22/(HCFC-22)	0.05	0.3	1000	TLV	None
HFC-23	0.0	6.0	1000	AEL	None
HFC-32	0.0	0.11	1000	TLV	14/31

* AEL is the recommended time-weighted average concentration of an airborne chemical to which nearly all workers may be exposed during an 8-h and/or 12-h day, 40-h week without adverse effect. Determined by the du Pont Company for compounds that do not have a TLV.

† TLV, established for industrial chemicals by the American Conference of Governmental Industrial Hygienists, is the recommended time-weighted average concentration of an airborne chemical to which nearly all workers may be exposed during an 8-h day, 40-h week without adverse effect.

SOURCE: Data from E. I. du Pont de Nemours & Co., Inc. Bulletin AG-1.

and safety classification of refrigerants is covered in ANSI/ASHRAE Standard 34-1992.

Recent evidence indicates that much of the damage to the atmospheric ozone layer is the result of decomposition of chlorofluorocarbon (CFC) chemicals. An international agreement known as the Montreal Protocol took effect in 1989 and a new Clean Air Act was signed into law in 1990 to limit the production and regulate the use and disposal of chlorofluorocarbons. The chlorofluorocarbons that cause ozone-layer depletion have been designated as CFC-type materials. Other refrigerants that are chlorofluorocarbons but cause little or no ozone destruction are designated as hydrochlorofluorocarbon (HCFC) or hydrofluorocarbon (HFC) refrigerants.

Several terms are used to specify the relative destructiveness of chlorofluorocarbon refrigerants to the ozone layer, and other effects on the environment. The **ozone depletion potential** (ODP) is the ozone-destroying power of a substance measured relative to Refrigerant 11 (CFC-11) (R-11). The **global warming potential** (GWP) is a relative measure of the ability of a substance to cause an increase in the temperature of the atmosphere by absorbing solar and earth radiation that is relative to the effect of Refrigerant 11. A listing of the ozone depletion potential, global warming potential, and 8-hour toxicity limits (AEL or TLV) for halo-carbon refrigerants is given in Table 19.1.1. Table 19.1.2 is a listing of replacement refrigerants for existing CFC compounds.

The **normal boiling point** is an important attribute of refrigerants for vapor compression systems. The desirable characteristic is a positive gage pressure throughout the refrigerating system. This will eliminate contamination of the refrigerant and resulting heat-transfer inefficiencies due to air inleakage. Table 19.1.3 shows some common fluids in order of boiling point: included are halocarbon compounds that offer reduced ozone depletion potential, hydrocarbons, sulfur compounds, and inorganic compounds.

The desirable thermal properties of a refrigerant are (1) convenient evaporation and condensation pressures, (2) high critical and low freezing temperatures, (3) high latent heat of evaporation and high vapor specific heat, (4) low viscosity and high film heat conductivity. Desirable practical properties include (1) low cost, (2) chemical and physical inertness under operating conditions, (3) noncorrosiveness toward ordinary construction materials, and (4) low explosive hazard both alone and mixed with air. The refrigerant should be nonpoisonous and nonirritating and should not cause deterioration in the lubricant used. Leakage should be detectable by simple tests, easily performed.

The **specific volume of refrigerant** to be handled determines the size of positive displacement compressors, but with centrifugal compressors a large volume is not objectionable and may be an advantage for small units.

A comparison of various refrigerants based on *ideal performance* is given in Table 19.1.4. The results are for the **standard temperature range** of 5 to 86°F (− 15 to 30°C) and also for certain other ranges. For these calculations, zero piston clearance and expansion through a throttle valve were assumed. The use of an expansion cylinder instead of a valve would have yielded work (available for supplying some of the work of compression), but this is always a negligible quantity, except when the compression pressure approaches the critical pressure, as for carbon dioxide. Theoretical *coefficients of performance* may be obtained by dividing 4.72 by the theoretical horsepower given in the table.

Ammonia (R-717) is used extensively in commercial, industrial, and moderately low-temperature installations. It is toxic, and recent laws regulate the operation, maintenance, and design of ammonia systems that contain 10,000 lb of refrigerant or more. (See Document 29 CFR 1910.119 of the Federal Register for further information.) Ammonia is provided commercially in several purity grades. It is recommended that refrigerant-grade anhydrous ammonia of 99.95 percent purity be used. Moisture contamination from impure ammonia will accumulate in evaporator coils and inhibit performance. Ammonia is corrosive to copper and copper-bearing alloys. It is not miscible to any large extent with lubricating oils. Exposure to it in a confined space at concentrations below 400 ppm is rarely fatal. Ammonia is immediately fatal at 0.5 to 0.6 percent by volume. The presence of ammonia can be detected by burning a sulfur candle in the vicinity of a leak. A white cloud will form if ammonia vapors are present. Hydrochloric acid solutions and wetted

Table 19.1.2 Replacement Refrigerant Compounds

Current refrigerant	Replacement refrigerant	Formula	End uses*
R-11/CFC-11	HCFC-123	$CHCL_2CF_3$	Chillers [+ 25°F (− 3.9°C)]
R-12/CFC-12	HFC-134a	CH_2FCF_3	Medium temperature [− 5°F (− 20.6°C)]
R-13/CFC-13	HFC-23	CHF_3	Low temperature [− 80°F (− 62.2°C)]
R-502	R-402A, R404A	Blend	Ice machines, freezers [− 50°F (− 45.6°C)]
R-22	HCFC-22	$CHClF_2$	General use [− 50°F (− 45.6°C)]

* Lower optimum use temperature.

Table 19.1.3 Selected Refrigerants and Gases in Low-Temperature Applications

Refrigerant	Refrigerant no.	Symbol	Molecular weight	Boiling point, °F	Critical point Temperature, °F	 Pressure, lb/in^2 abs
Helium	704	He	4.0	−452	−450.0	33
Hydrogen	702	H_2	2.0	−423	−400.0	188
Nitrogen	728	N_2	28.0	−320	−232.0	492
Air	729	—	29.0	−312	−220.3	547
Oxygen	732	O_2	32.0	−297	−182.0	730
Methane	50	CH_4	16.0	−258.9	−115.8	673
Carbon tetrafluoride	14	CF_4	88.0	−198.4	−49.9	542
Ethylene	1,150	C_2H_4	28.0	−155.0	48.8	732
Ethane	170	C_2H_6	30.0	−127.5	90.1	708
Trifluoromethane	23	CHF_3	70.01	−115.66	78.6	701.4
Carbon dioxide	744	CO_2	44.0	−109.3	87.8	1,071
Propane	290	C_3H_8	44.1	−44.2	202	661
Monodichlorodifluoromethane	22	$CHClF_2$	86.5	−41.36	204.8	716
Ammonia	717	NH_3	17.0	−28.0	271.4	1,657
1,1,1,2-tetrafluoroethane	134a	CH_2FCF_3	102	−15	213.9	588.3
Methylchloride	40	CH_3Cl	50.5	−10.8	289.4	969
Isobutane	601	C_4H_{10}	58.1	10.3	272.7	537
2-chloro-1,1,1,2-tetrafluoroethane	124	$CHClFCF_3$	136.5	12.2	252	524.5
Sulfur dioxide	764	SO_2	64.1	14.0	314.8	1,142
Butane	600	C_4H_{10}	58.1	31.3	306	550
Dichloromonofluoromethane	21	$CHCl_2F$	102.9	48.1	353.3	750
2,2-dichloro-1,1,1-trifluoroethane	123	$CHCl_2CF_3$	152.9	82	362.8	533.2
Ethyl-ether	610	$C_4H_{10}O$	74.1	94.3	522.1	381
Methylene chloride	30	CH_2Cl_2	84.9	103.6	480	670
Water	718	H_2O	18.0	212	706.1	3,226

Table 19.1.4 Ideal Performance of Refrigerants for Various Temperature Ranges

Refrigerant and (number)	Operating temperature range, °F	Suction pressure, lb/in^2 abs	Head pressure, lb/in^2 abs	Ratio of head to suction pressure	With dry and saturated suction vapor, per ton Weight of vapor, lb/min	 Piston displacement, ft^3/min	 Theoretical hp	Temperature at end of compression, °F	Type of compressor
Air (729)	5–86	14.7	73.5	5.0	7.02	82.3	2.82	277.0	Recip. and exp. cyl
Water (718)	32–86	0.0885	0.6152	6.95	0.1957	647.0	0.618	332	Centrif. or ejector
	32–100	0.0885	0.9492	10.73	0.1985	656.2	0.819	420	
	40–100	0.1217	0.9492	7.80	0.1978	483.4	0.687	366	
Carbon dioxide (CO_2) (744)	5–86	332.0	1,043.0	3.14	3.61	0.960	1.827	160.3	Recip.
Ammonia (NH_3) (717)	5–86	34.27	169.2	4.94	0.421	3.44	0.99	209.8	Recip.
	20–100	48.21	211.9	4.40	0.421	2.49	0.94	212.8	
	40–100	73.32	211.9	2.89	0.427	1.70	0.65	176.0	
SUVA 123	5–86	2.303	15.9	6.9	3.26	45.6	0.95*	90.5	Recip.
($CHCl_2CF_3$)	20–100	3.48	20.8	5.97	3.33	31.7	0.91	101.	Centrif.
(HCFC-123)	40–100	5.79	20.8	3.59	3.18	18.8	0.6*	100.	Centrif.
SUVA 134a	5–86	23.77	111.8	4.7	3.14	6.1	1.04	98.	Centrif.
(CH_2FCF_3)	20–100	33.13	139.0	4.19	3.28	4.62	1.0*	109.	
(HFC-134a)	40–100	49.77	139.0	2.79	3.14	3.0	0.68	106.	
SUVA HP 81	5–86	54.37	200.5	3.68	3.56	3.37	1.07	99.	Screw or rotary or recip.
blend [R-402	20–100	72.12	243.9	3.38	3.78	2.71	1.04	121.5	
(38/2/60)]	40–100	102.26	243.9	2.385	3.62	1.83	0.68*	114.	
R 22	5–86	43.02	174.5	4.06	2.926	3.65	1.03	131.7	Recip.
($CHClF_2$)	20–100	57.98	212.6	3.67	3.023	2.83	0.99	152.7	
(22)	40–100	83.72	212.6	2.54	2.936	1.93	0.68	131.0	
Methylene	5–86	1.28	10.07	8.56	1.485	74.0	0.96*	205.1*	Centrif.
chloride	20–100	1.92	13.25	6.90	1.520	47.72	0.91*	157.1*	
(CH_2Cl_2) (30)	40–100	3.38	13.25	3.92	1.493	27.76	0.63*	167.7*	
Methyl chloride	5–86	21.15	94.70	4.48	1.331	5.95	0.96	178.1	Recip.
(CH_3Cl)	20–100	29.16	116.7	4.00	1.363	4.51	0.90	184.4	
(40)	40–100	43.25	116.7	2.69	1.342	3.07	0.62	157.0	
Sulfur dioxide	5–86	11.81	66.45	5.63	1.415	9.08	0.97	191.4	Recip.
(SO_2) (764)	20–100	17.18	84.52	4.92	1.453	6.52	0.92	193.4	
	40–100	27.10	84.52	3.12	1.444	4.17	0.63	162.6	
Propane	5–86	42.1	155.3	3.69	1.653	4.10	1.35*	92.9*	Recip.
(C_3H_8) (290)	20–100	55.5	187.0	3.37	1.730	3.29	1.32*	103.9*	
	40–100	78.0	187.0	2.40	1.646	2.26	0.90*	101.5*	
Ethane (C_2H_5) (170)	5–86	236.0	675.9	2.87	3.41	1.82	2.180	105	Recip.
Ethyl chloride	5–86	4.65	27.10	5.83	1.405	24.0	0.95*	106.3*	Rotary
(C_2H_5Cl)	20–100	6.80	34.79	5.12	1.425	17.19	0.92*	116.1*	
(160)	40–100	10.79	34.79	3.22	1.375	10.73	0.63*	109.9*	

* Values may be slightly in error.

Table 19.1.5 Properties of Superheated Ammonia

(v = specific volume in ft^3/lb; h = enthalpy in Btu/lb; s = entropy; h_f and s_f are measured from −40°F.)

Pressure, lb/in² abs	Temperature of saturated vapor, °F		Temperature of superheated vapor, °F								
			−30	−20	−10	0	10	20	30	40	50
		v	26.58	27.26	27.92	28.58	29.24	29.90	30.55	31.20	31.85
10	−41.34	h	603.2	608.5	613.7	618.9	624.0	629.1	634.2	639.3	644.4
		s	1.4420	1.4542	1.4659	1.4773	1.4884	1.4992	1.5097	1.5200	1.5301
		v			13.74	14.09	14.44	14.78	15.11	15.45	15.78
20	−16.64	h			610.0	615.5	621.0	626.4	631.7	637.0	642.3
		s			1.3784	1.3907	1.4025	1.4138	1.4248	1.4356	1.4460
		v				9.250	9.492	9.731	9.966	10.20	10.43
30	−0.57	h				611.9	617.8	623.5	629.1	634.6	640.1
		s				1.3371	1.3497	1.3618	1.3733	1.3845	1.3953
		v						7.203	7.387	7.568	7.746
40	11.66	h						620.4	626.3	632.1	637.8
		s						1.3231	1.3353	1.3470	1.3583
		v							5.838	5.988	6.135
50	21.67	h							623.4	629.5	635.4
		s							1.3046	1.3169	1.3286
			100	120	140	160	180	200	240	280	320
		v	4.190	4.371	4.548	4.722	4.893	5.063	5.398	5.730	
80	44.40	h	658.7	670.4	681.8	693.2	704.4	715.6	738.1	760.7	
		s	1.3199	1.3404	1.3598	1.3784	1.3963	1.4136	1.4467	1.4781	
		v	3.304	3.454	3.600	3.743	3.883	4.021	4.294	4.562	
100	56.05	h	655.2	667.3	679.2	690.8	702.3	713.7	736.5	759.4	
		s	1.2891	1.3104	1.3305	1.3495	1.3678	1.3854	1.4190	1.4507	
		v	2.712	2.842	2.967	3.089	3.190	3.326	3.557	3.783	
120	66.02	h	651.6	664.2	676.5	688.5	700.2	711.8	734.9	758.0	
		s	1.2628	1.2850	1.3058	1.3254	1.3441	1.3620	1.3960	1.4281	
		v	2.288	2.404	2.515	2.622	2.727	2.830	3.030	3.227	3.420
140	74.79	h	647.8	661.1	673.7	686.0	698.0	709.9	733.3	756.7	780.0
		s	1.2396	1.2628	1.2843	1.3045	1.3236	1.3418	1.3763	1.4088	1.4395
		v	1.969	2.075	2.175	2.272	2.365	2.457	2.635	2.809	2.980
160	82.64	h	643.9	657.8	670.9	683.5	695.8	707.9	731.7	755.3	778.9
		s	1.2186	1.2429	1.2652	1.2859	1.3054	1.3240	1.3591	1.3919	1.4229
		v	1.720	1.818	1.910	1.999	2.084	2.167	2.328	2.484	2.637
180	89.78	h	639.9	654.4	668.0	681.0	693.6	705.9	730.1	753.9	777.7
		s	1.1992	1.2247	1.2477	1.2691	1.2891	1.3081	1.3436	1.3768	1.4081
		v	1.520	1.612	1.698	1.780	1.859	1.935	2.082	2.225	2.364
200	96.34	h	635.6	650.9	665.0	678.4	691.3	703.9	728.4	752.5	776.5
		s	1.1809	1.2077	1.2317	1.2537	1.2742	1.2935	1.3296	1.3631	1.3947
		v		1.443	1.525	1.601	1.675	1.745	1.881	2.012	2.140
220	102.42	h		647.3	662.0	675.8	689.1	701.9	726.8	751.1	775.3
		s		1.1917	1.2167	1.2394	1.2604	1.2801	1.3168	1.3507	1.3825
		v		1.302	1.380	1.452	1.521	1.587	1.714	1.835	1.954
240	108.09	h		643.5	658.8	673.1	686.7	699.8	725.1	749.8	774.1
		s		1.1764	1.2025	1.2259	1.2475	1.2677	1.3049	1.3392	1.3712
		v		1.182	1.257	1.326	1.391	1.453	1.572	1.686	1.796
260	113.42	h		639.5	655.6	670.4	684.4	697.7	723.4	748.4	772.9
		s		1.1617	1.1889	1.2132	1.2354	1.2560	1.2938	1.3258	1.3608

SOURCE: Condensed from NBS Circ. 142, 1923.

litmus papers will also react with ammonia to give visible evidence of the presence of vapors. See Tables 19.1.5 and 19.1.6 for thermodynamic properties of R-717.

Carbon dioxide (R-744) was used for a long time as a ''safety refrigerant.'' It is not toxic, and exposure to it is not dangerous unless concentrations are high. Like many other refrigerants, it is heavier than air, and caution must be exercised in confined spaces, as it may displace oxygen and result in asphyxiation. With cooling water at 70°F (21°C), the condensing pressure of CO_2 is high, and since its critical temperature is 87.8°F (31°C), condensation will not occur at water temperatures above this. Specific power consumption is high on CO_2 machines.

Sulfur dioxide (R-764) has rapidly gone out of use. It is extremely corrosive unless absolutely anhydrous. It is extremely irritating in small concentrations and toxic in high concentrations.

Methyl chloride (R-40) (CH_3Cl), an anesthetic in amounts of 5 to 10 percent by volume, has been used in air-cooled units of moderate or small sizes. It is miscible with mineral oils; small amounts of moisture in a methyl chloride system will cause trouble by freezing in expansion valves.

For industrial work, butane (C_4H_{10}), propane (C_3H_8), and, at low temperatures, ethane (C_2H_6) are used. **Dieline** (dichloroethylene, $C_2H_2Cl_2$) and **Carrene** (dichloromethane, CH_2Cl_2) have been used to some extent, usually in centrifugal compressors.

As a result of legislation requiring the discontinuation of production

Table 19.1.6 Properties of Saturated Ammonia
(h_f and s_f are measured from − 40°F.)

		Specific volume, ft³/lb		Enthalpy, Btu			Entropy		
Temp., °F, t	Pressure, lb/in² abs, p	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, h_f	Vapor-ization, h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Vapor-ization, s_{fg}	Sat. vapor, s_g
− 40	10.41	0.02322	24.86	0.0	597.6	597.6	0.0000	1.4242	1.4242
− 38	11.04	0.02326	23.53	2.1	596.2	598.3	0.0051	1.4142	1.4193
− 36	11.71	0.02331	22.27	4.3	594.8	599.1	0.0101	1.4043	1.4144
− 34	12.41	0.02335	21.10	6.4	593.5	599.9	0.0151	1.3945	1.4096
− 32	13.14	0.02340	20.00	8.5	592.1	600.6	0.0201	1.3847	1.4048
− 30	13.90	0.02345	18.97	10.7	590.7	601.4	0.0250	1.3751	1.4001
− 28	14.71	0.02349	18.00	12.8	589.3	602.1	0.0300	1.3655	1.3955
− 26	15.55	0.02354	17.09	14.9	587.9	602.8	0.0350	1.3559	1.3909
− 24	16.42	0.02359	16.24	17.1	586.5	603.6	0.0399	1.3464	1.3863
− 22	17.34	0.02364	15.43	19.2	585.1	604.3	0.0448	1.3370	1.3818
− 20	18.30	0.02369	14.68	21.4	583.6	605.0	0.0497	1.3277	1.3774
− 18	19.30	0.02374	13.97	23.5	582.2	605.7	0.0545	1.3184	1.3729
− 16	20.34	0.02378	13.29	25.6	580.8	606.4	0.0594	1.3092	1.3686
− 14	21.43	0.02383	12.66	27.8	579.3	607.1	0.0642	1.3001	1.3643
− 12	22.56	0.02388	12.06	30.0	577.8	607.8	0.0690	1.2910	1.3600
− 10	23.74	0.02393	11.50	32.1	576.4	608.5	0.0738	1.2820	1.3558
− 8	24.97	0.02399	10.97	34.3	574.9	609.2	0.0786	1.2730	1.3516
− 6	26.26	0.02404	10.47	36.4	573.4	609.8	0.0833	1.2641	1.3474
− 4	27.59	0.02409	9.991	38.6	571.9	610.5	0.0880	1.2553	1.3433
− 2	28.98	0.02414	9.541	40.7	570.4	611.1	0.0928	1.2465	1.3393
0	30.42	0.02419	9.116	42.9	568.9	611.8	0.0975	1.2377	1.3352
2	31.92	0.02424	8.714	45.1	567.3	612.4	0.1022	1.2290	1.3312
4	33.47	0.02430	8.333	47.2	565.8	613.0	0.1069	1.2204	1.3273
6	35.09	0.02435	7.971	49.4	564.2	613.6	0.1115	1.2119	1.3234
8	36.77	0.02440	7.629	51.6	562.7	614.3	0.1162	1.2033	1.3195
10	38.51	0.02446	7.304	53.8	561.1	614.9	0.1208	1.1949	1.3157
12	40.31	0.02451	6.996	56.0	559.5	615.5	0.1254	1.1864	1.3118
14	42.18	0.02457	6.703	58.2	557.9	616.1	0.1300	1.1781	1.3081
16	44.12	0.02462	6.425	60.3	556.3	616.6	0.1346	1.1697	1.3043
18	46.13	0.02468	6.161	62.5	554.7	617.2	0.1392	1.1614	1.3006
20	48.21	0.02474	5.910	64.7	553.1	617.8	0.1437	1.1532	1.2969
22	50.36	0.02479	5.671	66.9	551.4	618.3	0.1483	1.1450	1.2933
24	52.59	0.02485	5.443	69.1	549.8	618.9	0.1528	1.1369	1.2897
26	54.90	0.02491	5.227	71.3	548.1	619.4	0.1573	1.1288	1.2861
28	57.28	0.02497	5.021	73.5	546.4	619.9	0.1618	1.1207	1.2825
30	59.74	0.02503	4.825	75.7	544.8	620.5	0.1663	1.1127	1.2790
32	62.29	0.02508	4.637	77.9	543.1	621.0	0.1708	1.1047	1.2755
34	64.91	0.02514	4.459	80.1	541.4	621.5	0.1753	1.0968	1.2721
36	67.63	0.02521	4.289	82.3	539.7	622.0	0.1797	1.0889	1.2686
38	70.43	0.02527	4.126	84.6	537.9	622.5	0.1841	1.0811	1.2652
40	73.32	0.02533	3.971	86.8	536.2	623.0	0.1885	1.0733	1.2618
42	76.31	0.02539	3.823	89.0	534.4	623.4	0.1930	1.0655	1.2585
44	79.38	0.02545	3.682	91.2	532.7	623.9	0.1974	1.0578	1.2552
46	82.55	0.02551	3.547	93.5	530.9	624.4	0.2018	1.0501	1.2519
48	85.82	0.02558	3.418	95.7	529.1	624.8	0.2062	1.0424	1.2486
50	89.19	0.02564	3.294	97.9	527.3	625.2	0.2105	1.0348	1.2453
52	92.66	0.02571	3.176	100.2	525.5	625.7	0.2149	1.0272	1.2421
54	96.23	0.02577	3.063	102.4	523.7	626.1	0.2192	1.0197	1.2389
56	99.91	0.02584	2.954	104.7	521.8	626.5	0.2236	1.0121	1.2357
58	103.7	0.02590	2.851	106.9	520.0	626.9	0.2279	1.0046	1.2325
60	107.6	0.02597	2.751	109.2	518.1	627.3	0.2322	0.9972	1.2294
62	111.6	0.02604	2.656	111.5	516.2	627.7	0.2365	0.9897	1.2262
64	115.7	0.02611	2.565	113.7	514.3	628.0	0.2408	0.9823	1.2231
66	120.0	0.02618	2.477	116.0	512.4	628.4	0.2451	0.9750	1.2201
68	124.3	0.02625	2.393	118.3	510.5	628.8	0.2494	0.9676	1.2170
70	128.8	0.02632	2.312	120.5	508.6	629.1	0.2537	0.9603	1.2140
72	133.4	0.02639	2.235	122.8	506.6	629.4	0.2579	0.9531	1.2110
74	138.1	0.02646	2.161	125.1	504.7	629.8	0.2622	0.9458	1.2080
76	143.0	0.02653	2.089	127.4	502.7	630.1	0.2664	0.9386	1.2050
78	147.9	0.02661	2.021	129.7	500.7	630.4	0.2706	0.9314	1.2020

Table 19.1.6 Properties of Saturated Ammonia (*Continued*)

(h_f and s_f are measured from − 40°F.)

		Specific volume, ft³/lb		Enthalpy, Btu			Entropy		
Temp., °F, t	Pressure, lb/in² abs, p	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, h_f	Vapor-ization, h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Vapor-ization, s_{fg}	Sat. vapor, s_g
80	153.0	0.02668	1.955	132.0	498.7	630.7	0.2749	0.9242	1.1991
82	158.3	0.02675	1.892	134.3	496.7	631.0	0.2791	0.9171	1.1962
84	163.7	0.02684	1.831	136.6	494.7	631.3	0.2833	0.9100	1.1933
86	169.2	0.02691	1.772	138.9	492.6	631.5	0.2875	0.9029	1.1904
88	174.8	0.02699	1.716	141.2	490.6	631.8	0.2917	0.8958	1.1875
90	180.6	0.02707	1.661	143.5	488.5	632.0	0.2958	0.8888	1.1846
92	186.6	0.02715	1.609	145.8	486.4	632.2	0.3000	0.8818	1.1818
94	192.7	0.02723	1.559	148.2	484.3	632.5	0.3041	0.8748	1.1789
96	198.9	0.02731	1.510	150.5	482.1	632.6	0.3083	0.8678	1.1761
98	205.3	0.02739	1.464	152.9	480.0	632.9	0.3125	0.8608	1.1733
100	211.9	0.02747	1.419	155.2	477.8	633.0	0.3166	0.8539	1.1705
105	228.9	0.02769	1.313	161.1	472.3	633.4	0.3269	0.8366	1.1635
110	247.0	0.02790	1.217	167.0	466.7	633.7	0.3372	0.8194	1.1566
115	266.2	0.02813	1.128	173.0	460.9	633.9	0.3474	0.8023	1.1497
120	286.4	0.02836	1.047	179.0	455.0	634.0	0.3576	0.7851	1.1427

of CFC refrigerants, two new families of refrigerants, hydrochlorofluorocarbons (HCFC) and hydrofluorocarbons (HFC) have been commercialized to replace them. Current information indicates that these replacement compounds have very low levels of toxicity. They are odorless and nonirritating. With the exception of HFC-32, HFC-143a, and HFC-152a, they are nonflammable. The replacement compounds include HCFC-22 (R-22): chloro-difluoromethane ($CHClF_2$) (see Fig. 19.1.1); HFC-23: trifluoromethane (CHF_3); HFC-32: difluoromethane (CH_2F_2); HCFC-123: 2,2-dichloro-1,1,1-trifluoroethane ($CHCl_2CF_3$) (see Fig. 19.1.2); HCFC-124: 2-chloro-1,1,1,2-tetrafluoroethane ($CHClFCF_3$); HFC-125: pentafluoroethane (CHF_2CF_3); HFC-143a: trifluoroethane (CH_3CF_3); HFC-134a: 1,1,1,2-tetrafluoroethane (CH_2FCF_3) (see Fig. 19.1.3); HFC-152a: 1,1-difluoroethane (CH_3CHF_2); HCFC-124: 2-chloro-1,1,1,2-tetrafluoroethane ($CHClFCF_3$); and HCFC-123: 2,2-dichloro-1,1,1-trifluoroethane ($CHCl_2CF_3$). HCFC-22 has been in use for many years, and because of its low ODP, will continue to be used in air-conditioning, commercial, and industrial refrigeration systems. HFC-134a is a medium-pressure refrigerant used as the replacement for R12. HFC-123 is a low-pressure refrigerant used as the replacement for R-11.

Several of the HCFC class of compounds have been blended together to create refrigerants with properties which simulate properties of banned refrigerants. One such blend has been proposed as the replacement for R-502. This blend is designated as R-402a, also known by the brand name SUVA HP80 and others. The thermodynamic properties of this mixture are shown in Fig. 19.1.4. Use caution when using blended refrigerants in centrifugal compressors. The compressor impeller may act to separate the components of the blend by molecular weight.

The HFC and HCFC refrigerants may not be miscible with the petroleum-based lubricants used in CFC systems. Existing systems depend on oil and refrigerant miscibility to return oil from the evaporators and piping back into the compressor lubricating system. Synthetic oils such as polyol ester and polyalkylene glycol must be used with the new refrigerants. In addition, seals, bushings, gaskets, motor insulation (hermetic machines) may also need replacement.

Building codes often require monitoring of refrigerant levels in equipment spaces. In addition to a monitor, an alarm system and exhaust facilities should be installed to prevent exposure above the acceptable exposure limit (AEL). ANSI/ASHRAE 15-1994, ‘‘Safety Code for Mechanical Refrigeration,’’ has recommendations covering equipment room safety.

Leak Detection Detection and repair of refrigerant leaks is an essential part of responsible refrigeration equipment operation. Worker safety, potential environmental damage, and refrigerant cost are the primary reasons for a detection and repair program. Area monitors are useful in determining that a leak has occurred. Hand-held detectors are useful in determining specific leak location. Refrigerants containing chlorine as part of the molecule are readily detected in minute amounts by passing them through a copper gauze kept hot by the essentially colorless flame of burning methyl alcohol. Even traces of such refrigerants in air are readily detectable by the intense green color imparted to the flame by their presence. The alternative refrigerants have either no chlorine or a lower amount of chlorine. They require new types of equipment which do not rely on chlorine detection. Nonselective, halogen-specific, and compound-specific detectors are available. Each type has limitations, and selection must be based on the specific application. For example, the nonspecific-type detector is poorly suited for area monitoring because of its high detection limit and potential for false alarms, but is a reliable, low-cost choice for pinpointing leaks. The halogen-specific detectors can be applied for either leak pinpointing or area monitoring. They are best applied where only one refrigerant is in use. The compound-specific detectors are relatively expensive, but are very selective. They are best applied in environments where several refrigerants or other chemical compounds may be present.

The pressure-temperature relations for the saturated vapors of many of the commercially important refrigerants are given in Fig. 19.1.5. If the temperature at which the refrigeration is desired and the temperature at which heat can be discarded (condenser temperature) are known, the chart is convenient for determining for any chosen refrigerant the pressures which must be maintained. For instance, if the use of methyl chloride is contemplated in an air-cooled cabinet food freezer located in a room where the air temperature may rise to 90°F (32°C), the compressor discharge must be at least 100 psia (690 kPa absolute), and if the cabinet cooling coil is to be held at − 30°F (− 34°C), compressor suction will be 9 psia (62 kPa absolute).

The large volumes required for HCFC-123 and dieline can be handled satisfactorily by centrifugal compressors. When the evaporating pressure is below atmospheric—as in the case of HCFC-123, dieline, ammonia in food freezers, sulfur dioxide, water vapor, and butane—air leaks are likely to occur. The air tends to accumulate in the condenser and the heat-transfer performance is degraded.

Refrigerating systems which operate with a positive pressure in the equipment will leak refrigerant into the atmosphere. Many HFC- and HCFC-type refrigerants are practically odorless. Dangerous concentrations can build up in unventilated enclosed rooms such as freezers should a significant leak occur. Leak detection devices are available commercially to provide warnings of equipment leakage before harmful concentrations are present in occupied spaces. Addition of certain fluorescent compounds into the refrigerant in an operating system will allow identification of equipment leaks. The additives are placed into the

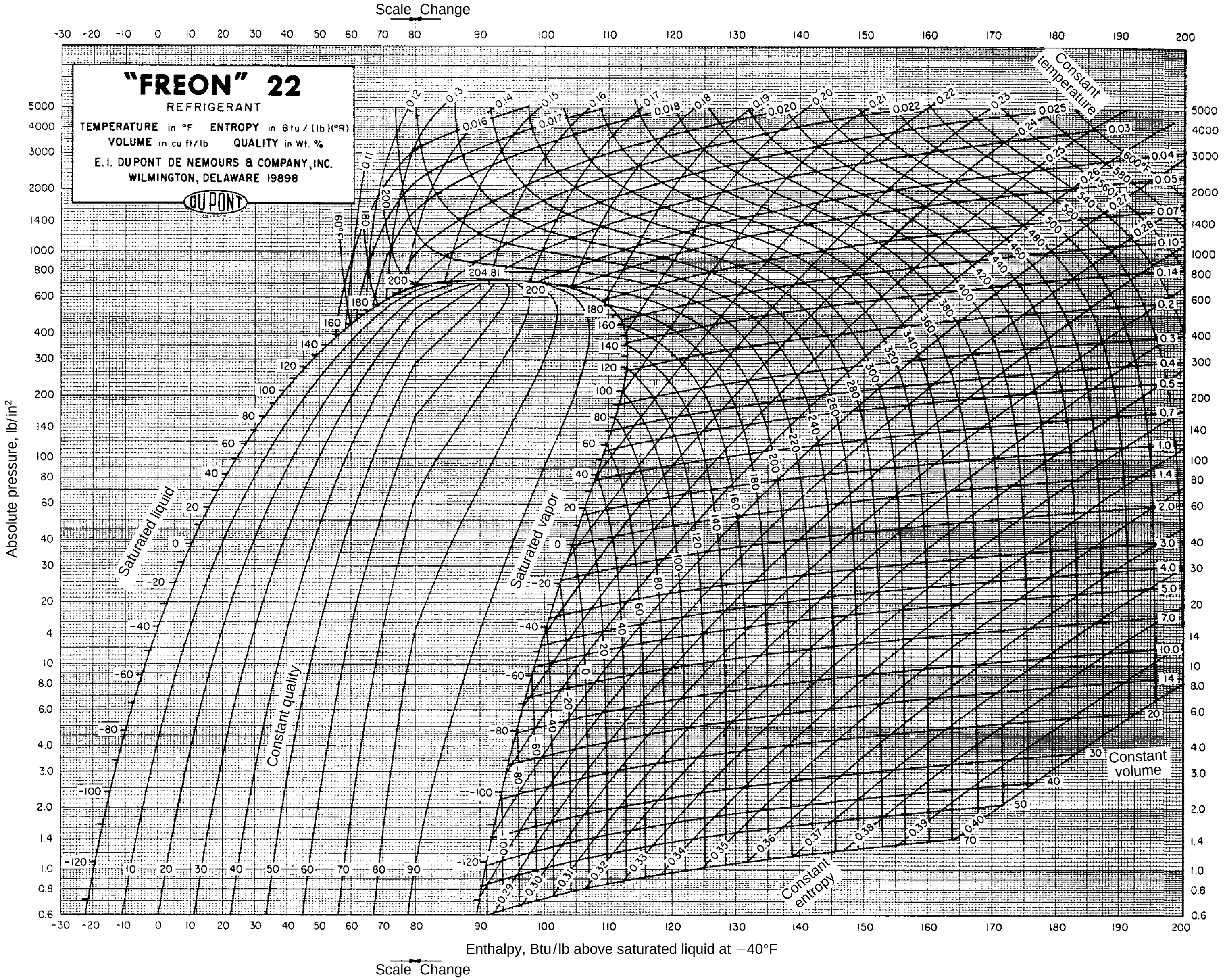

Fig. 19.1.1 Thermodynamic properties of R-22. *(Reprinted with permission of E. I. du Pont de Nemours & Co.)*

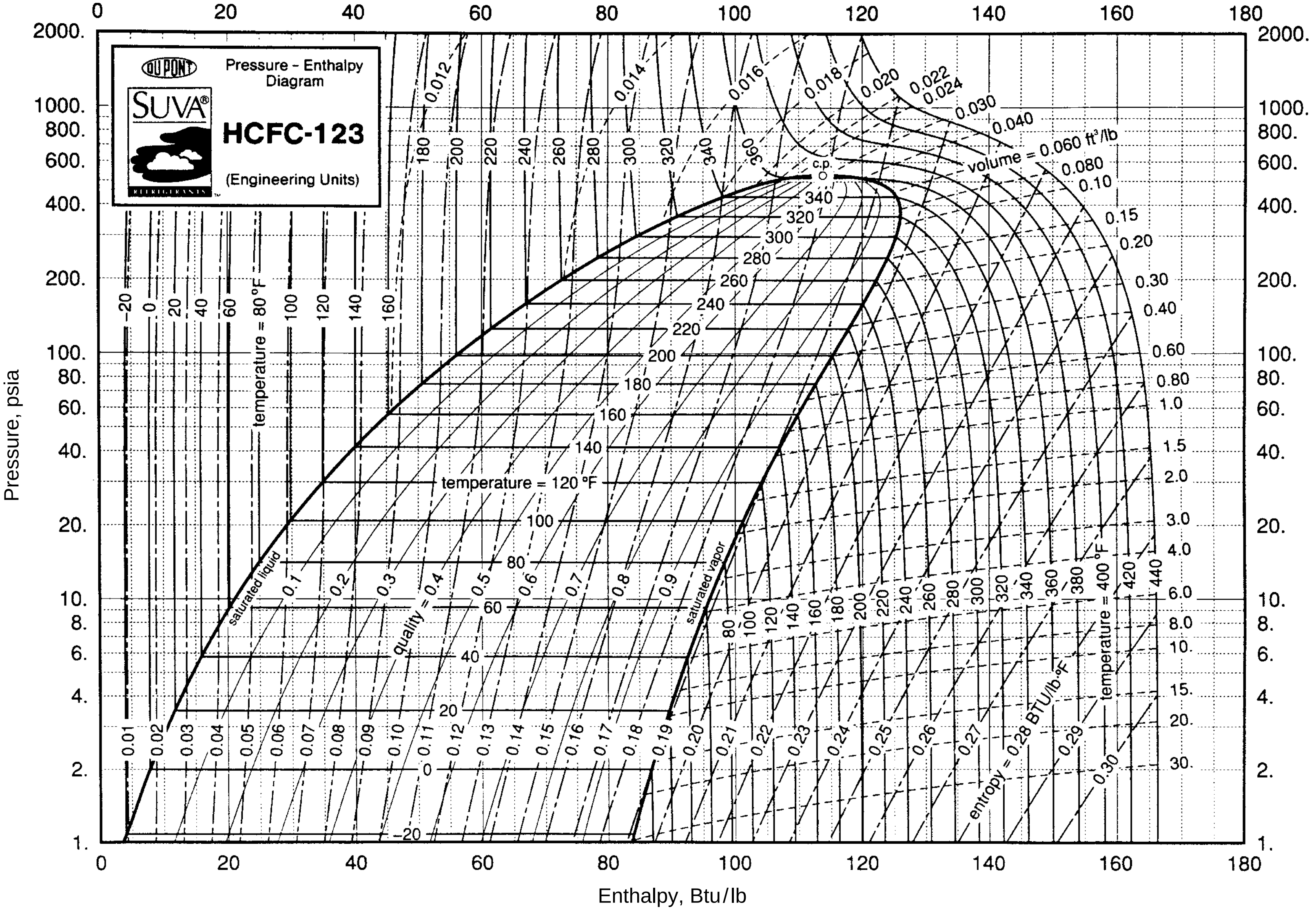

Fig. 19.1.2 Thermodynamic properties of HCFC-123. *(Reprinted with permission of E. I. du Pont de Nemours & Co.)*

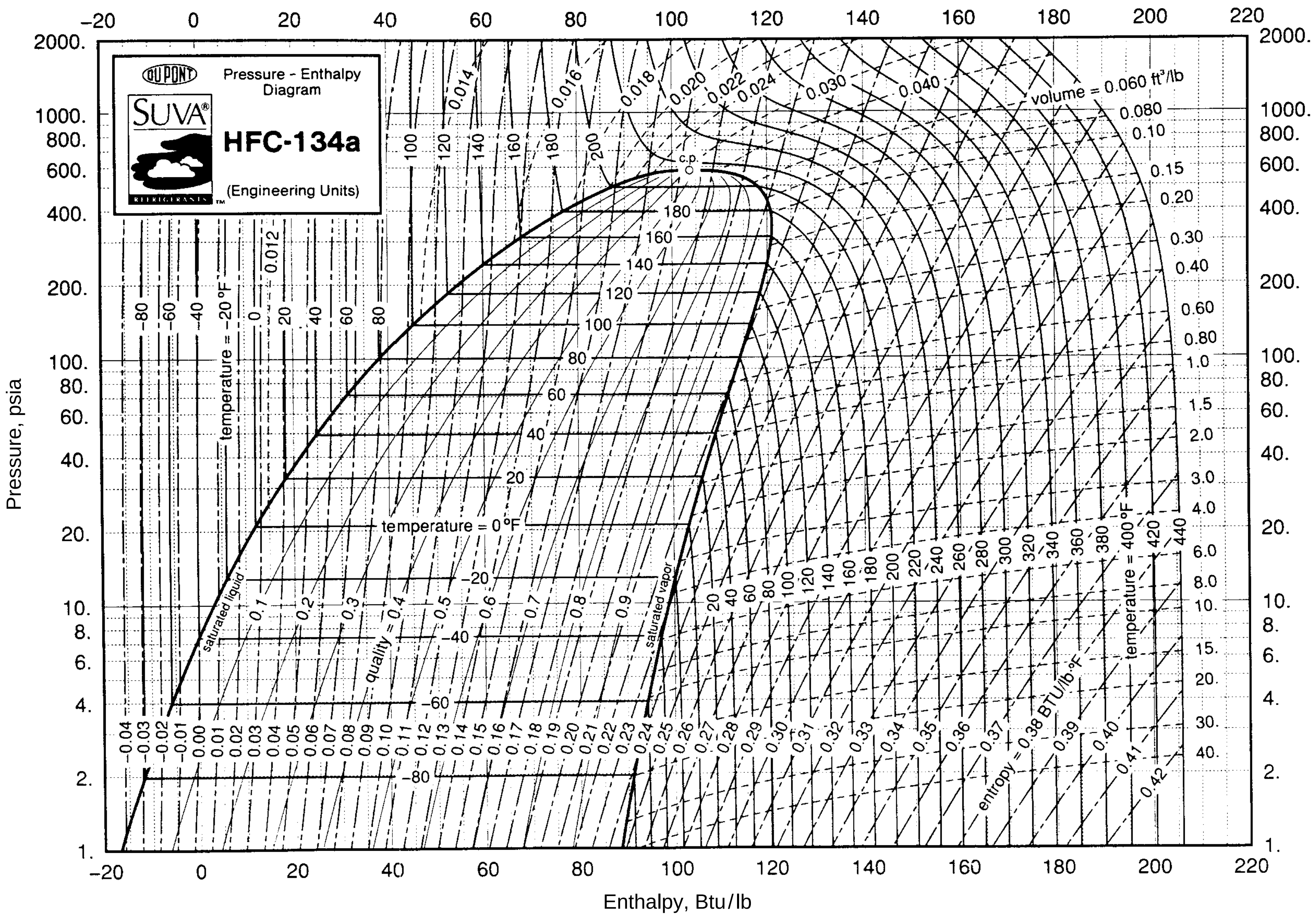

Fig. 19.1.3 Thermodynamic properties of HCFC-134a. *(Reprinted with permission of E. I. du Pont de Nemours & Co.)*

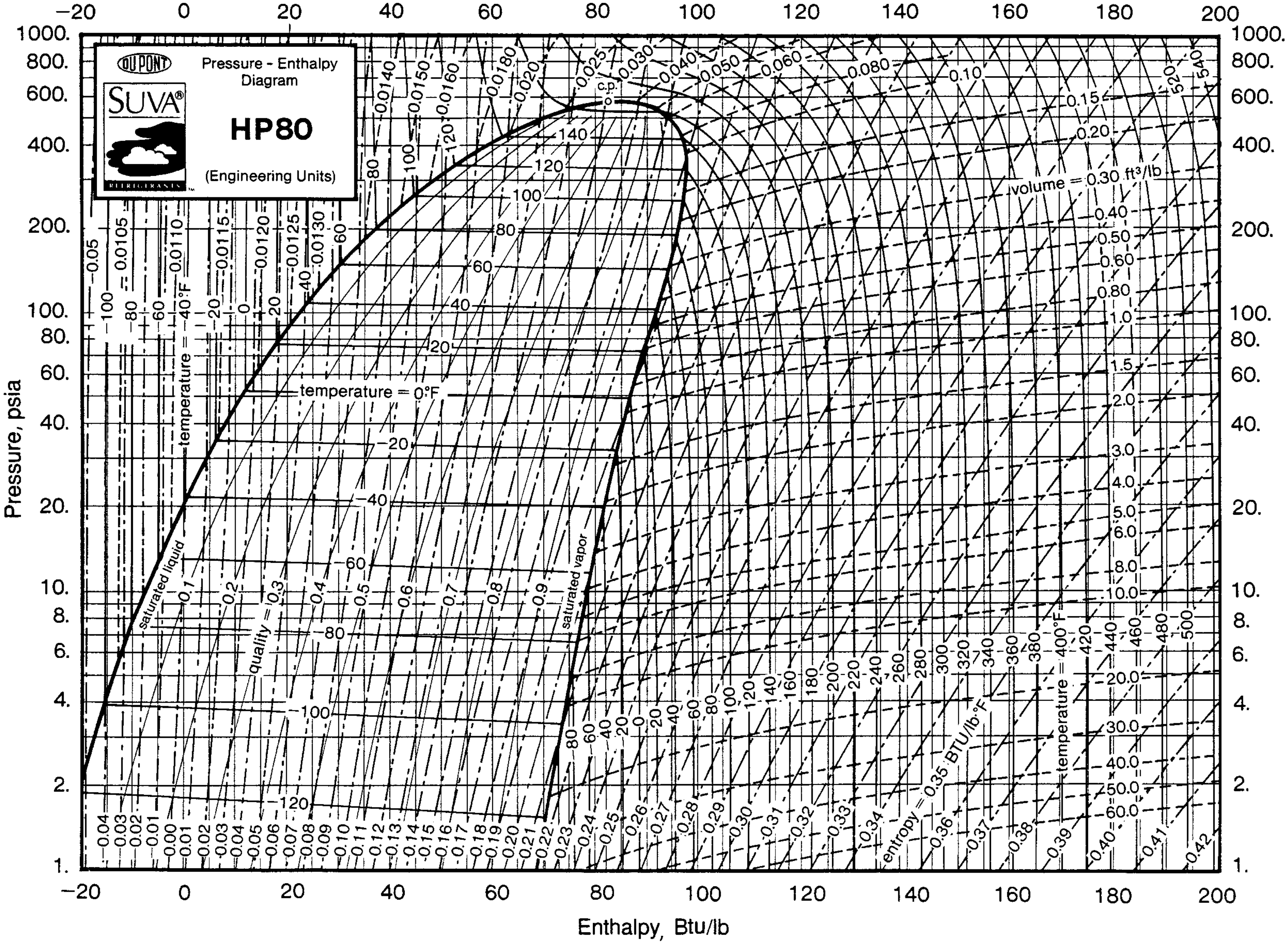

Fig. 19.1.4 Thermodynamic properties of SUVA HP80 (R-402A). *(Reprinted with permission of E. I. du Pont de Nemours & Co.)*

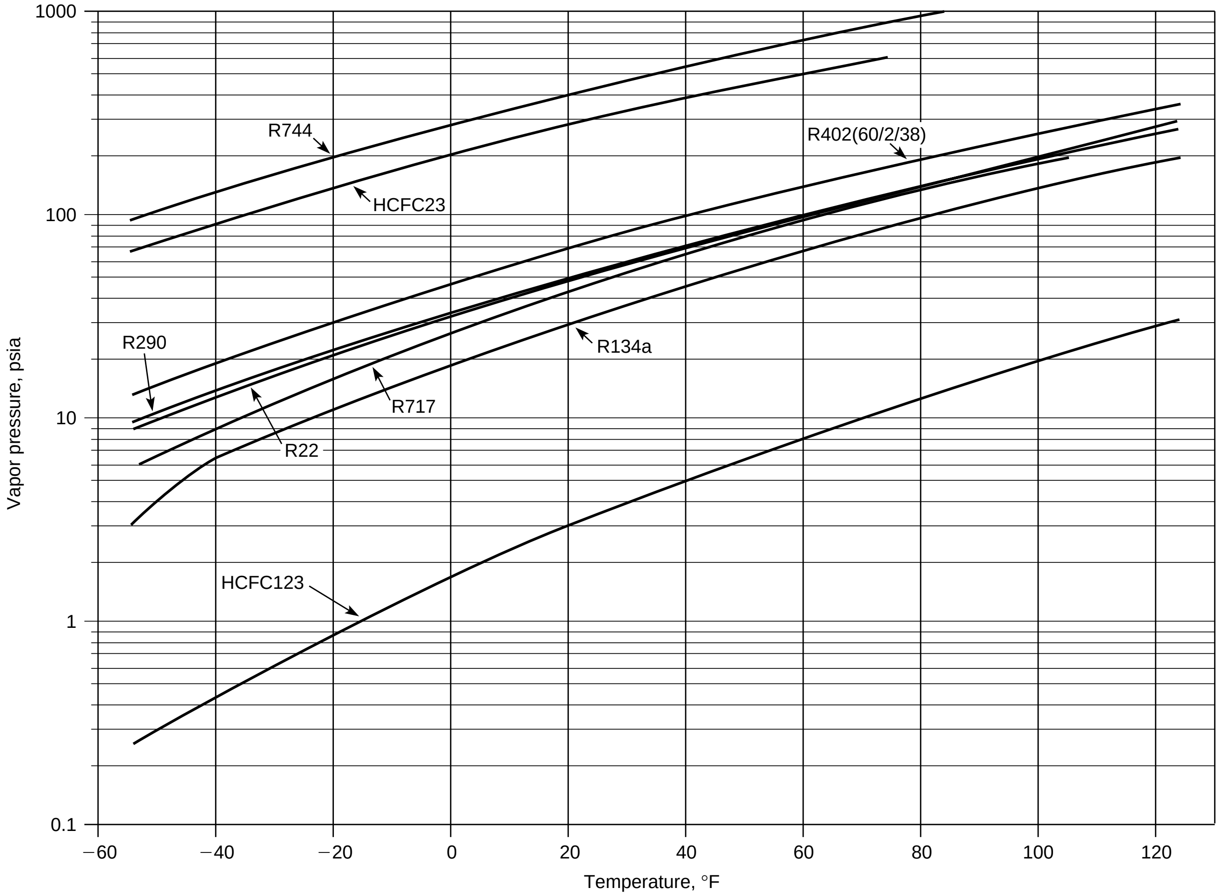

Fig. 19.1.5 Pressure-temperature relations for saturated liquids of refrigerants.

refrigeration lubricant when the system is serviced or charged. The compound travels through the system with the refrigerant and oil mixture. At the point of leakage, the fluorescent compound escapes from the system, and is left as a deposit on the outside surface of equipment. When illuminated by ultraviolet light, the deposit at the leak point becomes visible.

OVERALL CYCLES

Figure 19.1.6 represents the simple closed-circuit **vapor-compression** system. The upper heat exchanger is a vapor condenser (with some superheat), and after the throttling expansion valve, the lower one is the evaporator in which the refrigerant liquid at reduced pressure and temperature evaporates with the inward refrigeration heat flow. The refrigerant vapor is elevated by the compressor to a higher pressure and condensing temperature so that it will liquefy in its transfer of heat to the atmospheric level. Figures 19.1.7 and 19.1.8 illustrate the cycle.

Figure 19.1.9 is the counterpart of Fig. 19.1.6 and represents the closed sensible-heat **gas-compression** cycle, with an expander (either displacement or turbine-type) parallel to a gas compression which may provide a compressor-work ''assist'' in addition to its exhaust stream of cold gas. This is the refrigeration version of the Brayton cycle (see Sec. 4), the upper exchanger serving to cool the hot compressed gas, and the lower exchanger handling the sensible-heat refrigeration effect. In place of the expander, an expansion valve could be used; this is particularly

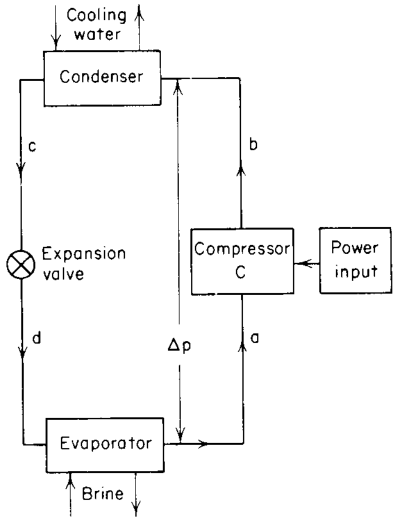

Fig. 19.1.6 Simple closed-circuit vapor-compression system.

effective on very low-temperature open-cycle operations, such as with gas liquefaction processes incorporating secondary ''cold-regenerating'' heat exchangers.

An extended temperature-entropy diagram (Fig. 19.1.10) illustrates paths for gas-compression sensible-heat cycles as follows: (1) 1-2-3-4 isentropic expander; (2) 1-2-3-4′ polytropic expander; (3) 1-2-5-6 throttling (isenthalpic); and (4) 1-2-7-8 throttling (low temperature).

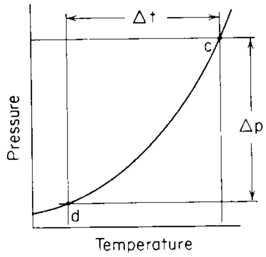

Fig. 19.1.7 Pressure-temperature curve illustrating overall differences.

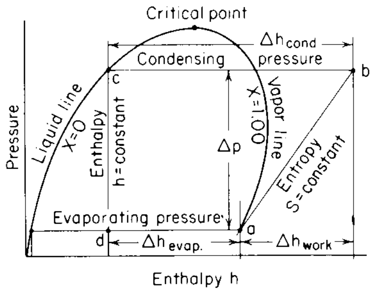

Fig. 19.1.8 Pressure-enthalpy diagram for simple ideal refrigeration cycle.

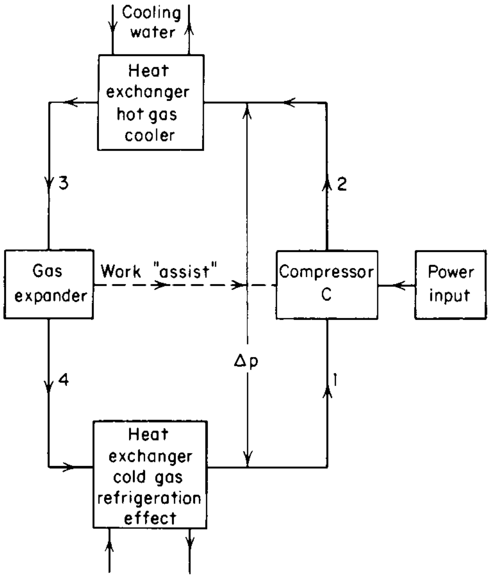

Fig. 19.1.9 Closed sensible-heat gas-compression cycle.

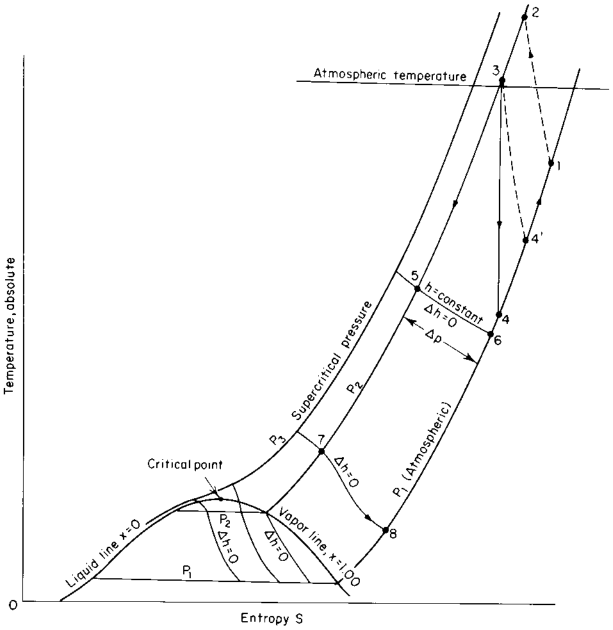

Fig. 19.1.10 Extended temperature-entropy diagram for air illustrating sensible-heat air-compression processes.

COMPONENTS OF COMPRESSION SYSTEMS

A refrigeration system is an energy-transport complex of assorted components. It may be a conventional system in which the same fluid, the refrigerant, recirculates continuously in a closed circuit, or it may entail a partially open system with discharge of some of the processed fluid either as a liquefied gas or as a solidified gas, with replacement makeup, e.g., by air liquefaction and solid carbon dioxide production. Components include vapor and gas compressors; liquid pumps; heat-transfer equipment (gas coolers, intercoolers, aftercoolers, exchangers, economizers); vapor condensers and the counterpart evaporators; liquid coolers and receivers; expanders; control valves and pressure-drop throttling devices (capillaries, refrigerant-mixture separating chambers, stream-mixing chambers); and connecting piping and insulation.

Codes and **standards** have been published by various associations and societies. These cover design, selection, designation, testing, sound ratings, safety, efficiency, and other areas relating to mechanical refrigeration. Publication of a comprehensive list is impractical because of frequent revisions. Several of the organizations include Air Conditioning Contractors of America (ACCA), American National Standards Institute (ANSI), Air-Conditioning and Refrigeration Institute (ARI), American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE), The American Society of Mechanical Engineers (ASME), Canadian Standards Association (CSA), International Institute of Ammonia Refrigeration (IIAR), and Underwriters Laboratories Inc. (UL). A list of current codes and standards can be obtained by contacting the particular organization.

Several of the more general refrigeration equipment standards include: Safety Code for Mechanical Refrigeration, ANSI/ASHRAE 15; Number Designation and Safety Classification of Refrigerants, ANSI/ASHRAE 34; and General Standard on Refrigeration Equipment, CAN/CSA-C22.2.

There are numerous additional standards relating to specific types of refrigeration equipment (air conditioners, heat pumps, chillers) or specific components (compressors, water- or air-cooled condensers, lubricating oil). Because refrigeration equipment requires auxiliaries (electric motors, steam turbines, piping, insulation) and is part of a larger system such as food processing or heating, ventilating, and air conditioning (HVAC), other standards not relating directly to refrigeration may apply. Any installation must also comply with building and fire code requirements.

Compression may be single stage (Fig. 19.1.6) or multiple stage. Displacement compressors include reciprocating piston, screw, rotary vane, scroll, and trochoidal compressors. The volumetric capacity of the compressors is always less than the swept volume of pistons or the volume displaced in cyclic rotation by the other forms of displacement devices such as screws, nutating scrolls, or rotating vanes. The true measure of compressor capacity is the volumetric efficiency. It is influenced by cylinder, screw mesh, or cavity clearance volume; pressure ratio; nature of the gas; pressure drops and passage volume in the valves or vanes; suction heating; and the equivalent compression exponent n. The numerical value of n expresses the effect of heat transfer to or from the gas during compression of the gas. Normally it lies between $n = 1.0$ and $n = k$ (adiabatic), such as for the ideal (isentropic). Figure 19.1.11 indicates the comparative volumetric efficiency of a pair of reciprocating and screw compressors. **Reciprocating compressors** are disadvantaged by high discharge temperatures and excessive clearance volume losses associated with compression ratios greater than 12 to 1 in a single compression stage. Compound compression staging is a practical solution if reciprocating machinery is applied across cycle compression ratios greater than 10 to 1. **Screw** compressors with oil injection systems can operate reliably as a single stage with compression ratios greater than 20 to 1. **Rotary vane** compressors are fixed-volume-ratio machines and can be used across compression ratios of 18 to 1, vane material and lubricating oil permitting; however, in the larger-size machines, a 7 to 1 compression ratio is commonly used in order to maintain efficiency. **Scroll** compressors have been used in commercial practice for systems that have capacities less than 10 tons (35 kW). **Trochoidal** compressors are often called ''Wankel'' compressors because, in one form, the rotor is similar to the Wankel internal combustion engine rotor.

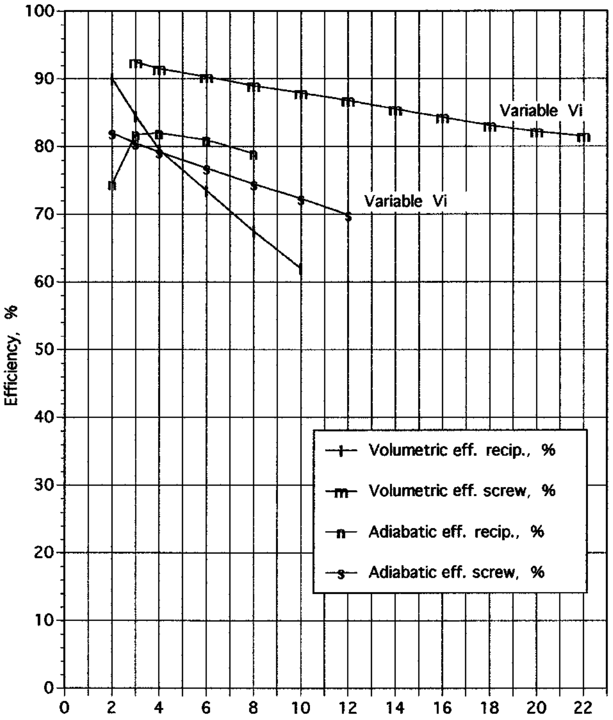

Fig. 19.1.11 Screw and reciprocating compressor volumetric and adiabatic efficiency versus pressure ratio. *(Courtesy of J. W. Pillis, ''Development of a Variable Volume Ratio Screw Compressor,'' Frick Company, 1983.)*

Reciprocating-piston compressors may be horizontal or vertical. The pistons can be designed to be either single acting or double acting with compression occurring on either stroke of the piston. Commercial designs have evolved into multiple cylinder machines arranged in a V block similar to an automotive engine. Average piston speed may be as great as 800 ft/min (4 m/s). Design characteristics of the suction and discharge valves installed on the cylinder determine the energy efficiency, and maintenance frequency. Valves most frequently used are the free-floating reed, clamped reed, and ring valve. Gas velocity through the valves is limited to 12,000 ft/min (61 m/s) with R-717 and 9,000 ft/min (46 m/s) with HCFC-22. For additional information on reciprocating compressors in refrigeration service, refer to the ASHRAE Handbook, 1992 Systems and Equipment, pp. 34.4 to 34.9.

Screw compressors are available in several designs, both single screw and twin screw, with oil-free and oil-injected designs in both types. Twin-screw oil-injected compressors are slightly more energy efficient at moderate compression ratios. Many of the commercially available screw compressors are fixed-volume-ratio machines. (See also Sec. 14.) If the compression ratio varies, the operational efficiency of the compressor also varies significantly. The energy reductions that are available by operating with cooler condensing pressures during spring, fall, and winter may not be fully achievable. A movable slide valve stop which will vary the compressor internal volume ratio to achieve optimum energy consumption is now available. Figure 19.1.11 illustrates the adiabatic efficiency of a twin-screw compressor with a movable slide stop control and a reciprocating compressor. Flash intercoolers (Fig. 19.1.13), also called *economizers*, can be provided to improve energy efficiency on the larger-capacity machines.

Air-cooled compressors are used where discharge temperatures are low; water cooling is used where discharge temperatures are high, as with ammonia, and on larger industrial units. **Oil separators** return to the compressor the lubricating oil carried over by the refrigerant vapors. **Rotary compressors** of the vane, eccentric, gear, and screw types are in use.

Kinetic compressors include high-speed centrifugal and axial flow machines, usually multistaged, and jet-entrainment devices. Centrifugal machines are especially adapted to high-volume flow (> 500 ft^3/min) (0.24 m^3/s). (See also Sec. 14.)

The entire machine-compression operation may be replaced by a secondary absorber-pump-generator system, in which the complex is known as an absorption refrigeration system.

Dual (or **multiple effect**) **compression** may be used when refrigeration at two temperatures is desired. The compressor takes vapor from a lower temperature expansion coil during the first part of its intake stroke, and from a higher temperature expansion coil at or near the end of the stroke. The mixture is then compressed and condensed.

In **wet compression,** cooling is obtained during compression by spraying liquid refrigerant into the compressor cylinder. The desuperheating of the compressed vapors results in better heat transfer in condensers and more nearly isothermal compression; the disadvantages are reduced compressor capacity and the problem of control of the amount of injection.

Table 19.1.7 shows some ideal performance values for a single-state ammonia system under different conditions.

Condensers are usually shell-and-tube type, with the refrigerant passing outside the tubes. Older industrial installations still use **double-pipe condensers.** In double-pipe condensers, gas flows between the two pipes while water passes through the inner. The outside pipe diameter may be 2 in (5 cm); the inner, 1¼ in (3.2 cm). In some instances, the pipes are exposed to the atmosphere either with or without water drip. In an **evaporative condenser** the refrigerant vapor is condensed as it passes through tubes over which water is sprayed; the water is then evaporated by air flowing over the wet tubes. In this way, cooling-water requirements are reduced to from 5 to 15 percent of the water requirements of a nonevaporative condenser with no water reuse. The evaporative condenser combines in a single unit a refrigerant condenser and the atmo-

Table 19.1.7 Theoretical Horsepower and Theoretical Volume of Dry Ammonia Gas Pumped per Minute to Produce 1 Ton of Refrigeration

Suction pressure and temperature		Condenser pressure, lb/in^2 gage (temperature, °F)									
		103 (65)		127 (75)		153 (85)		182 (95)		215 (105)	
lb/in^2 gage	°F	hp	ft^3/min	hp	ft^3/min	hp	ft^3/min	hp	ft^3/min	hp	ft^3/min
4	− 20	1.058	5.84	1.205	5.96	1.361	6.09	1.525	6.23	1.691	6.43
6	− 15	0.997	5.35	1.145	5.46	1.300	5.58	1.461	5.70	1.546	5.83
9	− 10	0.903	4.66	1.045	4.76	1.193	4.86	1.347	4.97	1.435	5.08
13	− 5	0.818	4.09	0.954	4.17	1.094	4.25	1.244	4.35	1.321	4.44
16	0	0.735	3.59	0.865	3.66	1.002	3.74	1.147	3.83	1.219	3.91
20	5	0.666	3.20	0.795	3.27	0.928	3.34	1.066	3.41	1.138	3.49
24	10	0.592	2.87	0.726	2.93	0.854	2.99	0.991	3.06	1.060	3.12
28	15	0.541	2.59	0.664	2.65	0.792	2.71	0.922	2.76	0.994	2.82
33	20	0.474	2.31	0.592	2.36	0.715	2.41	0.842	2.46	0.903	2.51
39	25	0.410	2.06	0.523	2.10	0.599	2.15	0.767	2.20	0.829	2.24
45	30	0.351	1.85	0.461	1.89	0.576	1.93	0.694	1.97	0.759	2.01
51	35	0.300	1.70	0.410	1.74	0.521	1.77	0.640	1.81	0.701	1.85

spheric cooling tower or spray pond which is required if the cooling water is to be reused. Air-cooled finned-tube condensers with forced ventilation are widely used on small units, and shell-and-coil or double-tube water-cooled condensers on medium units.

Evaporators may have plain or finned surfaces. Fin density is available from 1 to 14 fins per inch. Fin density of 2 to 4 fins per inch is specified for air-cooling service below 28°F (− 2.2°C), because air can flow through the coil while thin layers of frost are forming. Four types of defrosting methods are used: hot gas from the compressor, electrical air heaters, hot water, and room air circulation. Flooded evaporators are operated practically full of refrigerant. The level is controlled by a float valve. Wet-expansion evaporators are operated with a level approaching flooded operation. Dry-expansion (once-through) units operate with an indefinite amount of liquid in the evaporator. Usually a thermal bulb measures the outlet pipe temperature and adjusts a thermostatic expansion valve to maintain 15°F (8.3°C) superheat in the outlet vapor. Flooded or wet operation gives high heat-transfer rates. Pumped recirculation of refrigerant is used to promote heat transfer in finned coil evaporators, and also on some shell and tube evaporators (liquid chillers) by completely wetting the surfaces with a rapidly moving layer of refrigerant. The most commonly used liquid feed ratio is 3 to 1. However, a feed ratio as high as 6 to 1 may be required to accommodate distribution imbalances in the piping system. Combined liquid and vapor **return piping** should be sloped at 0.25 in/ft (2.08 percent) to prevent formation of waves in the liquid-phase layer that can react with the flowing vapors to cause liquid hammer that, in turn, can severely damage piping components. **Receivers** and **recirculating pumps** must be carefully matched to the operating system to prevent pump cavitation, liquid droplet carryover, and gross liquid overlow due to surges of liquid that may occur as a result of equipment operations such as defrosting processes. Specifically, the tank drop leg liquid velocity should be limited to 100 ft/min (0.51 m/s) and the vapor velocity over the liquid level in the tank should also be limited to 100 ft/min (0.51 m/s).

For a low-cost unit operating in limited space under conditions of motion, dry operation is preferred.

Controls are required on the liquid level of the refrigerant and on the temperature of the refrigerated space. The liquid control regulates the flow of refrigerant into the evaporator and also serves as the pressure barrier between the high operating pressure of the condenser and the lower operating pressure of the evaporator. It may take the form of a **capillary tube** between the condenser (high side) and the evaporator (low side), in which case it is nonadjustable. Plugging due to dirt is a common difficulty. Capillary-tube liquid control is largely confined to relatively small units assembled and charged at the factory and particularly for hermetically sealed systems. The **constant-pressure expansion valve,** maintaining a constant evaporator pressure, and the **thermal expansion valve,** maintaining a constant superheat leaving the evaporator, are standard liquid controls for most commercial applications. A **low-side float** liquid control, used with a flooded evaporator operating at evaporator (low) pressure, consists of a float-operated valve to admit liquid refrigerant to the evaporator in accordance with demand so that a constant liquid level is held in it. A **high-side float** liquid control is often used with a single flooded evaporator; the float operating the valve between the evaporator and the condenser is in a float chamber containing liquid refrigerant at the condenser (high-side) pressure. As the liquid level in the float chamber falls, the valve closes, thus preventing hot gaseous refrigerant from passing from the high to the low side as is possible when using a capillary or a low-side float.

Several types of noncondensable-gas removal devices have been developed to remove tramp air from refrigerating systems. The exhaust of the purge system contains a portion of the refrigerant. This refrigerant loss is detrimental to the environment, and the replacement of lost refrigerant is costly. Three types of refrigerant purge systems have been developed. One uses a scavenging condenser, cooled by system refrigerant, followed by a carbon absorption device to remove the last vestiges of refrigerant from the purge vapors. Other types, such as oil-absorption systems and reverse-osmosis systems, are developed but not offered commercially. The efficiency of purge systems is specified by the weight ratio of refrigerant lost to the air that is exhausted. Present-day systems release as little as 0.0047 unit of refrigerant per unit of released air. Systems designed to meet earlier requirements may lose 3 to 7 units of refrigerant per unit of air.

Refrigerant charge in a system must be removed in an environmentally acceptable and economical manner to accomplish occasional repairs. Pump-out systems remove the refrigerant from equipment and store it for reuse. These systems consist of a compressor, air-cooled condenser, and receiver tank. They are available in portable or stationary configurations. The capture and recycle of CFC and HCFC refrigerants are mandated by the 1990 Clean Air Act.

Vapor-Compression Circuits

Compound Low-Temperature Systems Multistaging is advantageous as it reduces operating temperatures by intercooling and power requirements if the fluid superheats considerably on adiabatic compression, as with ammonia; with F-12 refrigerant, propane, and butane, the superheat is negligible (see Figs. 19.1.12 to 19.1.15).

Gas Intercooler Plus Liquid Injection Since gas temperature in the intercooler is limited by cooling water temperature, further favorable reduction of gas temperature, prior to entrance to the second stage, may be accomplished by direct injection, via an expansion valve, of refrigerant liquid (Fig. 19.1.12).

Flash Intercooler With a flash intercooler separator located midway between successive expansion valves, the gas leaving the low-stage (I) compressor passes through the flash intercooler, gets cooled by direct contact with the liquid, and, augmented by the flash vapor and at reduced temperature, enters the second stage (II) (Fig. 19.1.13).

For positive-displacement compressors, selection of flash intercooler temperature is determined by the following relation:

$$P_{\text{intermediate}} = (P_{\text{I suction}} \times P_{\text{II discharge}})^{1/2}$$

where P = pressure in absolute units, lb/in^2 abs (Pa). This equation is valid for centrifugal machines with modification. The common practice is to limit single-impeller-stage adiabatic head to approximately 10,000 ft (3,000 m) for low-molecular-weight gas and 6,000 ft (1,800 m) for the heavier HCFC and HFC refrigerants. Economical rotor design may require an interstage pressure that does not match the ideal flash intercooler pressure. Where choice is required, select a pressure slightly greater than the ideal pressure given by the above equation.

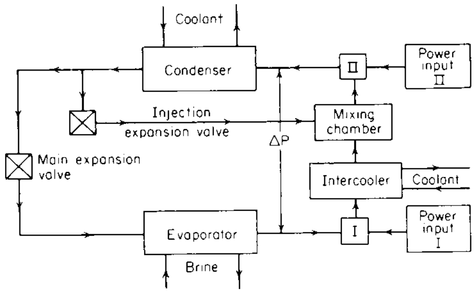

Fig. 19.1.12 Two-stage system with conventional intercooler plus refrigerant injection.

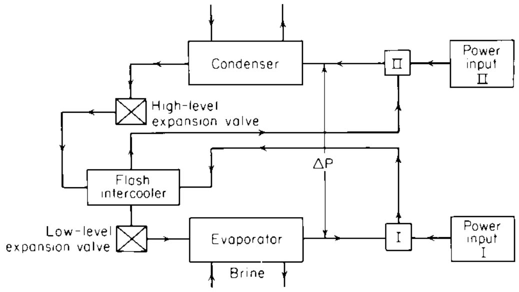

Fig. 19.1.13 Two-stage system with flash intercooler.

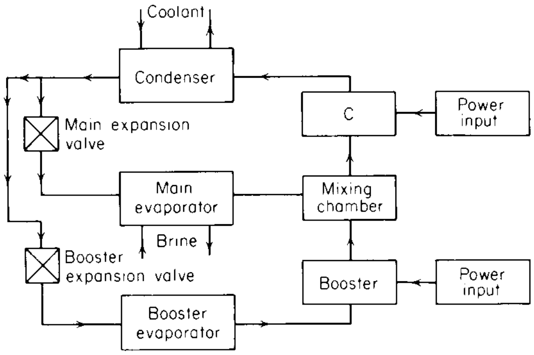

Fig. 19.1.14 Two-stage system incorporating low-temperature booster circuit.

Low-Temperature Booster Plus Single Stage For the addition of a second lower-temperature evaporator to an existing single-stage system, a booster for the low-temperature vapor may be employed, its output feeding into the single-stage suction. The booster is a centrifugal or rotary compressor, suitable for handling the large vapor volumes encountered at low temperatures (and pressures); the reciprocating compressor is more advantageous at high pressures (Fig. 19.1.14).

Cascade Systems with Segregated Circuits Here two separate circuits are employed with the low-stage vapor condensed in the second-stage evaporator. Two different refrigerants may be appropriately chosen, a lower-boiling-point refrigerant serving for the first stage. Thus, for low temperatures (< -50°C or -50°F), CO_2 may be employed in the lower stage with NH_3 in the second stage condensing the CO_2. Or the same refrigerant may be used in both stages, limiting the migration of the lubricant to the low-temperature realm (Fig. 19.1.15).

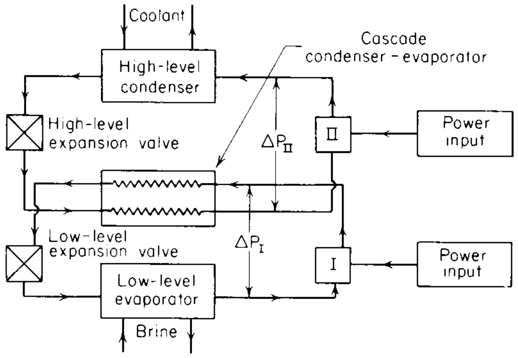

Fig. 19.1.15 Two-stage system in cascade form.

Water-vapor refrigeration is necessarily limited to near-atmospheric temperature levels; it involves large suction volumes at high vacuum and utilizes a nontoxic refrigerant; this type of refrigeration was frequently used in the early days of space air conditioning. It utilizes (1) centrifugal compressors or (2) steam-jet compressors, entraining low-density water vapor from the evaporator. The centrifugal water-vapor compressor, usually multistage, must operate at high speeds (7,000 to 10,000 r/min). Operation at high vacuum in the evaporator requires an air pump at the condenser to eliminate leakage effects.

Steam-jet refrigeration systems are used where cooling to temperatures above 0°C is desired. Applications include industrial air conditioning and cooling of city gas to condense out tar and other objectionable impurities, of gas absorbers to increase efficiency of absorption, of reaction units where heat removal and temperature control are important during chemical transformations, and of wort and mash in the brewing and other fermentation industries particularly in the summer months. Jet refrigeration has been used for cooling passenger trains and is coming into popularity for marine installations, e.g., cooling of banana boats and large passenger vessels. The system is a compression-type refrigerator: it uses water as a refrigerant, a part of which is evaporated to produce cooling of the remainder, steam-jet ejectors to compress the water vapor resulting from evaporation, and a condenser, either the surface type, where refrigerant water is to be reused, or the barometric type, where refrigerant water can be discarded. The advantages of the system are: low installation cost; the absence of moving parts except for small liquid pumps; and safety, since no noxious or toxic refrigerants are used.

For considerable differences in temperatures between condensing-water temperature and refrigerating temperatures (10°C or more), cooling of the refrigerant water by evaporation in two stages operating at different pressures (and temperatures) will usually give better operating economy, but at somewhat higher installation cost.

Figure 19.1.16 shows the variation of refrigerating capacity with variation of the chilled-water temperature, capacity increasing as chilled-water temperature rises.

Household compression machines are designed for continuous automatic operation and for conservation of the charges of refrigerant and oil. These units are almost universally motor-driven air-cooled compressors, the principal exception being the Electrolux-Servel absorption unit. The compressor may be hermetically sealed, with the compressor and motor enclosed in the same casing, or may use a shaft seal, embodying some form of sylphon bellows, with an outside coil spring at the place where the crankshaft passes out of the casing. The compressor may be reciprocating, but rotary types are being increasingly used either with a floating piston ring driven by an eccentric or with sliding blades. Lubrication is by internal forced-feed circulation in most cases. The condenser may be of the radiator, coil, or plate type, cooled by natural draft, by a fan on the main motor, or by a separate motor. Refrigerant feed control to the evaporator may be by (1) float feed (flooded system), (2) pressure-actuated diaphragm valve, or most commonly (3) fixed orifice. The fixed orifice may be either a plate orifice or a capillary tube. The evaporator is usually made of aluminum. Controls are generally pressure-operated thermostatic switches. The power consumption of an average home refrigerator (18 ft^3) (0.5 m^2) is 133 kWh per month. A typical home deep freezer consumes approximately 152 kWh per month.

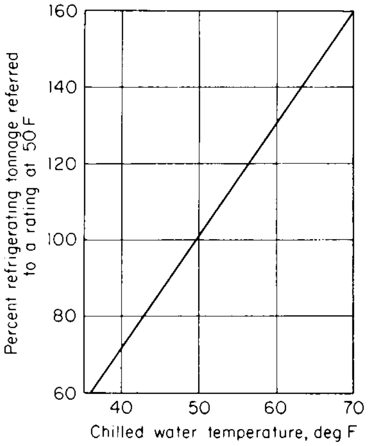

Fig. 19.1.16 Variation of refrigerating capacity with the jet-chilled water temperature in steam refrigeration.

ABSORPTION SYSTEMS

Absorption refrigeration machines are essentially vapor-compression plants (Fig. 19.1.6) in which the mechanical compressor has been replaced by a thermally activated arrangement (Fig. 19.1.17). The basic element are the **absorber, pump, heat exchanger, throttle valve,** and **generator**.

In an absorption cycle, a secondary fluid, the **absorbent,** is used to absorb the primary fluid, the **refrigerant,** after the refrigerant has left the evaporator. The vaporized refrigerant is converted back to liquid phase in the absorber. Heat released in the absorption process must be rejected to cooling water. The solution of absorbent and refrigerant is then pumped to the generator. A heat exchanger may be used for heat recovery and a corresponding increase in efficiency. In the generator, heat is added, and the more volatile refrigerant is separated from the absorbent through distillation. The refrigerant continues to the condenser, expansion valve, and evaporator, while the absorbent returns to the absorber.

Although there are many refrigerant-absorbent combinations, the most common systems are **ammonia-water** and **water-lithium bromide**. A common variation of the absorption system is the **Electrolux-Servel** process (Platen-Munters Patent). It has had wide application in gas-fired household refrigeration machines and air-conditioning systems. The use of hydrogen in a hermetically sealed system eliminates the use of pumps and other moving parts.

Absorption machines represent only a small percentage of refrigeration systems in operation. This is because they are generally bulky and difficult to operate efficiently. There has been renewed interest in absorption machines with the advent of high energy costs. This is because an absorption machine is an excellent way to utilize waste heat. Examples of potential applications are varied. One example may be on an exothermic process, or where extensive use of steam turbine with limited turndown results in vented steam especially during warm weather when air conditioning is required. A site where peak electrical demand is set by air-conditioning load and ratcheted demand charges are high may justify absorption refrigeration if adequate steam-generation facilities are already present.

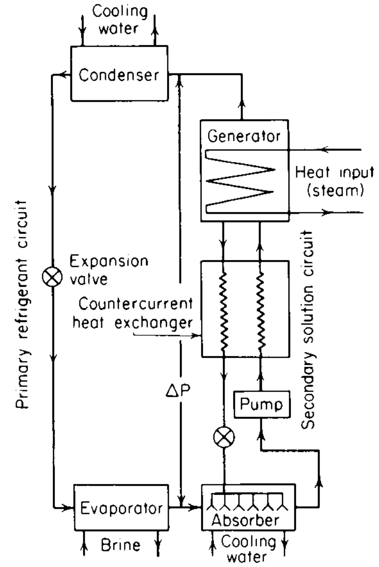

Fig. 19.1.17 Elemental absorption system circuit.

A typical absorption machine requires about 20 lb (9 kg) of steam for 1 ton (3.5 kW) of refrigeration effect at 45°F (7°C).

A more detailed explanation of absorption systems is found in ASHRAE Handbook, 1993 Fundamentals, pp. 1.20 to 1.25.

AIR MACHINES

Compressed-air machines in which air is compressed, cooled, and then adiabatically expanded in an engine while doing work are obsolete for usual industrial refrigeration because of bulk and inefficiency. Their coefficient of performance is less than unity as compared with 4 or 5 for vapor machines. Operation may be closed cycle, in which the air is reused, or open cycle, in which the air is discarded after expansion. Compression ratios of about 3 seem best. In the **dense-air** closed system, the pressure at the compressor intake averages 75 lb/in^2 gage (560 kPa) and up to 250 lb/in^2 gage (1,700 kPa) at the outlet. The use of superatmospheric pressure throughout the cycle decreases the size, and the reuse of dried air eliminates the ice formation on engine valves which occurs with open-system operation. Air cycles are used for liquefaction of the so-called permanent gases and to an increasing extent for comfort cooling in high-speed planes where the small lightweight equipment and the safety of the cooling medium are important. High-speed (90,000 to 100,000 r/min) gas turbines are used to extract work from 5 to 100 lb/in^2 gage compressed air.

THERMOELECTRIC COOLING

Thermoelectric cooling utilizes the **Peltier effect** whereby a temperature differential will occur between two junctions of a closed loop of dissimilar metallic conductors when an electric current is imposed. It is variously called *Peltier effect cooling, thermoelectric refrigeration,* and *electronic cooling.* It is the inverse of the thermocouple operating principle whereby a temperature differential across the junctions of two dissimilar metals produces an electric current.

The capacity of a thermoelectric couple is small. System capacity can be increased by increasing the number of elements used. The temperature produced can be lowered by cascading the systems. Capacities approaching 25 tons (87.9 kW) cooling effect have been achieved, and temperatures as low as − 266°F (− 166°C) have been attained.

Aside from limited practical capacities, the major disadvantage of a thermoelectric cooling system is its poor coefficient of performance. The absence of moving parts, silent operation, ability to function in zero gravity, and lack of pressurized vessels, has given thermoelectric cooling wide space-program application. Other applications include use on submarines, electronic equipment cooling, and small units such as drinking water coolers or units for use in recreational vehicles.

The elements of a thermoelectric cooler are quite simple and are equivalent to those of a thermoelectric generator (see Sec. 9). For development of various output capacities, the number of basic elements is proportionately increased; they all act in parallel, and their outputs add together. **Semiconductor** materials, such as bismuth-telluride-selenide and bismuth-antimony-telluride alloys, are employed.

With an input to the circuit of low-voltage direct current and with heat continuously abstracted at one junction at room temperature, the other junction will become cold. Essentially each of the two junctions becomes an ''activity cell,'' in which at the lower refrigeration temperature level, heat is converted into electrical effect, and at the higher atmospheric temperature, the electrical effect is converted into heat. This establishes the basis of ''heat-pump'' operation: the lower-temperature heat requirement must be supplied, essentially as in the evaporator of a vapor-compression system, by heat withdrawal via heat transfer, from all connected surroundings. The energy-transport circuit thus parallels the machine-activated circuit of Fig. 19.1.6. The amount of heat for a single-component circuit depends on the circulating electric current, on the properties and dimensions of the two conductors (*p* and *n* types), and on the resultant temperature difference between the high side room temperature and the low side input heat that is divided into two groups, useless and useful. The **useless heat** is the heat conduction between the high and low temperature through the thermoelements plus the current-generated resistance-loss heat within the elements that flows to the colder junction. The **useful heat** is the refrigeration effect in a regular evaporator. All the heat must in turn be discharged via the Peltier effect at the higher room temperature, as in the condenser of a vapor-compression system. Ultimate performance of the circuit depends on the voltage-generating nature of the two dissimilar conductors and on their electrical resistance, thermal conductivity, and physical dimensions. The different properties are mathematically combined into a **figure of merit, Z.**

METHODS OF APPLYING REFRIGERATION

In **direct expansion systems** the evaporator is placed in the space which is to be cooled; in **indirect systems** a secondary fluid **(brine)** is cooled by contact with the evaporator surface, and the cooled brine goes to the space which is to be refrigerated. Brine systems require 40 to 60 percent more surface than do direct expansion; they have an equalizing effect due to the large heat capacity of cold brine, they are safer (particularly if the refrigerating effect must be carried considerable distance or widely distributed), and they permit closer temperature regulation than is possible with direct expansion. If two temperatures are to be held, the lower may be direct expansion, the higher by brine cooling. Development of better controls and newer piping methods has made direct expansion more attractive than previously.

Brines used for industrial refrigeration are usually aqueous solutions of calcium chloride, ethylene glycol, propylene glycol, or undiluted methylene chloride, and silicone-based alkylated fluids. Calcium chloride should not contain over 0.2 percent magnesia, calculated as magnesium chloride. Calcium chloride brines are recommended down to − 45°F (− 43°C). Brines should be chemically neutral: acidic brines attack ferrous materials, alkaline brines attack zinc, ammonia in brine (resulting from leaks in the ammonia system) is especially harmful to most nonferrous metals. Corrosion by brine is increased by the presence of oxygen, air, or carbon dioxide and by galvanic action between dissimilar metals. Corrosion inhibitors are widely used, a satisfactory one being about 1,600 g of sodium chromate or dichromate per 1 m^3 of calcium chloride brine or 3,200 g per 1 m^3 of sodium chloride brine.

The properties of sodium and calcium chloride brines are given in Tables 19.1.8 to 19.1.10.

The density of brine is measured by a **salinometer** (or **salometer**), which is a simple hydrometer the indications on which are 4 times greater than on the corresponding Baumé scale (see Sec. 1).

It is undesirable to use a **strength of solution** of salt greater than is necessitated by its freezing temperature, as the specific heat (Table 19.1.9) decreases as the concentration of the brine increases, and consequently the stronger the brine, the less heat a given amount of it is able to convey between certain definite temperatures and the more power is required to pump the brine. Moreover, brine which is too strong may cause clogging of pipes, etc., by depositing salt. On the other hand, if the solution is too weak it may not be able to withstand the temperature existing in the expansion coil, so that a layer of thin ice will form around the latter and interfere with the absorption of heat from the brine. The surface of the expansion coils in the brine tank should be inspected from time to time to see if any ice has formed on them. In larger plants, it is customary to use a solution with a freezing point not less than 10°F (5.5°C) below the lowest temperature which will be obtained in the operation of the plant. In smaller isolated plants and where careful supervision is not ensured, it is customary to make the solution as strong as possible without being unstable, usually 1.240 to 1.250 sp gr.

Ethylene glycol brine is used in various strengths from 15 to 50 percent by weight for refrigeration temperatures down to − 1°F (− 18°C). It is toxic, and therefore its usage is not recommended in food or beverage processing equipment where a leak can contaminate the product. A corrosion inhibitor is recommended to protect carbon steel equipment and piping. Specific heat ranges from 0.975 to 0.76 Btu/lb · °F (4,080 to 3,180 J/kg · K) in typical systems. Specific gravity varies from 1.08 to 1.05.

Propylene glycol brine in 15 to 35 percent weight strength is considered nontoxic and is often used in brewing and other food and beverage applications. The brine is more viscous (10 centipoise at − 6.7°C for 30 percent weight solution) than ethylene glycol. Specific heat varies from 0.9 to 0.975 Btu/lb · °F. Specific gravity ranges from 1.05 to 1.01. Propylene glycol has often replaced ethylene glycol due to the latter's toxicity.

Methylene chloride brine is often used in systems requiring − 20 to − 125°F (− 30 to − 87°C) low temperatures. It has low flammability. Hydrolysis and water contamination must be prevented to avoid equipment corrosion. Specific heat varies from 0.269 to 0.273 Btu/lb · °F (1,126 to 1,142 J/kg · K). Specific gravity ranges from 1.4 to 1.5.

Proprietary chemical blends, such as Dowtherm J and Syltherm, are used where the fluid properties are advantageous. Dowtherm J, a mixture of isomers of an alkylated aromatic compound, is usable between the temperatures of 358°F (181°C) and − 120°F (− 84°C). The usual range of service is between 0°F (− 18°C) and − 70°F (− 21°C). Physical properties vary from 0.412 to 0.390 Btu/lb · °F (0.412 to 0.390 cal/g · °C) for specific heat; 0.894 to 0.926 for specific gravity (ref. 68°F water); 0.079 to 0.084 Btu · ft/h · ft^2 · °F (0.136 to 0.145 W/m · K) for thermal conductivity; and 1.72 to 4.61 cP in the range of usage.

Sodium chloride brine usage is being reduced due to its corrosivity and its relatively high freezing point.

Table 19.1.11 gives a comparison of the performance of the four major brines in a typical shell and tube chiller. This chart does not indicate effects of specific heat, ingredient cost, pumping power requirements, or system investment cost in the selection of brines.

Brine coolers may be of three types: shell-and-tube, shell-and-coil, and double pipe. The shell-and-tube type is the most widely used, the brine flowing through the tubes which are surrounded by the evaporating refrigerant. Tubes may be arranged for multipass operation. The effective heat-transfer surface varies from 8 to 15 ft^2 per ton (0.21 to 0.4 m^2/kW), varying with temperature and brine velocity. A submerged coil in an open brine tank is used for ice making by the can process.

Table 19.1.8 Properties of Calcium Chloride Solutions

(For variation of specific gravity, with temperature, see Table 19.1.9.)

Parts of $CaCl_2$ by weight in 100 parts of the solution	Specific gravity at 60°F	Deg Baumé	Weight per gal, lb	Weight per cu ft, lb	Freezing point, °F	Specific heat at					
						− 4°F	14°F	32°F	50°F	68°F	86°F
6	1.050	7.0	8.76	65.52	28.0	—	—	—	—	—	—
8	1.069	9.33	8.926	66.70	24.2	—	—	0.882	0.887	0.892	0.897
10	1.087	11.57	9.076	67.83	21.4	—	—	0.853	0.858	0.863	0.868
12	1.105	13.78	9.227	68.95	18.2	—	—	0.825	0.831	0.836	0.842
14	1.124	15.96	9.377	70.08	14.4	—	—	0.799	0.805	0.811	0.817
16	1.143	18.12	9.536	71.26	9.9	—	0.768	0.775	0.781	0.787	0.792
18	1.162	20.24	9.703	72.51	4.7	—	0.745	0.752	0.759	0.764	0.769
20	1.182	22.32	9.853	73.63	− 1.0	—	0.723	0.731	0.738	0.744	0.749
22	1.202	24.38	10.04	75.0	− 7.3	0.695	0.704	0.711	0.718	0.724	0.729
24	1.223	26.41	10.21	76.32	− 14.1	0.678	0.686	0.693	0.700	0.706	0.712
26	1.244	28.41	10.38	77.56	− 22.0	0.663	0.670	0.677	0.683	0.690	0.696
28	1.265	30.39	10.56	78.94	− 32.0	0.649	0.656	0.662	0.669	0.675	0.682
30	1.287	32.34	10.75	80.35	− 46.0	0.638	0.643	0.648	0.655	0.661	0.668

Table 19.1.9 Specific Gravities of Brines

(To change to lb/ft^3 multiply by 62.43; to change to lb/gal multiply by 8.35.)

Parts by weight of salt in 100 parts of brine	Sodium chloride				Calcium chloride				Magnesium chloride			
	Temperature, °F											
	14	32	50	68	14	32	50	68	14	32	50	68
6	—	1.046	1.044	1.041	—	1.053	1.051	1.049	—	1.053	1.051	1.049
8	—	1.061	1.059	1.056	—	1.071	1.069	1.066	—	1.070	1.069	1.067
10	—	1.077	1.074	1.071	—	1.090	1.087	1.084	—	1.089	1.087	1.084
12	—	1.093	1.090	1.086	—	1.108	1.106	1.103	1.108	1.107	1.105	1.102
14	—	1.108	1.105	1.101	—	1.127	1.124	1.121	1.127	1.126	1.123	1.121
16	1.128	1.124	1.121	1.116	1.150	1.147	1.144	1.140	1.147	1.145	1.142	1.139
18	1.144	1.140	1.136	1.132	1.170	1.167	1.163	1.159	1.166	1.164	1.161	1.158
20	1.161	1.157	1.152	1.148	1.190	1.187	1.183	1.179	1.186	1.183	1.181	1.178
22	1.178	1.173	1.169	1.164	1.211	1.208	1.203	1.199	1.206	1.203	1.201	1.197
24	1.195	1.190	1.185	1.180	1.233	1.229	1.224	1.219	1.226	1.224	1.221	1.218

Table 19.1.10 Weight of Commercial Calcium Chloride in Brine

Sp gr	1.10	1.12	1.14	1.16	1.18	1.20	1.22	1.24	1.26	1.28	1.30	1.32
Wt per gal, lb	1.41	1.70	2.00	2.30	2.59	2.90	3.20	3.50	3.83	4.13	4.46	4.78
Wt per cu ft, lb	10.55	12.72	14.96	17.20	19.37	21.69	23.94	26.18	28.62	30.89	33.36	35.75

NOTE: Specific gravity is at 60°F for both brine and water. The weights are of 73 to 75 percent solid calcium chloride per gal of brine at 60°F. For flake 77 to 80 percent calcium chloride multiply the weights given by 0.94.

Table 19.1.11 Comparative Performance of Brines†

Brine	Heat-transfer coefficient, U_o*	Fouling factor		Refrigerant no.	Tube material
		Inner	Outer		
Calcium chloride, 24%	53	0.001	0.002	R717	Carbon steel
Propylene glycol, 40%	26.3	0.0005	0.0	HCFC 22	Copper
Ethylene glycol, 40%	44.7	0.0005	0.0	HCFC 22	Copper
Methylene chloride	111.9	0.0	0.0	HCFC 22	Copper

* Values presented in Btu/h · ft² · °F; to convert to W/m² · K, multiply by 5.678.

† Values calculated using ¾-in diameter tubes; integrally finned; 19 fins per inch; 7 ft/s velocity; brine is inside the tube; brine bulk temperature is 5°F.

Table 19.1.12 Capacity of Multipass Shelf-and-Tube Brine Coolers, Flooded

(Tons refrigeration)

Diameter of shell, in	Length of shell, ft	Velocity of brine through cooler, ft/min							
		200				400			
		Total brine, gal/min	Mean temp diff, brine and ammonia, °F			Total brine, gal/min	Mean temp diff, brine and ammonia, °F		
			7½	12½	17½		5	10	15
26	6	140	7.74	12.9	18.1	290	7.15	14.3	21.2
26	9	140	11.5	19.3	26.0	290	10.7	21.4	32.0
26	12	140	15.5	25.8	36.2	290	14.3	28.6	42.9
34	9	230	18.5	30.7	43.1	440	17.1	34.1	51.1
34	12	230	24.6	41.0	57.4	440	22.6	45.3	68
34	18	230	37.0	62.0	86.2	440	34.1	68.2	101
42	12	510	45.0	74.8	105	970	41.5	83	123
42	18	510	67.4	112	157	970	62.2	124	187

The **double-pipe cooler** is usually of 2-in (50-mm) inner or brine-flow pipe and 3-in (75-mm) outer pipe. The commercial rating is 15 to 20 ft (5 to 6 m) length of coil per ton of refrigeration.

The **shell-and-tube cooler** is used with closed heads and is erected both vertically and horizontally; brine flows through the tubes and ammonia is in the shell. It is made in sizes from 1 to 350 tons with ratings of 8 to 15 ft^2 effective surface per ton, varying with the temperature and brine velocities; tubes 1 to 2½ in (25 to 63 mm) arranged multipass. This type of cooler has largely displaced all other types in recent installations (Table 19.1.12).

REFRIGERANT PIPING

(See also Sec. 8.)

It is important to the proper operation of a refrigeration system that the piping or mains interconnecting the compressors, condensers, evaporators, and receivers be properly sized. This piping must be considered in three categories, viz., liquid lines, suction lines, and discharge lines. Essentially pipeline sizing is governed by pressure drop, first cost, operating cost, noise, and oil entrainment. Excessive pressure drops penalize compressor efficiencies and may affect control-valve operation adversely. Liquid-line velocities for most refrigerants are in the order of 60 to 400 ft/min (3.3 to 22 m/s), suction lines from 700 to 4,600 ft/min (38 to 250 m/s), and discharge lines from 1,000 to 5,000 ft/min (55 to 275 m/s). Pressure drops vary approximately as the square of the velocity (or tonnage) and directly as the length of the piping. For liquid lines, when evaporator is located above condenser, the following pressure drops in pounds per square inch per foot (kilopascals per meter) of static lift should be allowed: ammonia, 0.26 (5.9); R 12, 0.57 (12.9); R 22, 0.51 (11.5); R 11, 0.64 (14.5). See also Secs. 3 and 4.)

Ammonia Mains For the average installation, Table 19.1.13 shows the maximum tons of refrigeration normally allowed for various sizes of standard pipe, based on 100 ft (30 m) equivalent length of piping (measured length plus allowance for valves and fittings). The gas pressure drops per 100 ft (30 m) equivalent length upon which Table 19.1.13 is based are as follows: suction lines at 5 lb/in^2 (gage) (34.5 kPa) = 0.25 lb/in^2 (1.7 kPa); suction lines at 20 lb/in^2 (gage) (138 kPa) = 0.50 lb/in^2 (3.4 kPa); suction lines at 45 lb/in^2 (gage) (310 kPa) = 1 lb/in^2 (6.9 kPa); and discharge lines = 1 lb/in^2 (6.9 kPa).

Standard-weight (schedule 40) steel pipe is used for ammonia mains, except for liquid lines 1½ (38 mm) and smaller where extra-strong (schedule 80) pipe is used. Joints may be either screwed, flanged, or welded, but welding is preferred.

R 12 Mains Maximum tons refrigerant for various pressure drops per 100 ft equivalent length of R 12 mains are given in Table 19.1.14. For average conditions, allowable pressure drops in R 12 suction lines are 0.5 to 1.0 lb/in^2 (3.4 to 6.9 kPa) below 0°F (−18°C) evaporator temperature; 1.0 to 1.5 lb/in^2 (6.9 to 10.3 kPa) from 0 to 25°F (−18 to −4°C); and 2.0 to 2.5 lb/in^2 (13.8 to 17.2 kPa) from 25 to 50°F (−4 to 10°C). Discharge-line pressure drops upon which Table 19.1.14 is based are approximately 1 lb/in^2 (6.89 kPa) per 100 ft (30 m) equivalent length. Pressure drops of 1 to 4 lb/in^2 (6.9 to 27.6 kPa) are permissible. Liquid-line pressure drops of 3 to 5 lb/in^2 (20.7 to 34.5 kPa) per 100 ft (30 m) equivalent length are normal, with 10 lb/in^2 (69 kPa) usually considered maximum. To facilitate oil return in vertical suction lines with the evaporator located below the compressor, the velocity in the vertical section should be 1,000 ft/min (38 m/s) or more. Suction-line capacities in Table 19.1.14 may be increased 9 percent for 50°F (10°C) evaporator temperature and will decrease approximately 7 percent for each 10°F (5.6°C) below 40°F (4.4°C).

R 22 piping will, in general, be smaller for the same tonnage than R 12 piping. Suction lines will handle about one-third more tonnage for the same pressure drop. Liquid and discharge lines can be the same as for the equivalent tonnage of R 12 or slightly smaller.

Piping may be standard-weight steel pipe, but copper tubing is used in most cases. Medium-weight type L copper tubing is normally used for land installations, and the heavier type K copper tubing is used for marine work. Joints in the copper tubing may be flared compression fittings for tubing ¾ in (18 mm) OD and smaller. However, hard solder (silver-base alloy melting above 1000°F) (540°C) is preferred for most tubing connections. Copper tubing is almost always used for liquid lines. Steel pipe may be used for the larger suction and discharge lines but should be sandblasted on the inside and carefully cleaned before installation.

Piping flexibility is an important consideration, especially in low-temperature systems [0°F(−18°C) or colder] and where large-diameter or long runs of piping are utilized. The coefficient of thermal expansion is about 9×10^{-6}/°F (16×10^{-6}/°C) for carbon steel pipe. This means a 20-ft (6-m) length will shrink about 0.15 in (3.8 mm) when cooled from 70 to 0°F (21 to −18°C). If the piping configuration were such to exert a moment on a compressor flange, premature equipment failure could result. Long runs of cold brine lines should be analyzed for proper anchoring, guiding, and support to prevent system failure.

Flexibility is provided by a combination of changes in direction and use of properly positioned anchors, spring hangers, and guides. Refer to Sec. 5 for additional information on piping flexibility.

Table 19.1.13 Maximum Tons Refrigeration for Ammonia Mains
(100 ft equivalent pipe length)

Pipe size, in	Suction line			Discharge line	Liquid line	
	Suction pressure, lb/in^2 gage				Condenser to receiver	Receiver to system
	5(−17.2°F)	20(5.5°F)	45(30°F)			
⅜	—	—	—	—	2.5	12.0
½	0.6	1.1	2.0	3.1	6.0	20.0
¾	1.2	2.2	4.1	6.0	14.0	75.0
1	2.2	4.0	7.5	11.4	24.0	137
1¼	4.4	8.0	15.0	22.4	50.0	245
1½	6.4	11.8	21.6	30.9	77.0	400
2	12.1	22.2	42.0	62.0	140	850
2½	19.1	35.5	65.0	97.5	220	1,475
3	31.5	59.0	108	160	375	2,400
3½	46.6	87.5	156	238	540	3,500
4	64.0	118	240	330	740	
5	117	208	385	560	1,320	
6	175	306	600	905	2,030	
8	362	650	1,200	1,810	4,200	
10	640	1,180	2,160	3,200		
12	940	1,850				

SOURCE: From ASRE, Air-Conditioning Refrigerating Data Book, 1955–1956 ed., based on ARI Equipment Standards; see ASHRAE, Handbook, 1994 Refrigeration, for more complete tables.

Table 19.1.14 Maximum Tons Refrigeration for R 12 Lines

Line sizes, in	Suction-line (based on 105°F condensing temp) pressure drop, lb/in² per 100 ft equivalent length at 40°F saturation						Discharge-line condensing temp.		Liquid-line (based on 105°F condensing temp; 25°F evaporating temp) pressure drop, lb/in² per 100 ft equivalent length (type L tubing)			
	½	1	2	3	4	5	115°F	90°F	3	5	10	20
⅜ OD	—	—	—	—	—	—	—	—	0.88	1.14	1.80	2.58
½ OD	0.14	0.20	0.28	0.35	0.41	0.45	—	—	2.89	3.64	5.56	8.50
⅜ IPS	0.17	0.24	0.34	0.42	0.49	0.54						
⅝ OD	0.25	0.35	0.51	0.62	0.73	0.81	1.43	1.15	4.86	6.81	10.2	15.8
½ IPS	0.35	0.45	0.65	0.79	0.93	1.03	1.87	1.50	4.86	6.81	10.2	15.8
⅞ OD	0.55	0.76	1.10	1.34	1.58	1.75	2.97	2.38	10.5	14.1	21.8	33.0
¾ IPS	0.68	0.94	1.35	1.65	1.92	2.12	3.26	2.62	9.73	12.6	18.5	27.0
1⅛ OD	1.26	1.80	2.57	3.17	3.76	4.15	5.05	4.05	21.4	28.2	41.3	60.8
1 IPS	1.43	2.01	2.89	3.54	4.17	4.60	5.29	4.25	21.4	28.2	41.3	60.8
1⅜ OD	2.21	3.12	4.45	5.50	6.38	7.05	7.72	6.19	36.9	48.1	70.5	101
1¼ IPS	2.70	3.82	5.37	6.72	7.68	8.48	9.16	7.35	36.9	48.1	70.5	101
1⅝ OD	3.40	4.78	6.79	8.42	9.77	10.8	10.92	8.75	62.0	80.2	114	160
1½ IPS	4.05	5.75	8.10	10.12	11.6	12.8	12.5	10.0	62.0	80.2	114	160
2⅛ OD	6.12	8.60	12.1	15.1	17.4	19.2	19.2	15.3				
2 IPS	7.66	10.9	15.3	19.2	22.2	24.5	20.6	16.5	124	161	231	328
2⅝ OD	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9				
2½ IPS	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9	230	297	426	607
3⅛ OD	19.1	27.2	38.2	47.8	55.0	60.7	51.5	39.8				
3 IPS	20.9	29.4	42.3	51.8	60.0	66.2	54.5	43.8	364	469	676	972
3⅝ OD	27.8	39.7	55.7	69.8	80.3	88.7	72.0	57.6				
3½ IPS	30.2	43.2	61.0	76.1	87.0	96.0	78.8	62.3	539	704	1,005	1,430
4⅛ OD	38.6	55.2	78.0	97.3	111	123	95.8	77.1				
4 IPS	40.7	58.6	83.0	103	118	130	101.6	81.6	753	972	1,385	1,945
5 IPS	71.3	100	141	176	203	224	171.5	137.8				
6 IPS	126	183	257	322	366	403	266	214				
8 IPS	211	297	422	523	602	664	461	370				
10 IPS	352	503	712	887	1,024	1,130	725	582				
12 IPS	550	780	1,106	1,373	1,582	1,748	1,041	836				

SOURCE: From ASRE, Air-Conditioning Refrigerating Data Book, 1955–1956 ed., based on ARI Equipment Standards; see ASHRAE, Handbook, 1994 Refrigeration, for more complete tables.

Mains of Other Refrigerants For sizing piping for methyl chloride, sulphur dioxide, carbon dioxide, and other refrigerants, reference may be made to the ASHRAE Handbook, 1985 Fundamentals, Chap. 34.

COLD STORAGE

The first consideration in cold-storage room design is the temperature to be maintained in the refrigerated space. The temperature can generally be divided into three ranges: above 32°F (0°C), 32°F (0°C) and below, and 0°F (− 18°C) and below. Methods of construction and degree of insulation are dependent on the maintained temperature. Other more specialized design considerations are humidity control, ventilation requirements, controlled atmosphere (control of O_2, N_2, or CO_2), fire protection, and material-handling facilities (see Table 19.1.15).

Cold-storage rooms can either be site fabricated or prefabricated. The increasing availability of preinsulated panels with low *U* factors has led to increasing use of prefabricated rooms. Many of these are available as complete packages with the refrigeration system included.

Vapor Barrier Although good insulation is important, a well-designed and installed vapor barrier is the key to maintaining satisfactory insulation performance and life. A poor vapor barrier allows moisture to condense in the insulation, greatly reducing its insulating value. A well-designed system is of little value if poorly installed. Poor workmanship, in fact, accounts for most vapor barrier inadequacies.

Common vapor barrier materials include metal foils, polyethylene, asphalted felts and kraft paper, hot-melt asphalt, and special vapor barrier paints. Vapor barriers must be installed on the warm side of the wall.

Table 19.1.15 Cold-Storage Usage and Transmission Factors

	Room volume, ft³									
	20	50	100	500	1,000	2,000	5,000	10,000	50,000	100,000
Usage load*										
Average	4.68	2.28	1.61	1.21	1.10	0.835	0.403	0.240	0.178	0.173
Heavy	5.51	3.55	2.52	1.87	1.67	1.29	0.625	0.408	0.305	0.295
Storage temp, °F	40 and above		25–40		15–25		0–15		− 25–0	
Insulation thickness, in	3		4		5		6		8	
Theoretical *U* factor†	0.086		0.066		0.055		0.046		0.035	
Practical *U* factor‡	0.111		0.083		0.066		0.055		0.046	
Piping *U* factor§	2.5		2.2		2.0		1.8		1.6	

* Btu per cu ft per 24 h per °F temp diff between outside and inside. Usage load includes 10°F or less of product cooling, and infiltration losses, etc., but does not include any freezing of product or fan motor loads when unit coolers are used. Values are for normal conditions and should not be used for unusual product loads.

† Theoretical *U* factor is based on corkboard ($k = 0.30$) with plaster on both sides, Btu per ft² per h per °F temp diff.

‡ Practical *U* factor allows for inefficiency of joints and structural supports in insulation.

§ Piping *U* factor is Btu/ft² outside pipe surface per h per °F temp diff between cooling medium and air (gravity circulation). Allowance has been made for frosting at the lower temperatures.

Table 19.1.16 Product Storage Data

Product	Quick-freeze temp, °F	Storage temp., °F		Humidity, % R.H.	Specific heat		Latent heat	Freezing point	Respiration, Btu per lb per day
		Long	Short		Above freezing	Below freezing			
Apples	−15	30–32	38–42	85–88	0.86	0.45	121.0	28.4	0.75
Asparagus	−30	32	40	85–90	0.95	0.44	134.0	29.8	
Bacon, fresh	—	0–5	36–40	80	0.55	0.31	30.0	25.0	
Bananas	—	56–72	56–72	85–95	0.80	0.42	108.0	28.0	4.18
Beans, green	—	32–34	40–45	85–90	0.92	0.47	128	29.7	3.3
Beans, dried	—	36–40	50–60	70	0.30	0.237	18		
Beef, fresh, fat	−15	30–32	38–42	84	0.60	0.35	79		
Beef, fresh, lean	−15	30–32	38–42	85	0.77	0.40	100		
Beets, topped	—	32–35	45–50	95–98	0.86	0.47	129	31.1	2.0
Blackberries	−15	31–32	42–45	80–85	0.89	0.46	125	28.9	
Broccoli	—	32–35	40–45	90–95	0.92	0.47	130	29.2	
Butter	+15	—	40–45	—	0.64	0.34	15	15.0	
Cabbage	−30	32	45	90–95	0.93	0.47	130	31.2	
Carrots, topped	−30	32	40–45	95–98	0.87	0.45	120	29.6	1.73
Cauliflower	—	32	40–45	85–90	0.93	0.47	132	30.1	
Celery	−30	31–32	45–50	90–95	0.92	0.48	135	29.7	2.27
Cheese	+15	32–38	39–45	—	0.70				
Cherries	—	31–32	40	80–85	0.87	0.45	120	26.0	6.6
Chocolate coatings	—	45–50	—	—	0.3				
Corn, green	—	31–32	45	85–90	0.80	0.43	108	29.0	4.1
Cranberries	—	36–40	40–45	85–90	0.90	0.46	124	27.3	
Cream	—	34	40–45	—	0.88	0.37	84		
Cucumbers	—	45–50	45–50	80–85	0.97	0.47	137	30.5	
Dates, cured	—	28	55–60	50–60	0.83	0.44	104		
Eggs, fresh	−10	30–31	38–45	—	0.76	0.40	98	31.0	
Eggplants	—	45–50	46–50	85–90	0.94	0.47	132	30.4	
Flowers	—	35–40	—	85–90					
Fish, fresh, iced	−15	25	25–30	—	0.82	0.41	105	30.0	
Fish, dried	—	30–40	—	60–70	0.56	0.34	65		
Furs	—	32–34	40–42	40–60					
Furs, to shock	—	15	15						
Grapefruit	—	32	32	85–90	0.91	0.46	126	28.4	0.5
Grapes	—	30–32	35–40	80–85	0.86	0.44	116	27.0	0.5
Ham, fresh	—	28	36–40	80	0.68	0.38	87		
Honey	—	31–32	45–50	—	0.35	0.26	26		
Ice cream	−20	—	0–10	—	0.5–0.8	0.45	96		
Lard	—	32–34	40–45	80	0.52	0.31	90		
Lemons	—	55–58	—	80–85	0.92	0.46	127	28.1	0.4
Lettuce	—	32	45	90–95	0.96	0.48	136	31.2	8.0
Liver, fresh	—	32–34	36–38	83	0.72	0.42	94		
Lobster, boiled	—	25	36–40	—	0.81	0.42	105		
Maple syrup	—	31–32	45	—	0.24	0.215	7.0		
Meat, brined	—	31–32	40–45	—	0.75	0.36	75.0		
Melons	—	34–40	40–45	75–85	0.92	0.35	115	28.5	1.0
Milk	—	34–36	40–45	—	0.92	0.46	124	31.0	
Mushrooms	—	32–35	55–60	80–85	0.93	0.47	130	30.2	
Mutton	—	32–34	34–42	82	0.81	0.39	96	29.0	
Nut meats	—	32–50	35–40	65–75	0.30	0.24	14	20.0	
Oleomargarine	—	34–36	—	—	0.65	0.34	35	15.0	
Onions	—	32	50–60	70–75	0.91	0.46	120	30.1	1.0
Oranges	—	32–34	50	85–90	0.90	0.46	124	27.9	0.7
Oysters	—	—	32–35	—	0.85	0.45	120.0		
Parsnips	−30	32–34	34–40	90–95	0.82	0.45	120.0	28.9	
Peaches, fresh	—	31–32	50	85–90	0.90	0.46	126	29.4	1.0
Pears, fresh	—	29–31	40	85–90	0.86	0.45	118	28.0	6.6
Peas, green	—	32	40–45	85–90	0.80	0.42	108	30.0	
Peas, dried	—	35–40	50–60	—	0.28	0.22	14		
Peppers	—	32	40–45	85–90	0.94	0.47	—	30.1	2.35
Pineapples, ripe	—	40–45	50	85–90	0.88	0.45	122	29.9	
Plums	—	31–32	40–45	80–85	0.88	0.45	123	28.0	

Table 19.1.16 Product Storage Data (*Continued*)

Product	Quick-freeze temp, °F	Storage temp., °F		Humidity, % R.H.	Specific heat		Latent heat	Freezing point	Respiration, Btu per lb per day
		Long	Short		Above freezing	Below freezing			
Pork, fresh	—	30	36–40	85	0.60	0.38	66	28.0	
Potatoes, white	−30	36–50	45–60	85–90	0.77	0.44	105	28.9	0.85
Poultry, dressed	−10	28–30	29–32	—	0.80	0.41	99	27	
Pumpkins	—	50–55	55–60	70–75	0.92	0.47	130	30.2	
Quinces	—	31–32	40–45	80–85	0.90	—	—	28.1	
Raspberries	—	31–32	40–45	80–85	0.89	0.46	125	30.0	3.3
Sardines, canned	—	—	35–40	—	0.76	0.410	101		
Sausage, fresh	—	31–36	36–40	80	0.89				
Sauerkraut	—	33–36	36–38	85	0.91	0.47	128	26	
Squash	—	50–55	55–60	70–75	0.92	0.47	130	29.3	
Spinach	—	32	45–50	85	0.94	0.48	132	30.8	
Strawberries	−15	31–32	42–45	80–85	0.92	0.48	129	30.0	3.3
Tomatoes, ripe	—	40–50	55–70	85–90	0.95	0.48	135	30.4	0.5
Turnips	—	32	40–45	95–98	0.93	0.40	137	30.5	1.0
Veal	−15	28–30	36–40	—	0.71	0.39	91	29	

Insulation In recent years newer insulating materials have largely supplanted the use of plaster and corkboard in cold-storage room construction. Insulation has generally been used in three forms for this application: board form, panelized, and poured, sprayed, or foamed in place.

Board form insulation includes corkboard, rigid polystyrene, polyisocyanurate, and polyurethane, foamed, and fibrous glass.

Panelized insulation has gained acceptance, particularly in prefabricated installations. The outer metal skin can serve as a vapor barrier, and the panels are factory insulated with polystyrene, polyisocyanurate, polyurethane, or fibrous glass. U values as low as 0.035 have been attained in 4-in (100-mm) thick panels.

Poured, sprayed, or foamed in place insulation has its greatest applications on existing installations. It is primarily a urethane material. The application requires special equipment, and precautionary measures must be taken.

Cold Storage Temperatures A great deal of research and experience are required to obtain authoritative information on optimum temperatures and humidities for various products in cold storage. Table 19.1.16 is representative of good practice. The safe storage period depends upon the product and the storage temperature, and operational techniques vary greatly. Modern cold-storage warehouses of the larger concerns are cooled by brine which is furnished at two different temperatures only. The higher temperature for the mild-temperature warehouses is 10 to 12°F (−12 to −11°C), and the low-temperature brine for the freezers is −10 to −12°F (−23 to −24°C). All temperatures above that of the brine are obtained by regulating the amount of brine circulated in any particular set of coils. In the low-temperature warehouses, the piping is arranged for two classes of service: (1) **sharp freezers,** where the goods which are to be frozen are kept in a blast of air whose temperature is −30°F (−34°C) while their temperature is brought down quickly to the holding temperature (say in from 6 to 10 h) after which they are stored in (2) **holding rooms** where the desired temperature is maintained.

The system of cooling which is now being installed in the highest type of warehouses consists of a coil room containing the necessary brine-type or wetted, direct-expansion-type coils, through which the air from the different rooms is circulated by a pressure blower. The inlet and outlet of each room are so arranged that the cooled circulating air will cover the entire room in its transit; this is usually accomplished by having the cold-air inlet in the center of the room and two cold air outlets—one at each end of the room. The piping ratio for the coil rooms in this system, assuming a high-grade insulation with 2 to 4 Btu transmission per ft^2 per 24 h per °F temp diff, should be 1 ft^2 of external pipe surface to 15 ft^3 of space to be cooled [with brine at −10 to −12°F (−23 to −24°C)] for warehouses carrying temperatures of zero and below, and 1 ft^2 of pipe surface to 24 ft^3 of space to be cooled [with brine at 10 to 12°F (−12 to −11°C)] in warehouses carrying mild temperatures of from 30 to 40° F (−1 to 4°C). Another system is used in which the blower and coils are supplemented by coils in the rooms. With this arrangement, it is possible to reduce temperatures quickly and hold them with the coils.

The average coil transmission in cold-storage rooms without forced circulation of the air is about 2 Btu per ft^2 of outside metal surface per h per °F temp diff with horizontal piping and 2.5 Btu with vertical piping. When forced air circulation is used, the transmission rate will increase to 20 Btu or more. In **brine circulation,** the brine, at same compressor back pressure, has a higher temperature than the ammonia, and consequently 1½ times as much pipe or more is used in brine circulation as is direct expansion for a given back pressure.

Piping of Rooms The **size of pipe** usually employed for piping rooms varies from 1 to 2 in (25 to 50 mm) with either brine circulation or direct expansion.

The extra cost of liberal piping allowance will often be offset by the consequent improvement in the efficiency of operation of the compressor. An expansion valve should be provided for every 500-ft (150 m) length of 1-in (25 mm) pipe, every 650 ft (200 m) of 1¼-in (30 mm) pipe, and every 1,000 ft (300 mm) of 2-in (50 mm) pipe when direct expansion is used.

Values of **overall coefficients of heat transfer** in Btu/h · ft^2 · °F for refrigerating practice are given in Tables 19.1.15 and 19.1.17. (See Sec. 4.)

Brine circulation is generally preferred to direct expansion so as to avoid danger from escaping ammonia or other refrigerant in case the pipes should leak. An advantage of the brine system is that there is always a considerable mass of refrigerated brine which can be drawn on in case the machinery should have to be stopped for any reason. In small plants, the general machinery may be stopped at night and only the brine pump be kept going to distribute the surplus refrigeration which has been accumulated in the brine during the day. Brine piping must consist of two lines, a flow and a return, usually of the same size. Brine storage is seldom used in large plants because of its bulk, its first cost, and the practical inability to store much refrigeration.

The **brine coils** in each cooled space are in parallel across the supply and return pipes. It has been common practice to allow 100 to 120 running ft (30 to 36 m) of **pipe** per circuit for low temperatures, and 400 to 440 (120 to 135 m) for high temperatures. The **tons refrigeration produced by brine** at various temperature differences and rates of pumping may be calculated approximately as follows: tons refrigeration = gal/min × °F range/28.

Air Conditioning of Low-Temperature Storage Rooms Bacterial growth and chemical decomposition are retarded by lowering the **tem-**

Table 19.1.17 Overall Coefficients of Heat Transfer
(Btu/ft^2 · h · °F)

Can ice-making piping		Brine coolers	
Old-style feed, non-flooded	12–15	Shell and tube	45–100
Flooded	20–40	Double pipe	150–300
High-velocity raceway trunk coils	80–110	Cooling coils	
Ammonia condensers		Boiling refrigerant to air in unit coolers	4–8
Submerged (obsolete)	30–40	Water to air in unit coolers	5–9
Atmospheric, gas entering at top	60–65	Brine to unagitated air	2.2–2.8
Atmospheric, drip or bleeder	125–200	Direct expansion to unagitated air*	1.6–2.5
Flooded	125–150	Water cooler, shell and coil	15–25
Shell and tube	150–300	Water cooler, shell and tube	50–150
Double pipe	150–250	Water cooler, shell and finned tube (Freon)	30–150
Baudelot coolers, counterflow, atmospheric type		Liquid-ammonia cooler, shell and coil accumulator	45
Milk coolers	75	Air dehydrator	
Cream coolers	60	Shell and coil (brine in coil) 1st coil	5.0
Oil coolers	10	Shell and coil (brine in coil) 2d coil	3.0
Water coolers		Double pipe	6–7
Direct expansion	60–150	Superheat remover, shell and tube	15–25
Flooded	60–200		

* U factor increases to 3.3 at 15°F temp diff with ammonia recirculation systems.
NOTE: Forced circulation of the air increases the coefficient to 1½ to 2½ times the values for still air. One inch of frost decreases the value 25 percent.

perature of perishable goods. Too low a temperature will freeze the goods and may result in spoilage. A holding temperature above the freezing point of the article is best for storage conditions. The maintenance of a proper relative **humidity** in a storage space holding unwrapped goods is as important as the maintenance of the proper temperature. High humidity favors the growth of mold and bacteria. Low humidities rob the product of moisture resulting in losses in value through impaired appearance and lost weight. **Air motion** is of importance in maintaining uniform conditions throughout the storage space. Too high air velocities cause excessive drying, and stagnant air through high humidities will cause mold. An optimum air motion falling between stagnation on one side and excessive drying on the other must be selected.

Low-temperature conditioning for storage purposes is preferably accomplished by the use of **cold-diffuser methods**. The cold diffuser connected to or located in the storage space consists of a fan and cooling means. The cooling means may be either a brine or cold-water spray or a surface-type cooler using brine, or a volatile refrigerant such as ammonia, Freon, etc. Unitary equipment, performing the functions of the cold diffuser, is commercially available. For the requirements for various perishable products, see ASHRAE Handbooks, 1993 Fundamentals and 1995 Applications, 1994 Refrigeration, and Table 19.1.16.

Quick freezing is freezing in 2 h or less; as employed for foods the temperatures may be as low as − 20 to − 50°F (− 29 to − 46°C). The methods of freezing include direct immersion in cold brine or a brine spray with the commodity held in a metal container. Another procedure is the use of a freezing tunnel approximately 50 ft (15 m) long, equipped with a stainless-steel conveyor belt. The commodities to be frozen pass through this tunnel in 15 to 30 min. The speed of the belt is variable, and the heat transfer is accomplished by high-velocity air circulation at a number of points in the tunnel, the circulation being transverse to the belt. The circulating air is cooled by brine sprays or wetted, direct-expansion coils which maintain the required temperature and the high humidity necessary to prevent shrinkage. In another process, packages of the material to be quick-frozen are clamped between steel platens containing evaporating refrigerant. Successful results are obtained with temperatures as high as 0°F (− 18°C). Storage may be at − 5 to − 10°F (− 20 to − 23°C) and transportation at 10°F (− 12°C) or lower.

Cold-storage lockers for individual family use are used in large numbers, particularly in suburban and rural areas. Lockers are rented usually on a yearly basis. A locker plant may have several hundred rental steel lockers, 6-ft^3 (0.17-m^3) capacity each, placed in rooms at about 0°F (− 18°C); a room for cutting and wrapping the meat; and a sharp freezer room held at about − 20°F (− 29°C). Two compressors are frequently specified or a single compressor using automatic control valves on the separate rooms. Ammonia and R 12, operating under direct expansion, are commonly used.

ICE MAKING

Ice refrigeration is economical for some applications. The use of natural ice is fast disappearing. Manufactured ice is made by several methods. In the can system, galvanized cans containing up to 400 lb (180 kg) of water are immersed in brine for 25 to 40 h for freezing. Air dissolved in the water separates as bubbles causing opacity unless the water is agitated during the early stages of cooling, and any dissolved impurities come out of solution to form a central off-color core in the ice block unless the core is removed and replaced with fresh water just before freezing. The time and investment required for making ice in cans are often uneconomical. In the **Flakice** method (Crosby Field, *Trans. ASME,* May 1951, pp. 347–357) refrigerant is sprayed against the inner wall of a flexible, slowly rotating cylinder which is partially immersed in a tank of water. As the cylinder rotates, a thin layer of ice forms on the outside and then is broken off as further rotation causes the cylinder surface to flex under the action of an internal cam. For a patent history of small ice machines of this type, refer to Crosby Field, *Refrig. Eng.,* **58,** Dec., 1950, p. 1163. In the **PakIce** process (Taylor, *Refrig. Eng.,* **22,** 1931, p. 307) liquid ammonia is evaporated in an annular space formed by two concentric cylinders, the inner being corrugated. Water sprayed on the inner corrugated surface quickly freezes in a thin layer about .25 mm thick and is continually removed by mechanical scrapers. In an **extrusion method** (Watt, *Trans. ASME,* 1949) a slightly tapered circular or rectangular vertical cylinder with large end up is open at the top, but closed by a ram at the bottom. This cylinder has a jacket in which refrigerant is evaporated directly against the cylinder walls. Water enters the cylinder and freezes to a tapered block at the start of the operation. As soon as this occurs, the ram lifts (about 6 mm), shearing the ice block from the cold cylinder walls. Water admitted to this space quickly freezes, the ram again operates and the block of ice is again lifted, and shearing occurs again between the freshly formed ice and the cylinder walls, but not at the fresh ice and old ice interface. In this way, a continuous cylinder of ice is formed by the series of nested ice shells. The installation and upkeep costs are small and so is the space required per ton of the ice produced; a pilot-plant 1-ton (907 kg) unit requires 3 ft^2 (0.3 m^2) of floor space.

Batch ice makers can manufacture up to 75 tons (68 tonnes) of ice per day. These machines use constant-diameter ice-forming tubes to produce ice in shell and toroidal forms. The ice forms on a bank of vertical tubes in either of two configurations: inside the tubes for pellets or outside of the tubes for shell ice. A water distribution system is provided at the top. An ice conveyor and mechanical size-reducing device is mounted below the tubes. The refrigerant is circulated through the annulus of concentric tubes when ice pellets are made, and inside of single tubes when shell ice is made. The refrigerant supply can be designed for flooded operation; however, equipment capacity is reduced. The cycle starts as water passes through the distributors and flows over the surface of the tubes as a film. The refrigerant freezes the water to form a thickening layer of ice as time passes. Unfrozen water drains out of the tube into a sump. Then the water is recirculated, by a pump, up to the distribution system at the top of the tube bank. A purge of water is taken from the sump to remove accumulated salts and other impurities. When harvest is initiated, the refrigerant system is drained, and hot gas from the condenser is introduced into the refrigerant side of the tube and the pressure is raised. This causes the ice to partially melt and release from the tube wall. The cylinder or shell of ice falls by gravity into the conveying and size-reducing machinery. The size of an ice pellet can vary from 0.75 in (19 mm) to 1.375 in (35 mm) outside diameter. Ice clarity can be controlled by flushing the ice with water before and during harvest. The machine is controlled by digital logic microprocessors. Energy consumption when using 65°F (18°C) feedwater varies from 1.94 to 2.33 tons (6.8 to 8.2 kW) refrigeration per ton (909 kg) of ice when using R-717 versus HCFC-22 in a recirculation system. Standard model ice makers can be operated with the refrigerant system either in the flooded condition or with circulating pumped refrigerant in the tubes. Similar units lose 18 percent capacity when operating in a flooded state compared to operating with pumped recirculating refrigerant.

Making clear ice using continuous methods is often accomplished by washing the newly frozen ice surface with cold water. This method can increase water requirements by up to 50 percent and energy usage by 15 percent.

The refrigeration tonnage required for making 1 ton (909 kg) of ice varies from 1.84 with 55°F (12.8°C) inlet water temperature to about 2.0 if the water to be frozen enters as high as 75°F (23.9°C).

SKATING RINKS

Rinks for ice skating, hockey, or curling vary in size from about 400 ft^2 (37 m^2) area to 16,000 ft^2 (1,486 m^2) or more. Construction of the ice floor is important. For a low-cost, efficient unit, brine pipes are laid directly in sand, which is kept wet during freezing to increase heat transfer. As freezing progresses, a layer of ice is built up by spraying water on the surface. The expense of maintaining such a floor in a satisfactory level condition may be considerable, and corrosion of the pipes, from the outside in, in contact with the wet sand may be excessive, particularly during shutdown periods. Moreover, it is not possible to use such a floor for other purposes. When a floor for diversified activities is needed, it is usual to provide a lower layer of insulation, primarily to prevent sweating on ceilings under the floor, surmounted by a concrete slab in which the steel pipes carrying the brine are imbedded and on which the ice layer is formed. This construction protects the pipe from external corrosion, affords fast freezing, and makes possible use of the floor in a few hours for purposes where an ice surface is not desired. Floor pipes can be mild steel or polyethylene and are usually 1 or 1¼ in (25 or 32 mm) spaced 3 to 6 in (75 to 150 mm) on centers; care must be exercised to ensure a uniform distribution of brine. Brine flow may be 10 gal/min per ton of refrigeration, corresponding to about 1 gal/min per 10 to 20 ft^2 of piping surface. In a few installations, cold brine is sprayed directly against the under side of a steel floor on which the ice layer is frozen. Close control of ice surface temperature is necessary for good skating, usually −3°C ± 0.5. Refrigeration capacity should be from 0.4 to 0.85 ton per 100 ft^2 (150 to 320 W per m^2) of floor surface, although the load may be much higher for open-air rinks or other unusual conditions.

REFRIGERATION IN THE CHEMICAL INDUSTRIES

The use of refrigeration in petroleum and other industries is widespread. In petroleum processing, refrigeration is used (1) to control vapor pressure of highly volatile constituents (methane, ethane, propane, and butane), such control being necessary during distillation, during processing, and for recovery of gasoline fractions from natural gas; (2) to shift the solubility relationship so that undesired constituents such as asphalt and wax in lubricating oils may be removed by precipitation; (3) to produce selective chemical reaction such as occurs when sulphuric acid is used to remove gum-forming constituents from light fuels or when an alkylate fuel fraction is formed by combining a low molecular weight unsaturated with a similar saturated hydrocarbon. Alkylate is a valuable and important constituent of aviation fuels.

For (1), temperature ranging from −35°F (−38°C) or lower to as high as 60°F (15°C) are required, the refrigerant being ammonia, propane, or butane. For the higher temperatures spray-cooled water or jet refrigerating units (see above) are sometimes satisfactory. For (2), it is possible to use propane as a combined solvent and refrigerant, evaporation of the propane causing the required cooling, temperature level [approximately −40°F (−40°C)] being controlled by the pressure held on the equipment. The propane vapor is recompressed, condensed, and reused as in any compression cycle. As much as ⅓ ton of refrigeration per barrel of lubricating oil dewaxed is needed. For (3), temperatures of up to 60°F (−16 to 15°C) are general, and ammonia is a satisfactory and widely used refrigerant.

The chemical industry uses refrigeration to control condensation polymerization processes in the manufacture of synthetic rubbers and plastics. Syntheses of refrigerants and other chemicals require systems to maintain temperatures which range from 5°C down to −85°C in process equipment. Large centralized refrigeration systems which use ethylene glycol and methylene chloride brine are common. Several modern chemical processes use the eventual product as a refrigerant (for example, the manufacture of ammonia). Cascade systems with segregated circuits are used to produce temperatures below −50°C.

Recently imposed regulations limit volatile organic vapor emissions to the atmosphere from storage tank vents and other process systems. The recovery efficiency required of cleanup systems has led to development of refrigerated liquid absorption processes to scrub vaporous effluents from organic chemical storage tanks. Recently installed systems have required liquid absorbent temperatures of −40°F (−40°C). The capacity of these systems varies from 2 tons (7 kW) to 500 tons (1760 kW). The absorbent liquid is often accumulated from the effluent stream over a period of time.

DEEP REFRIGERATION

Dry ice (solid CO_2) is useful as a refrigerating medium in special cases. The main objection to the general use of dry ice is its cost, but the ease in its handling, its low temperature [−110°F (−79°C) at atmospheric pressure], its noncorrosive and nontoxic properties, its high latent heat, and the absence of liquid drip make it desirable. The heat absorbed per pound of solid CO_2 during sublimation at atmospheric pressure and −110°F (−70°C) is 245 Btu approx. The specific heat of the gas at constant pressure is about 0.2.

Carbon dioxide is obtained commercially either by fermentation or by burning. The gas is compressed, usually in three stages, and cooled to atmospheric temperature, thereby forming a liquid at about 1,000 psia (690 kPa); the liquid is then throttled to below 5.3 atm pressure. During throttling part of the CO_2 solidifies and is compressed to form a dense ice. The fraction that returns to the vapor phase is recompressed.

Production of Solid Carbon Dioxide Figure 19.1.18 shows a simple cycle and Fig. 19.1.19 the elemental pressure-enthalpy diagram for the production of solid carbon dioxide. From the high pressure at atmospheric temperature the liquid is throttled (isenthalpic) to atmospheric pressure, the resultant point being within the sublimation solid-vapor zone below the triple-point realm. In this resultant throttling expansion, part of the CO_2 solidifies; it is then removed in the separator from the residual vapor and is compacted to form a dense CO_2 ice. The fraction representing the vapor phase is augmented by fresh makeup gas to continue the production cycle.

Beyond the manufacture of dry ice, operation at deep refrigerations,

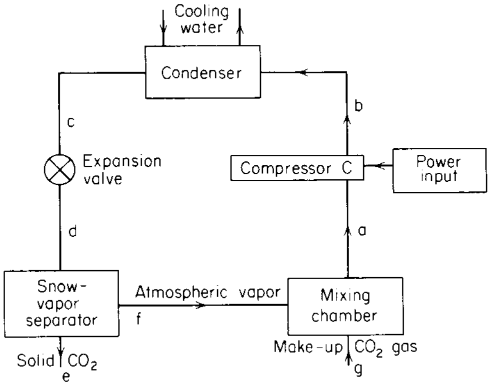

Fig. 19.1.18 Elemental dry-ice production circuit.

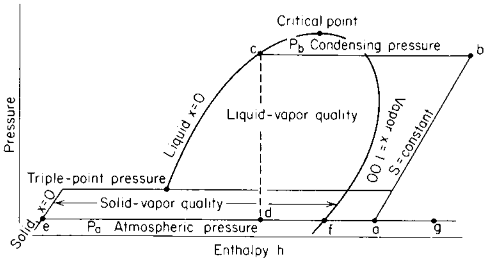

Fig. 19.1.19 Pressure-enthalpy diagram illustrating dry-ice production circuit.

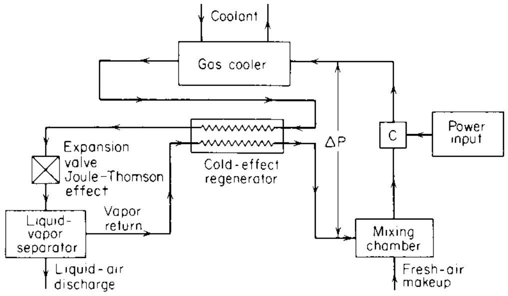

Fig. 19.1.20 Elemental Linde system circuit.

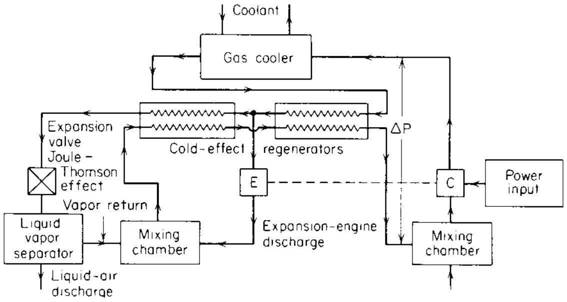

Fig. 19.1.21 Elemental Claude system circuit.

such as air liquefaction [at − 312°F (− 191°C) approx], lead into **cryogenics** (see Sec. 19). Illustrative systems are (1) Linde (throttling) and (2) Claude (expander). In the elemental **Linde system** (Fig. 19.1.20), by virtue of the ‘‘cold-effect’’ regenerator, the process air is reduced in temperature by the return fraction of the air from the separator. The throttling process thus ends in the ‘‘wet zone’’ at atmospheric pressure with generation of low-temperature liquid air that flows from the separator. In the **Claude system** (Fig. 19.1.21), use is made of an expander (whose work output reduces the net work requirement of the compressor) and of an expansion valve with the Joule-Thomson effect.

19.2 CRYOGENICS

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Notation

Some symbols in this section differ from the notation of engineering practice but are retained to facilitate use of the prevalent scientific sources of information.

A = area, ft^2 (m^2)
b.c.c. = body centered cubic lattice structure
C = specific heat, Btu/lbm · °R(J/kg · K)
C_p, C_v = specific heats at constant pressure and volume respectively, Btu/lbm · °R(J/kg · K)
f.c.c. = face centered cubic lattic structure
H = enthalpy, Btu/lbm (J/kg)
H = magnetic field, $G(T)$
h.c.p. = hexagonal close-packed lattice structure
κ = thermal conductivity, Btu/h · ft · °R (W/m · K)
σ = electrical conductivity (reciprocal of resistivity) or pair separation for zero potential in Lenard-Jones equation
s = Boltzmann’s constant
L = length, ft (m)
LH_2 = liquid hydrogen (H_2)
LHe = liquid helium (He)
LN_2 = liquid nitrogen (N_2)
LNG = liquefied natural gas
LO_2 = liquid oxygen
n = number of moles

P = pressure, psia or torr (vacuum) (Pa)
R_e = electrical resistance, Ω
T = temperature, °R (K)
V = volume, ft^3 (m^3)
α = thermal linear-expansion coefficient, °R^{-1} (K^{-1}), or accommodation coefficient
η = viscosity, lbm/ft · h (N · s/m^2)
ρ = density, lbm/ft^3 (kg/m^3)
ΔT = temp difference, °R(K)
* = Denotes property divided by quantum parameter (Table 19.2.5) which is not the critical point value.

Cryogenics is the study, production, and utilization of low temperatures. Cryogenic temperatures have been defined so ambiguously that ''upper limits'' to the cryogenic range from 216 to 396° R (120 K to 220 K) may be found in the literature. In this section the cryogenic range for a property of a given substance is considered to embrace the scale between absolute zero and the temperature above which the property has the expected or normal behavior. Cryogenics thus embraces the unusual and unexpected variations which appear at low temperatures and make extrapolations of properties from ambient to low temperatures unreliable.

Progressively lower temperatures become increasingly difficult to attain in practice. As the working temperature of a refrigerator is lowered, the work required to transfer a given amount of heat increases as demonstrated by the Carnot limitation, to wit, $W = Q[(T_1 - T_2)/T_2]$, where W is the work required to extract the heat Q at a low temperature T_2 and reject it at a higher temperature T_1 (see Sec. 4). The actual work is always greater than this because of inefficiencies of mechanical equipment, thermal losses associated with finite temperature differences in heat exchangers, and heat leaks from the surroundings to the cold equipment.

REFRIGERATION METHODS

(See also Sec. 4 and Sec. 19.1, ''Mechanical Refrigeration.'')

Cryogenic refrigerators (cryocoolers) may be classified by (1) the functions they perform (e.g., the delivery of liquid cryogens, the separation of mixtures of gases, and the maintenance of spaces at cryogenic temperatures, (2) their refrigerating capacities and (3) the temperatures they reach. Large industrial-sized plants (*a*) deliver LNG (~ 120 K), LO_2 (~ 83 K), LN_2 (~ 77 K), LH_2 (~ 20 K), and LHe (~ 4 K), (*b*) separate gaseous mixtures, e.g. the constituents of the atmosphere, H_2 from petroleum refinery gases, H_2 and CO from coke oven and coal-water gas reactors, and He from natural gas, and (*c*) provide refrigeration to maintain spaces at low temperatures. For the latter, i.e., (*c*), a unit has been manufactured and installed that delivers 13 kW of refrigeration at less than 3.8 K.

An important area of refrigerator development that is being commercially exploited now is laboratory-sized cryogenic refrigerators for laboratory research and development at LHe temperatures (< 5 K). These refrigerators are used for many different purposes, refrigerating for example: high-field superconducting electromagnets, LHe bubble chambers for high-energy (nuclear) particle research, experimental superconducting electric generators and motors and superconducting magnets for levitation of railroad trains. Another area of commercial exploitation that is important for the progress of cryogenic physics research is the development of refrigerators for the continuous production of refrigeration at temperatures below 1 K. These include L^{3}He evaporation refrigerators and L^{3}He-L^{4}He dilution refrigerators that reach temperatures of 0.4 and 0.003 K, respectively.

Of various methods of refrigeration, the most commonly used to produce temperatures as low as 1 K are (1) the evaporation of a volatile liquid (referred to as the **cascade** method when applied in several successive stages using progressively lower-boiling liquids), (2) Joule-Thomson (isenthalpic) expansion of a compressed gas, and (3) an adiabatic (isentropic) expansion of a compressed gas in an engine (reciprocating or turbine) or from a bomb or cylinder through a throttling valve (Simon expansion). Using L^{3}He, method (1) is capable of reaching temperatures as low as ~ 0.4 K. Numerous refrigeration cycles have been devised utilizing various combinations of the above three methods. The final stage of refrigeration in plants that deliver liquid cryogens is generally a Joule-Thomson (isenthalpic) expansion. **Expansion engines** (reciprocating and turbine) are commonly used for producing the refrigeration needed to reach the final stage. Turbine expanders are preferred for the large refrigerators because their efficiencies (70 to 85 percent) are higher than for reciprocating expanders. The efficiencies of the heat exchangers for counterflowing ''cold'' and ''warm'' gases are an important factor in the overall refrigerator efficiency. In industrial plants heat exchanger efficiencies reach 98 percent. Their design represents a compromise between the attainment of a high heat-transfer coefficient, on the one hand, and a low resistance to the flow on the other.

Figure 19.2.1 typifies a commonly employed modified Brayton cycle which uses an expansion engine for generating the refrigeration. The theoretical figure of merit (the ratio of the work W_c done by the compressor to the heat Q, transferred to the refrigerant at the low temperature) is for this cycle

$$\frac{W_c}{Q} = \frac{RT_1 \ln (P_1/P_5)}{M(H_4 - H_3)}$$

where M is the molecular weight of the gas whose enthalpy per unit mass is H. The other symbols are defined by reference in Fig. 19.2.1.

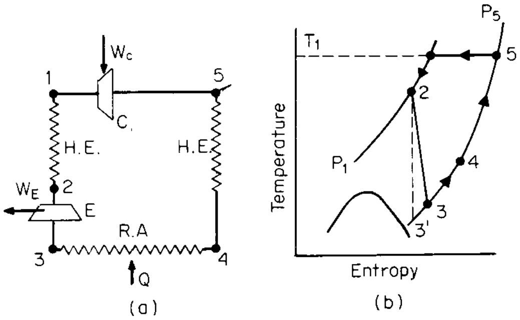

Fig. 19.2.1 (*a*) Schematic of a modified (isothermal compression) Brayton cycle, and (*b*) the process path. W_c is the energy (work) to drive the compressor *C*. *Q* is the heat absorbed by the refrigerator from the refrigerated area R.A., where the working fluid passes from state 3 to state 4. The heat-exchanger (H.E.) processes are isobaric. The dashed line 2-3′ is the ideal isentropic expansion; 2-3 represents the actual expansion in the expander E.

The **Stirling refrigerating engine** and modifications of it are used in a number of commercial makes of laboratory and miniature-sized refrigerators. The Stirling-cycle refrigerator is well suited to (1) the liquefaction of air on a laboratory scale (~ 7 l/h), (2) the recondensation of evaporated liquid cryogens, and (3) the refrigeration of closed spaces where the refrigeration load is not very large. These laboratory scale refrigerators supply ~ 1 kW of refrigeration at ~ 80 K and ~ 2 kW at 160 K. They are used with liquid air fractionating columns for the production of LN_2 (~ 7 l/h) and LO_2 (~ 5 l/h).

The Stirling-cycle refrigerator consists of a piston for compressing isothermally the working fluid (usually He), and a displacer which can operate in the same cylinder with the piston. The displacer and piston are connected to the same electrically driven shaft but displaced in phase by 90°. The displacer pushes compressed gas isochorically from the warm region where it was compressed, through the regenerator into the cold region where the compressed gas is expanded isothermally doing work on the piston and producing refrigeration. The regenerator consists of a porous mass (packed metal wool of high heat capacity) in which a steep temperature gradient is established between the warm region of compression and the cold region of expansion. The displacer returns the gas after expansion to the region for compression through the regenerator in which it is warmed to the temperature of the isothermal compression. The refrigerant (He) is recycled. The regenerators in the larger refrigerators are usually placed around the outside of the engine cylinders, but in small capacity, lower temperature refrigerators they are put inside the displacers.

In Stirling refrigerators that reach temperatures of 17 K and produce 1 W of refrigeration at 25 K, the compressed He is expanded in two or three stages at temperatures intermediate between room temperature and the lowest temperature reached. There is only a single piston for compression. The expansion in stages allows part of the heat that leaks into the coldest region of a single stage engine to be absorbed and removed at a higher (intermediate) temperature where the efficiency for transferring heat is greater.

The **Vuilleumier refrigerating engine** is a modification of the Stirling refrigerator. It *resembles* two single Stirling engines placed back to back. One of the engines operates as a heat engine and the other as a refrigerator. The lower temperature at which the heat engine discharges its waste heat, is also the top temperature at which the refrigerator engine discharges its waste heat. Hence, the Vuilleumier refrigerator operates at elevated, intermediate and low temperature levels which may be 800, 300 (ambient), and 90 K, respectively. Each engine has a cylinder, a displacer, and a regenerator, and the two engines are connected to a common crankshaft but displaced by 90°. Very little external power is needed to operate the two displacers which are operated in sinusoidal motion, displaced 90° in phase. The working fluid (He) is recycled.

The **Gifford-McMahon refrigerator** is another modification of the Stirling-cycle refrigerator. It may be single stage for refrigeration at higher temperatures (~ 80 K), or multistage for lower temperatures. Each stage has a cylinder, a displacer and a regenerator. The displacer pushes gas (usually He) under a very small head of pressure from the top of the cylinder, where the temperature is ambient, through the regenerator, into the bottom of the cylinder which is the cold region of the refrigerator. Compressed He is supplied from an external source—the refrigerator has no piston for compressing gas. Only a small source of external energy is needed for driving the displacer which has very little work to do in overcoming the forces of friction. The valves that control the admission of compressed He to the top of the cylinder, and the expansion of the cold compressed He from the bottom of the cylinder are externally operated by a drive mechanism. The cycling rate is < 2 Hz. The expansion of the cold He is of the adiabatic-Simon-bomb type. Expanded gas is discharged from the refrigerator.

Some applications of cryogenic refrigerators require very high reliability without maintenance. The fewer the working parts, the more likely is the needed reliability to be achieved. The **pulse-tube** and **thermoacoustic refrigerators** are promising candidates in this respect in that they have only one moving part, and that part operates at ambient temperature. Pressure oscillations are introduced into a tube with a closed end and fitted with the proper arrangement of heat exchangers. For the optimum frequency of the pulsation, a temperature gradient is established between the heat exchangers, and this gradient can pump heat from a low temperature to a higher one. A number of different designs have resulted, including single-stage and multistage units, and operating frequencies have ranged from about 1 Hz up to acoustic frequencies. This type of refrigerator, still in the research stage, is capable of reaching low temperatures down to 28 K with a single-stage unit, 26 K with a two-stage unit, and 11.5 K with a three-stage unit, all with ambient temperature heat sinks. Present research is also focused on the use of an acoustically driven pulse-tube refrigerator that would have no moving parts and no close tolerances of the components, thus making it inexpensive and easy to produce.

Magnetic refrigeration was originally proposed in 1925 but has yet to find acceptance for commercial exploitation. Magnetic refrigeration depends upon the ability of paramagnetic materials at low temperature (or ferromagnetic materials near their Curie temperatures) to warm up (or expel heat at constant temperature) with the application of a magnetic field. Conversely, these materials will cool (or absorb heat at constant temperature) upon removal or reduction of the magnetic field. With the use of a paramagnetic material magnetized at a higher temperature of about 1.5 K, temperatures of about 0.001 K have been reached by a single demagnetization. Although a continuously operating paramagnetic salt refrigerator was marketed in the late 1950s, commercial magnetic refrigerators are not now available. However, magnetic refrigeration is applicable over a wide range of temperatures, and recent research work is leading to continuous application of this process by mechanisms that promise a considerable increase in efficiency over that of more conventional refrigerators.

^{3}He-^{4}He dilution refrigerators for reaching temperatures between 0.01 and 0.3 K and generating continuously as much as 750 ergs/s of refrigeration are commercially available. Refrigeration results from the solution of L^{3}He in L^{4}He, the heat of solution being negative. Figure 19.2.2 is a schematic diagram of a dilution refrigerator. ^{3}He vapor from the ‘‘still’’ at ~ 1 K (the upper operating temperature of the refrigerator) is collected and returned by a pump to the ‘‘mixing’’ chamber where solution takes place. The ^{3}He arrives at the mixing chamber as L^{3}He near the mixing chamber temperature (the lower operating temperature of the refrigerator) after flowing through a heat exchanger counter to the flow of cold ^{3}He-^{4}He solution on its way to the ‘‘still.’’ In the electrically heated still, ^{3}He is evaporated from the solution. The evaporated ^{3}He is withdrawn by the collecting pump (diffusion) that returns the ^{3}He to the mixing chamber. At 1 K the vapor pressure of L^{4}He is negligible. L^{4}He in the still, freed of ^{3}He, returns to the mixing chamber by superfluid flow, building up in the mixing chamber an osmotic pressure that drives ^{3}He atoms towards the still against the forces of viscous flow.

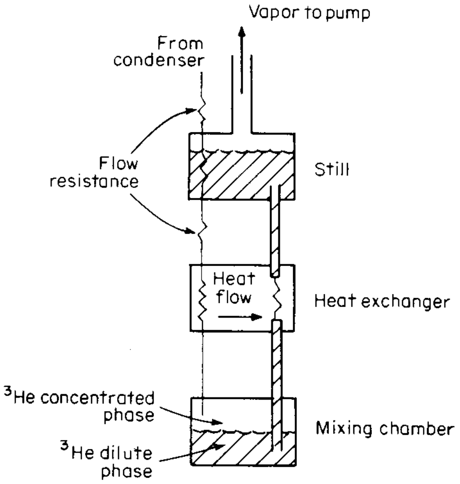

Fig. 19.2.2 Schematic diagram of the mixing chamber and still of a ^{3}He-^{4}He dilution refrigerator.

In the mixing chamber there is a two-phase separation of ^{3}He and ^{4}He. A layer of nearly pure L^{3}He of lower density rides on a heavier L^{4}He-rich layer containing ~ 6 percent ^{3}He. At temperatures lower than 0.87 K liquid solutions of ^{3}He and ^{4}He separate into two liquid phases, one ^{3}He-rich and the other ^{4}He-rich, in thermodynamic equilibrium. At 0 K, the ^{3}He-rich phase is 100 percent ^{3}He, whereas the ^{4}He-rich phase (in equilibrium with L^{3}He phase) contains ~ 6 percent ^{3}He.

In the mixing chamber, L^{3}He enters the upper ^{3}He-rich phase; solution of ^{3}He in ^{4}He takes place at the interphase boundary. The lower (denser) ^{4}He-rich phase connects with the still.

Starting at 50 mK (a temperature reached in a ^{3}He-^{4}He dilution refrigerator), temperatures in the 2- to 3-mK range are attainable with a **Pomeranchuk refrigerator**. Refrigeration is generated by compression of a mixture of liquid and solid phases of ^{3}He at pressures in excess of 28.9 atm, the minimum P at which these phases can coexist in equilibrium. The entropy of solid ^{3}He exceeds the entropy of L^{3}He, which is contrary to the normal behavior for other substances. This occurs in ^{3}He because of its nuclear magnetic properties. The ^{3}He nucleus is magnetic and at temperatures in the mK range and above, in solid ^{3}He, the nuclear moments are randomly oriented whereas in L^{3}He, at temperatures lower than ~0.3 K, the nuclear moments are paired or partially paired, antiferromagnetically. An adiabatic increase of *P* converts liquid to solid with a reduction in *T*, whereas an isothermal increase of *P* results in an absorption of heat. ^{3}He, condensed, is confined in a container with flexible metal walls in the ''cold'' region of a ^{3}He-^{4}He dilution refrigerator. The ^{3}He container is surrounded with L^{4}He which serves as the pressure transmitting fluid.

GAS LIQUEFACTION

A conventional method of gas liquefaction utilizes a gas compressor, a countercurrent heat exchanger, and a throttling valve through which the gas expands isenthalpically (Joule-Thomson) and cools if it has already been cooled below its inversion temperature. After expansion, the cold gas returns through the heat exchanger, in which it exchanges heat with the countercurrent compressed gas flowing to the throttling valve. It leaves the heat exchanger near the temperature of the entering compressed gas. The temperature decreases at the throttling valve until the condensing temperature is reached, and then liquefaction occurs at a rate determined by the rate of refrigeration.

The rate of liquefaction *x* in pounds per hour is $x = \{[H$ (expanded *gas* out) $- H$ (compressed *gas* in)$]\, w - q\}/[H$ (compressed *gas* in) $- H$ (*liquid* following valve)], where the *H*'s are enthalpies in Btu per pound for the final heat exchanger, *w* is the flow rate in the same units as *x*, and *q* is the heat leak in Btu per hour from outside into the heat exchanger. For an ideal heat exchanger the expanded gas leaves at the temperature of the entering compressed gas. In practice, there is a small difference in temperature which represents a small loss of refrigeration. For ideal gases, *H* is independent of *P* for a given *T*, and hence no liquefaction of an ideal gas results. The *H*'s of air, O_2, N_2, A, CO_2, the hydrocarbons, and the normal refrigerants decrease with increasing pressure at ambient *T*, except at very high *P*'s, and these gases are liquefiable by this kind of isenthalpic (Joule-Thomson) expansion. The *H*'s of H_2 and He at ambient *T*, however, increase with increasing *P*, even at low *P*'s, and hence an auxiliary mechanism is required for the liquefaction of these gases. In one method, the stream of compressed gas is split in the liquefier, and a part goes to an engine in which it is cooled by an isentropic expansion with the performance of work. This part of the flow, thus cooled, is sent to a heat exchanger, where it flows counter to the other fraction of compressed gas, cooling it to a temperature below the inversion temperature (see Sec. 4). This flow of compressed gas, thus precooled, is run through another and final heat exchanger with a throttling valve at its lower end, with the result that the quantity *x* is liquefied.

Another method of precooling H_2 and He below their inversion temperatures makes use of a boiling liquid cryogen through which the compressed gas flows in a heat exchanger. LN_2 is used for precooling H_2, and LH_2 for He.

APPLICATIONS OF CRYOGENICS

Gases, such as O_2, N_2, natural gas, H_2, and He, are liquefied for transportation because of the gain in density at low pressure. Shipments have been made of LH_2 by trailer trucks in quantities of 13×10^3 gal (50 m^3) for thousands of miles and by railway tank cars holding as much as 28×10^3 gal (100 m^3). LHe is regularly shipped in quantities from 25 gal (0.1 m^3) to 10,000 gal (40 m^3). Liquid cryogens are used as liquid refrigerants. The aerospace, steel, and bottled-gas industries are the principal users of liquefied cryogens. Important applications as coolants can be found in vacuum technology, electronics, biology, medicine, and metal forming.

In recent years, large superconducting magnets have been undergoing development for a variety of applications including fusion reactors, energy storage, electric-power transmission, high-energy physics experiments, and magnetically levitated trains. Magnets of Nb-Ti are available with fields up to 80 kG (8 T) with large magnetic field volumes (1 m^3). Higher field magnets [up to 150 kG (15 T)] of Nb_3Sn are available with smaller magnetic field volumes (10^{-4} m^3). The Los Alamos National Laboratory installed a 30-MJ superconducting magnetic energy-storage system in the Bonneville Power Administration's Tacoma, Washington, substation to provide stabilization of utility long-line power-transmission systems. The response characteristics of the unit have been demonstrated on the utility's power grid, and the unit has been run successfully for extended periods. Two 100-m long superconducting ac transmission cables have been successfully tested at Brookhaven National Laboratory at 4,100 A and the equivalent of 138 kV three-phase. At the other end of the size spectrum, Josephson junction devices are being considered for applications such as radiation detectors, highly precise and accurate electrical measurements, geophysical instrumentation, medical instrumentation, and computer memory cells.

Cryogenics is also becoming increasingly important in the storage and transport of energy. Large quantities of natural gas are routinely transported as a liquid (LNG). Peak shaving in municipal gas systems is accomplished by storage of LNG. If, due to depletion of fossil fuels, hydrogen becomes an important synthetic portable fuel, cryogenic hydrogen will play an important role as it already does in space applications including the shuttle. Another potential application is aircraft fuel. This may be extended to surface transportation applications. Slush hydrogen, a mixture of liquid and solid hydrogen, is being considered as a fuel for the National Aerospace Plane.

PROPERTIES OF SOLIDS AT LOW TEMPERATURES

(See also Sec. 4.)

Specific heats of solids in general decrease with decreasing *T*, becoming zero at 0°R (Fig. 19.2.3). The downward approach to $C_p = 0$ at 0° R is interrupted for some substances (principally compounds) by ''bumps'' on the curve (excess C_p that rises to a maximum and then decreases). Paramagnetic salts undergoing transitions to either a ferromagnetic or an antiferromagnetic state are examples. This excess specific heat of a paramagnetic salt is connected with the effectiveness of the salt for reaching low temperatures by the method of adiabatic demagnetization. There are also transitions in solids from a more orderly to a less orderly arrangement of atoms and molecules in the lattice that give rise to excess C_p. The transition in solid ortho- and normal H_2 below 20°R (11 K) is an example. In Fig. 19.2.3, C_p's are plotted for various materials.

Heat is transferred in dielectrics by lattice vibrations, or waves. In

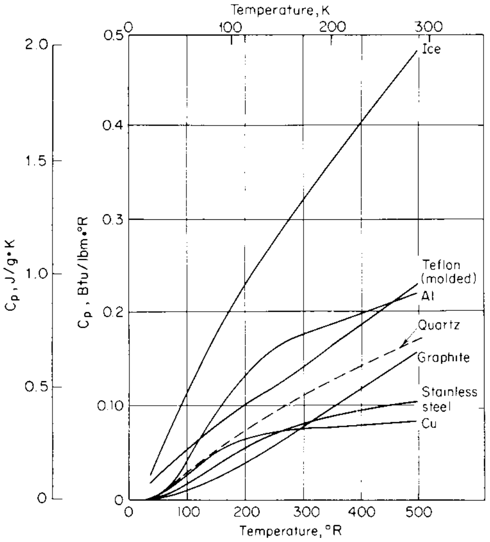

Fig. 19.2.3 Specific heats of solids at low temperatures.

good electrical conductors, heat is transferred principally by the conduction electrons, and thermal and electrical conductivities are related by the Wiedemann-Franz law; $\kappa/\sigma T$ = const. This means that the ratio of the thermal and electrical conductivities at a given T is approximately the same for the good conductors. In the poor electrical conductors, alloys for example, both lattice waves and conduction electrons play important parts in the transfer of heat. The κ's of ''pure'' dielectrics and ''pure'' metals rise with increasing T from $\kappa = 0$ at 0°R (proportional to T for metals and to T^3 for dielectrics), reach a maximum, normally between 10 and 100°R (5 and 55 K), and then decrease to a value approximately independent of T (Fig. 19.2.5 and ice in Fig. 19.2.4). The κ's of alloys (Fig. 19.2.4) are an order of magnitude smaller than for ''pure'' metals and *do not* exhibit the maxima characteristic of the ''pure'' metals at low T. Lattice disorder introduced by alloying, even in small amounts, and working a metal, even a pure metal, reduces κ. Annealing, in general, raises κ.

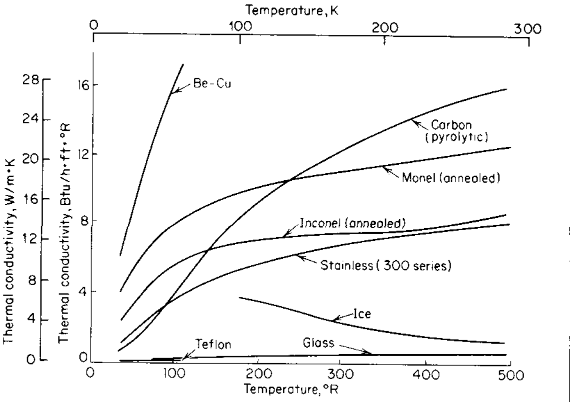

Fig. 19.2.4 Thermal conductivity of solids at low temperatures, part 1.

At 0°R, an absolutely pure and perfect single crystal of metal would have zero electrical resistance, $R_e = 0$. **Electrical resistance** arises from the scattering of the conduction electrons as they move through the lattice of metal ions under the influence of an externally applied electric field. Scattering arises for two reasons: (1) the amplitudes of the thermal vibrations of the lattice which increase with T at a rate proportional to $\sqrt{C}$ of the metal, and (2) the imperfections in the regularity of the lattice as caused by impurity atoms (solid-solution alloys included), lattice vacancies, dislocations, and grain boundaries. Resistances for ''pure'' metals increase with T and are roughly proportional to $C \cdot T$, except at very small T's where they are proportional to T^5 for absolutely pure crystals. The resistance due to impurities and imperfections is very roughly independent of T (Matthiessen's rule). For very pure metals, R_e is approximately constant below 20°R (11 K). Magnetic impurities can give rise to a minimum in resistivity [usually below 36°R (20 K)] called the Kondo effect. Resistivity in $\Omega \cdot$ cm (Fig. 19.2.6) is the R_e of a 1-cm cube.

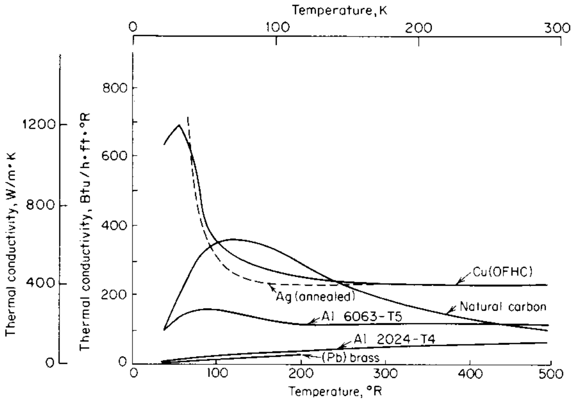

Fig. 19.2.5 Thermal conductivity of solids at low temperatures, part 2.

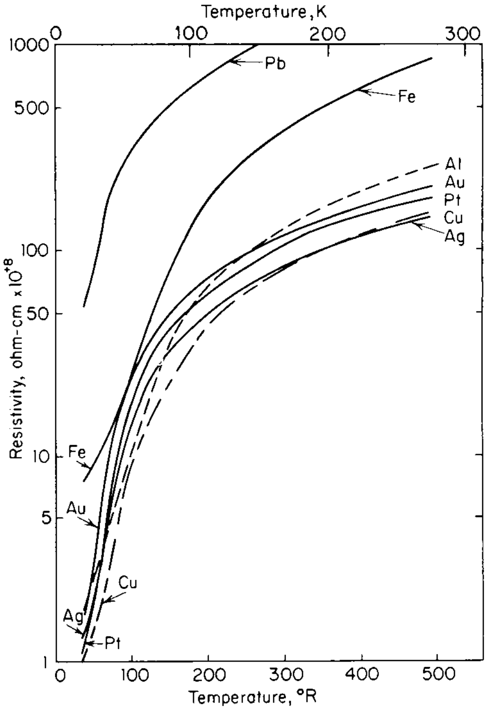

Fig. 19.2.6 Electric resistivity of solids at low temperatures.

Some metals, including elements (none from column 1 of the Mendeleeff table or the ferromagnetic elements), intermetallic compounds, metal alloys, and a number of oxides exhibit the phenomenon of **superconductivity,** characterized by a zero dc electrical resistance from 0°R up to a transition temperature T_c, at which normal resistance (value extrapolated from higher T) appears. In pure, single crystals the transition range ΔT, from $R_e = 0$ to the normal value, may be as small as a few millidegrees. Until the discovery of the so-called high-temperature superconductors (HTSCs), the higher T_c in a practical superconductor was that of the compound Nb_3Ge at 41.9°R (23.3 K). The search for higher T_c materials continues with T_c's that have already been reported as high as 239°R (133 K). Closed circuits consisting entirely of superconductors can support a persistent, resistanceless current without an external source of voltage. A superconducting circuit therefore maintains constant the value of the total magnetic flux that was enclosed by the circuit at the time it entered the superconducting state. Hence, superconducting materials influence the magnetic fields in their environment. For this reason, their use in the construction of equipment and apparatus must sometimes be avoided when expected temperatures could be below their T_c. Lead brasses and some solders, in particular Pb-Sn alloys, become superconductors.

Superconductivity, in materials designated Type 1 superconductor, is characterized by (1) perfect electric conductivity ($R_e = 0$) and (2) the internal magnetic induction $B = 0$ when H (external) is not zero (the Meissner effect). Persistent currents at the surface of the specimen shield the interior from H (external). H parallel to the surface of a specimen is continuous at the surface and falls off exponentially below the surface. The penetration depth of H for perfect (chemically and mechanically) specimens is of the order of 5×10^{-8} m. The penetration depth is larger for alloys, specimens with lattice imperfections, and all superconducting specimens near their superconducting transition temperatures T_c.

The normal-to-superconducting transition temperature T_c is also a function of H (external) as well as of the current being carried by the superconductor. For a Type I superconductor, the effect of H (external) is expressed by $T_c = T_{c0}\,[1 - (H_c/H_0)]^{1/2}$ where T_{c0} is the value of T_c at essentially zero H, and H_0 is the value of H_c at 0 K. For every $H < H_c$, a further decrease in T_c occurs when the superconductor is carrying a current, the maximum current density for maintaining the superconducting state being denoted J_c. Thus the superconducting state exists when $T < T_c$, $H < H_c$, and $J < J_c$, and the normal, or resistive, state exists when either $T > T_c$, $H > H_c$, or $J > J_c$. Because T_c decreases with increases in H and J, the practical application of a superconductor requires that it be maintained at a temperature considerably below its T_c.

Tables 19.2.1 and 19.2.2 are only representative. The number of superconductors including compounds and alloys runs into the hundreds. Of the chemical elements, 38 are known to become superconductors. Some that are not superconductors at normal pressures have high-pressure allotropic modifications that are superconductors, e.g., Bi, Si, and Ge at pressures of ~25, 130, and 120 × 10^8 Pa, respectively. Many of the superconducting alloys have nonsuperconducting constituents, and there are compounds all of whose constituent elements in their pure state are nonsuperconductors (see Table 19.2.2). It is interesting that good conductors Cu, Ag, and Au do not become superconductors to the lowest temperatures at which they have been tested (~0.01 K).

Type I superconductor is the designation given to those superconductors that exhibit a complete Meissner effect for $0 < H < H_c$; i.e., $B = 0$ until H_c is reached and penetration of the specimen by B becomes complete. The superconducting elemental metals in the pure and mechanically perfect state are Type I superconductors. Superconducting alloys (intermetallic compounds, solid solutions, and mixed phases) and work-hardened (unannealed) metals are Type II superconductors. For Type II superconductors, the Meissner effect is complete only for $0 < H < H_{c1}$ where H_{c1} is smaller than H_c. For $H_{c1} < H < H_{c2}$, the magnetic field penetrates the body of the specimen in the form of current vortices, called *fluxons,* that generate a fixed quantized amount of magnetic flux. Each fluxon is a cylindrical region of circulating current centered on a resistive core of material in which the superconductivity is suppressed. Hence the region defined by $H_{c1} < H < H_{c2}$ is called the *mixed state* of a Type II superconductor. The number of fluxons increases with H from zero at H_{c1} to a density at which complete overlap of the cores occurs along with the restoration of the normal state at H_{c2}. Thus H_{c2} is an upper limit to the superconducting state that increases as T decreases.

The alloys commonly used for high-field superconducting magnets (Nb_3Sn, NbTi, and NbZr, see Table 19.2.2) are Type II superconductors. H_c's for Type I superconductors are small, whereas magnet alloy wires have H_{c2} values that are 100 to 1,000 times higher.

Table 19.2.1 Transition Temperature T_0 and Critical Magnetic Fields H_0 of Some Type I Superconductors

Superconductor	T_0, K	H_0, A/m*
Nb	9.25	1.57×10^5
Pb	7.23	6.4×10^4
V	5.31	8.8×10^4
Hg	4.154	3.3×10^4
Sn	3.722	2.4×10^4
In	3.405	2.2×10^4
Al	1.175	8.4×10^3
Mo	0.916	7.2×10^3
Zr	0.53	3.7×10^3
Ti	0.39	8.0×10^3
W	0.0154	9.2×10
Na Bi	2.2	The elements, in pure state at normal pressure, are not superconductors.
Au_2Bi	1.7	
Cu S	1.6	

* To convert to oersteds, divide by 79.57.

Table 19.2.2 Transition Temperatures T_0 and Upper Critical Fields H_{c2} of Some High-Field, Type II Superconductors

Superconductor	T_0, K	H_{c2}, A/m*	T, K, for H_{c2}
$Al_{0.75}Ge_{0.25}Nb_3$	18.5	3.4×10^7	4.2
Nb_3Sn	18.0	1.9×10^7	4.2
$N_{0.93}Nb$	15.9	1.3×10^7	0
GaV_3	14.8	1.9×10^7	0
NbZr	10.8	7.4×10^6	0
$Nb_{0.2}Ti_{0.8}$	7.5	6.4×10^6	4.2
$Ti_{0.6}V_{0.4}$	7.0	8.8×10^6	2

* To convert to oersteds, divide by 79.57.

The fluxons in the mixed state of a Type II superconductor transporting a current are acted on by a Lorentz force that is proportional to **i** × **B** and acts in a direction perpendicular to **B**, and to **i** (the current intensity). Unless the fluxons are pinned to crystal lattice sites they are propelled across the superconductor under the influence of the Lorentz force. This movement involves the performance of work by the current and results in (1) production of heat within the superconductor and (2) the appearance of electrical resistance to the flow of current and, eventually, the destruction of the superconducting state. Fluxons are pinned by lattice imperfections (chemical and mechanical) and their displacement is resisted until the Lorentz force exceeds a breakaway value, freeing the fluxons to move. Lattice imperfections are introduced in the wires for high-field superconducting magnets to pin fluxons as well as to increase H_{c2}. Currents avoid the normal core of a pinned fluxon and thus retain their resistanceless character.

A current exceeding the critical value destroys the superconducting state. This critical current is determined by the critical Lorentz force (the product of current and magnetic field) at which pinning forces are exceeded. Hence, the critical value of an applied field depends on the current transported by the superconductor and vice versa. In general, the larger the applied field, the smaller is the critical current and vice versa. For superconducting magnets, it is essential that the values of the limiting magnetic field and the limiting current be simultaneously large.

A cryogenic refrigerator operating on the Carnot cycle (ideal) will require the least amount of input work to effect refrigeration (remove heat) at a low temperature and reject the heat at a higher temperature (usually ambient temperature). The ideal ratio of input work to refrigeration produced (both expressed in watts) can be given as $(T_1 - T_2)/T_2$ (the Carnot efficiency) where T_1 represents the temperature at which the heat will be rejected and T_2 the refrigeration temperature. Cryogenic refrigerators built to date do not approach this limit very closely: typically less than 40 percent of Carnot efficiency for the largest refrigerators and progressively less as the size of the refrigerator becomes smaller. For the Types I and II superconductors discussed above, practical application requires operation in the temperature region of 12 K or lower. Thus the lower limit (ideal refrigeration) for the work to heat removal ratio is seen to be 24 for operation between ambient temperature and 12 K. Raising the operating temperature of a superconducting system to 20, 40, or 77 K (the temperature of liquid nitrogen boiling at normal atmospheric pressure) reduces this ratio to 14, 6.5, and 2.9 respectively. Although the actual refrigerator may take 2.5 or more times this ratio, the saving in input energy that can be effected by raising the operating temperature is obvious. The equipment for producing refrigeration at these higher temperatures is also less complicated and less expensive than that needed for producing refrigeration at the lower temperatures required for the operation of the low-temperature superconductors.

In 1986 a true breakthrough was achieved in producing superconductors able to enter the superconducting state at much higher temperatures. The HTSCs are oxide compounds consisting of several elements. They are more like ceramics and have a complicated crystal structure. Many of them exhibit the superconducting phenomenon only along

Cu-O planes and their performance is greatly influenced by the oxygen content of the compound. If the crystals are not properly oriented, the HTSCs have small sections of superconductor connected by rather poorly conducting sections. This has been referred to a ''having weak links.'' Also the ability of these superconductors to conduct useful quantities of current can be severely limited by an imposed magnetic field at high temperatures. At temperatures below 30 K, their extraordinarily high H_{c2}'s allow them to carry high current densities at much higher magnetic fields than is possible with the conventional superconductors. The HTSC's are fabricated as thin films and multifilimentary wires. The necessity for optimum chemistry of the constituents, proper crystal orientation, and the fact that the ceramic-like nature of the superconductor makes it brittle complicate both the manufacture of the wires in useful lengths and the subsequent fabrication of equipment such as coils for motors, generators, etc. A great deal of research is continuing in the attempt to improve the commercial usefulness of the HTSCs. Table 19.2.3 lists several of the HTSCs along with pertinent properties. There is much active research on methods to produce HTSCs, including research on a number of other materials in addition to those listed in Table 19.2.3.

The **coefficient of linear expansion** α is $(1/L)$ (dL/dT). Thermal expansion of asymmetric crystals differs along different crystal axes. For an isotropic solid, the volume or cubical-expansion coefficient $(1/V)$ $(dV/dT) = 3\alpha$. Expansion coefficients do not vary appreciably in the temperature region near ambient temperatures, but they all approach zero at 0°R, and the approach to zero is tangential to the T axis ($d\alpha/dT = 0$ at 0°R.) Some expansion coefficients are negative at low temperatures, e.g., stainless steel, some Invars, and fused quartz. The expansivities of some crystals are negative in some directions even though their volume coefficients are positive. Cold-working may produce differences in expansivity in different directions. Annealing restores isotropy. Figure 19.2.7 shows the relative change in length, the integral of α from T to 528°R (293 K).

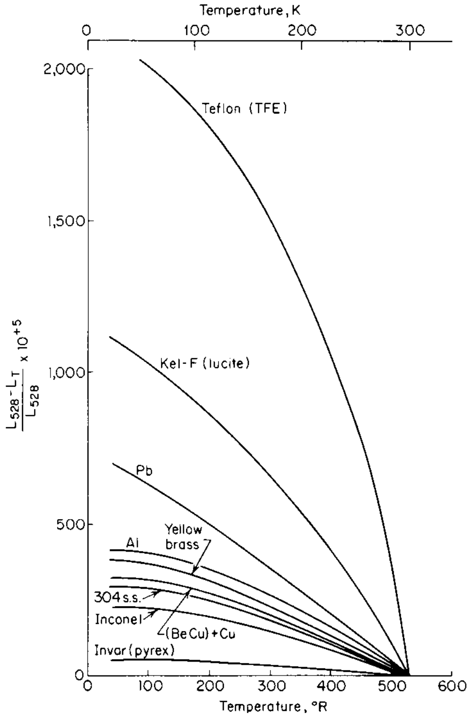

Fig. 19.2.7 Relative change in length from T to 528°R (293 K), equal to $\int_t^{528°\mathrm{R}} \alpha\, dt$. Bracketed materials are within ~5 percent of the curve shown.

Table 19.2.3 Properties and Status of some HTSCs

HTSC*	T_c, K	Comments
$La_{2-x}Ba_xCuO_4$	35	First HTSC discovered
$YBa_2Cu_3O_7$ (YBCO or 1-2-3)	95	First HTSC with T_c above normal boiling point of liquid nitrogen, H_{c2} = 30 T at 77 K; decreased interest due to weak link problem
$HgBa_2Ca_2Cu_3O_{8+\delta}$	133	Highest T_c to date
$Bi_2Sr_2Ca_2Cu_3O_8$ (2223 BSCCO)	110	$H_{c2} = 30$ T at 77 K, $J_c > 2 \times 10^4$ A/cm^2 at 77 K and $H_{ext} = 0$; industrial pilot-plant-scale production
$Tl_2Ba_2CaCu_2O_8$ (2212 TBCCO)	~115	$J_c = 7 \times 10^6$ A/cm^2 at 75 K and $H_{ext} = 0$

* These HTSCs are representative of classes of superconductors.

Because the cold interiors of cryogenic equipment must at times be warmed to ambient temperature, provisions have to be made for those changes in dimensions of the interior that results from large changes in T. Even for equipment of ordinary size, these changes can be too large for accommodation within the elastic limits of the materials of construction or of the structure. In such situations, flexibility has to be designed into the structure. For example, bellows and U bends are commonly employed to obtain this flexibility in insulated transfer lines for liquid cryogens.

The **mechanical properties** important for the design and construction of cryogenic equipment are the same as for other temperatures (see also Sec. 5). Frequently, the choice of materials for the construction of cryogenic equipment will depend upon other considerations besides mechanical strength, e.g., lightness (density or weight), thermal conductivity (heat transfer along structural support members), and thermal expansivity (change of dimensions when cycling between ambient and low temperatures). Frequently, mechanical properties at low temperature are significantly different from properties at ambient temperature. This makes room-temperature data unreliable for engineering use at low temperatures.

It is not possible to make generalizations that would not have numerous exceptions about the temperature variations of the mechanical properties. In this discussion, Figs. 19.2.8 to 19.2.10 are only illustrative guides. There is no substitute for test data on a truly representative specimen when designing for the limit of effectiveness of a cryogenic material or structure. Just as the mechanical properties at ambient temperatures are dependent upon the impurities (metallic and nonmetallic), their chemical nature and concentration, the thermal history of the specimen, the amount and kind of working (microstructure and dislocations of the lattice), and the rate of loading and type of stress (uni-, bi-, and triaxial), so also are the changes, quantitative and qualitative, in the mechanical properties when changing the temperature from ambient to low.

Metals, nonmetallic solids (glass), plastics, and elastomers are discussed in order. The metals are classed by their lattice (crystal) symmetry.

The **f.c.c. metals** and their alloys are most commonly used for the construction of cryogenic equipment. Al, Cu, Ni, their alloys, and the austenitic stainless steels of the 18-8 type (300 series) are f.c.c. They do not exhibit an impact (or a notched-tensile) ductile-to-brittle transition at low temperatures. As a general rule, which has some exceptions, the mechanical properties of these metals improve as T is reduced: (1) Young's modulus at 40°R (22 K) is 5 to 20 percent larger than at 530°R (294 K), (2) the yield strength at 40°R (22 K) is considerably greater than the strength at 530°R (294 K). (Cu is an exception; see Fig. 19.2.9), and (3) the fatigue properties at low T are improved. There is a large difference between the yield strength and the ultimate tensile strength of these metals and alloys, especially when they have been annealed. Pb (f.c.c.) and In (face centered tetragonal) are used for low-T deformable gaskets because of their creep properties. Low-T creep data are meager, but the rate of creep decreases with T.

Beta brass (f.c.c.) is ductile down to 7°R, though it is like all alloys of

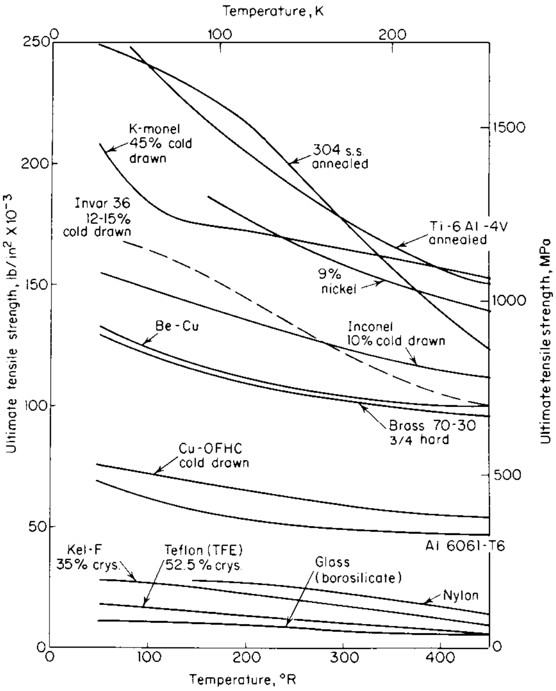

Fig. 19.2.8 Ultimate tensile strength of solids at low temperatures.

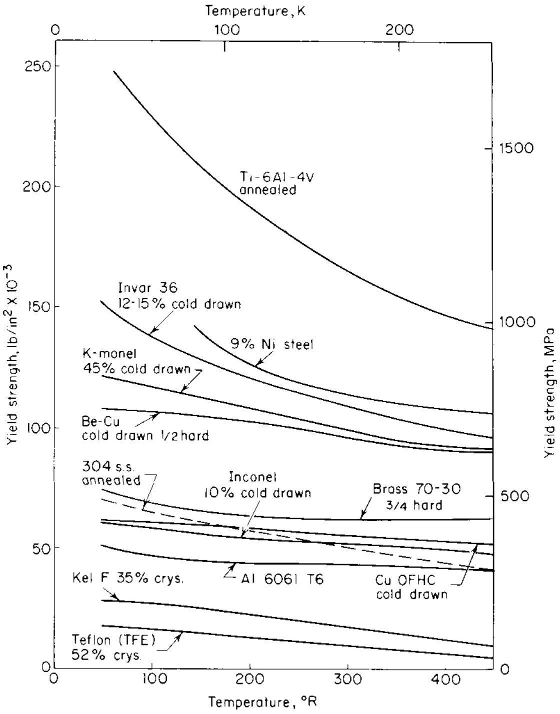

Fig. 19.2.9 Yield strength of solids at low temperatures.

Cu in being less ductile than Cu itself. Thin brass is sometimes porous, and this limits its usefulness for high-vacuum enclosures at low temperatures. Free-machining brass normally contains Pb, which even in low concentrations can render the brass superconducting at LHe temperatures. In the superconducting state, it may then affect the magnetic field in its vicinity (see electrical properties, above).

The b.c.c. **metals and alloys** are normally classed as undesirable. These include Fe, the martensitic steels (low carbon and the 400 series of stainless steels), Mo, and Nb. If not brittle at room temperature, they have a ductile-to-brittle transition at low *T*. Working can induce the austenite-to-martensite transition in some steels. AISI 301 austenitic stainless is an example. It remains moderately ''tough'' at very low *T* and is a valuable material for construction, but cold-drawing, in excess of 70 percent strain, induces a partial transformation of structure and reduces the elongation and notched-strength ratio to nearly zero. This same type of reduction in toughness is observed in the ''heat-treatable'' stainless steels. Improved tensile properties can be obtained by cold-working and by heat-treating. These alloys usually have decreased toughness at low temperatures. Alloys of V, Nb and Ta, although not f.c.c., behave well as regards brittleness at low temperatures. These alloys have the advantage of being suitable for use at high *T*. Carbon

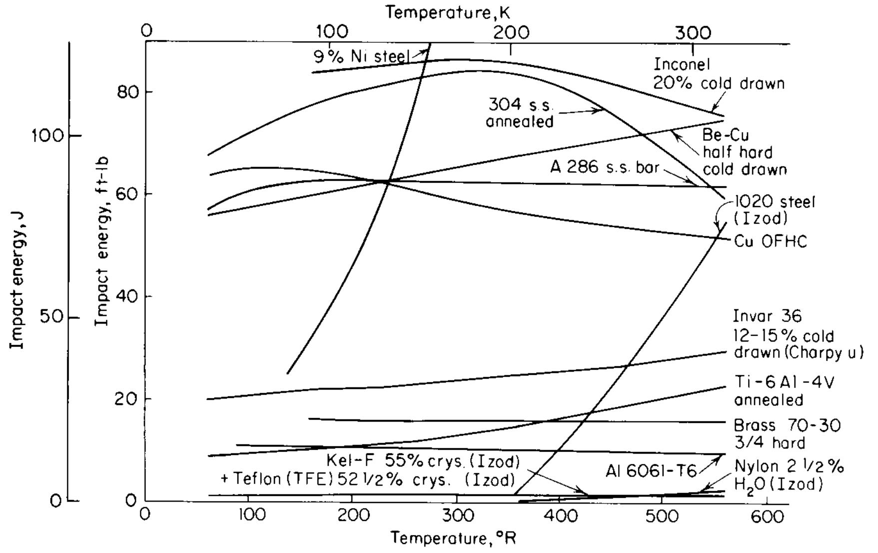

Fig. 19.2.10 Impact energy for solids at low temperatures (Charpy V unless noted). For Kel-F, nylon, and Teflon, the impact energy units are foot-pound per inch of notch width.

steels, though brittle, have found special uses at low temperatures, e.g., in the construction of the expansion engines of the Collins He liquefier and cryostat.

The **h.c.p. metals** exhibit properties intermediate between those of the f.c.c. and b.c.c. metals; e.g., Zn undergoes a brittle-to-ductile transition, whereas Zr and pure Ti do not. Ti and some Ti alloys, having a h.c.p. structure, remain moderately ductile at low *T* and are excellent for many applications. They have high strength-to-weight and strength-to-thermal-conductivity ratios. The low-*T* properties of Ti and its alloys are extremely sensitive to even small amounts of O, N, H, and C.

Brittle materials, when in very thin sheets, can have a high degree of flexibility, and this makes them useful for some cryogenic applications. **Flexibility** cannot be used as a criterion for ductility.

Ordinarily, normal welding practices are observed in the construction of cryogenic equipment. These practices, however, are modified in accordance with available knowledge of the performance of the metals at low *T*.

Nonmetal materials for construction are in many cases brittle, or they are susceptible to brittle fracture. The strength of **glass,** measured at a constant rate of loading, increases on going to low *T*. Failure occurs at a lower stress when the glass surface contains cracks and abrasions. The strength of glass can be improved by tempering the surface, i.e., by putting the surface under compression.

The **plastics** increase in strength as *T* is decreased, but this is accompanied by a rapid decrease in elongation in a tensile test and a decrease in impact resistance. Teflon and the glass-reinforced plastics (e.g., glass-reinforced epoxy resin) retain appreciable impact resistance as *T* is lowered. Teflon, which is polytetrafluoroethylene, can be deformed plastically at *T*'s as low as 7°R (3.9 K). The amount is considerably less than at room temperature, but it is enough to make Teflon very useful for some cryogenic applications. The glass-reinforced epoxies, besides having appreciable impact resistance at low *T*'s, also have high strength-to-weight and strength-to-thermal-conductivity ratios.

All the **elastomers** become brittle at low *T*. However, elastomers like natural rubber, nitrile rubber, Viton A, and plastics such as Mylar and nylon that become brittle at low *T* can be used for static seal gaskets when *highly compressed* at room temperature, prior to cooling.

PROPERTIES OF CRYOGENIC FLUIDS (CRYOGENS)

Table 19.2.4 indicates the **temperature ranges** accessible with liquid cryogen baths. There are two inaccessible ranges: (1) between helium and hydrogen [9.36 to 24.9°R (5.2 to 13.8 K)] and (2) between neon and oxygen [80 to 97.9°R (44.4 to 54.4 K)]. The limiting temperatures are set by the critical and triple points. Pumping on a cryogen bath to lower the pressure results in lower temperatures ultimately reaching the triple point; except for helium, further pumping leads to solidification and in practice poor heat-transfer. Raising the bath pressure results in higher boiling temperatures with the limit set by the critical temperature, above which liquid and vapor phases cannot coexist. Cryogen baths are most frequently operated near atmospheric pressure.

The algebraic sign of the Joule-Thomson coefficient determines whether a gas cools or warms upon free expansion. Figure 19.2.11, based on the law of corresponding states, allows rough calculations of inversion temperatures and pressures. Free (Joule-Thomson) expansion inside the ''cooling'' region results in cooling, i.e., refrigeration; outside this region, heating results. Upper inversion temperatures at 14.7 psia (0.101 MPa) are given in Table 19.2.4.

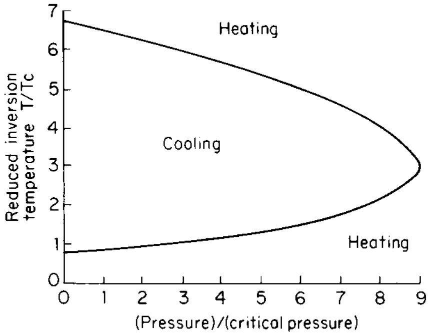

Fig. 19.2.11 Reduced inversion temperature versus reduced pressure.

Volumetric latent heats (Table 19.2.4) decrease with the lower-boiling cryogens and emphasize the importance of the insulation in storing and handling LH_2 and LHe.

Two forms of **hydrogen** and **deuterium** exist: **ortho** and **para.** The ortho and para molecules differ in the relative orientation of the nuclear spins of the two atoms composing the diatomic molecule. In the ortho form of H_2 the nuclear spins of the two atoms in the molecule are parallel (in the same direction), whereas in the para form the spins are antiparallel. The relative orientations for ortho and para D_2 differ from those for H_2. The thermodynamic equilibrium composition of ortho and para varieties is temperature dependent as shown in Fig. 19.2.12.

In liquid hydrogen, the uncatalyzed ortho-para reaction is second order (proportional to the square of the o-H_2 concentration) with a rate constant of 0.0114/h. Thus, in the course of uncatalyzed liquefaction, normal LH_2 (75 percent ortho) is produced. Since the heat of conversion

Table 19.2.4 Common Cryogen Properties

		Triple-point					Heat of vaporization*				
Cryogen	Boiling point, °R	Temp., °R	Pressure, lb/in² abs	Critical temp., °R	Critical pressure, lb/in² abs	Upper inversion temp,* °R	Btu/lbm	Btu/ft³	Liquid density,† lbm/ft³	Vapor density,† lbm/ft³	Gas density,‡ lbm/ft³
He^3	5.7			6	17	72	3.6	13.4	3.72	1.50	0.0084
He^4	7.6			9.36	33.2	92	8.8	68.6	7.80	1.04	0.0111
H_2 (equilib.)	36.5	24.9	1.02	59.4	187.7	368	192	853	4.42	0.0837	0.00561
D_2 (normal)	42.6	33.7	2.49	69.0	240		131	1,343	10.25	0.14	0.0112
Ne	48.7	44.2	6.26	80.0	395.3	486	37.1	2,790	75.35	0.58	0.056
N_2	139.2	113.7	1.86	227.27	492.3	1,115	85.9	4,294	50.4	0.28	0.078
Air	(141.8)						88.22	4,813	54.56	0.28	0.0807
A	157.1	150.8	9.97	271.3	709.8	1,300	70.2	6,135	87.4	0.36	0.111
F_2	151	(96.4)§		259.3	808		74.1	6,965	94	0.50	0.106
O_2	162.3	97.9	0.022	278.59	736.3	1,605	91.5	6,538	71.24	0.28	0.089
CH_4	201.1	163.2	1.69	343.27	673		219.2	5,804	26.5	0.115	0.0448

NOTE: To convert °R to K, multiply °R by 0.556; to convert lb/in² to MPa, multiply by 0.00689; to convert Btu/lbm to J/g, multiply by 2.326; to convert Btu/ft³ to J/m³, multiply by 3.73 × 10^4; to convert lbm/ft³ to kg/m³, multiply by 16.01.

* 14.7 lb/in² abs.

† At normal boiling point.

‡ At 14.7 lb/in² abs and 492°R.

§ Melting point, 14.7 lb/in² abs.

of o-H_2 to p-H_2 at the normal boiling point is 302 Btu/lbm (702 J/g) (greater than the heat of vaporization; see Table 19.2.4), long-term storage of normal LH_2 is impractical. Therefore, in the commercial production of LH_2 a catalyst is used to produce LH_2 with more than 95 percent para which for practical purposes is equivalent to the equilibrium composition (99.8 percent para).

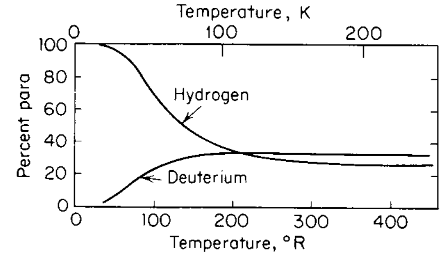

Fig. 19.2.12 Thermodynamic equilibrium composition of ortho and para varieties of H_2 and D_2; percent ortho = 100 − percent para.

For pressures of no more than 150 lb/in^2 (1.03 MPa) and temperatures at least twice the critical temperature, the ideal gas law ($PV = nRT$) enables one to calculate *PVT* data with sufficient accuracy for many engineering purposes. For cases in which the ideal gas law is not adequate or in which experimental data are not available, the procedures outlined in Sec. 4 may be used. The theoretical prediction of *PVT* data for liquids is considerably more difficult than for gases. Consequently no universal equation analogous to the perfect gas law exists for extensive ranges of *P* and *T*.

Reduced quantum mechanical correlations of saturated liquid densities, viscosities, and thermal conductivities for several cryogens are shown in Figs. 19.2.13 to 19.2.15. Dashed lines represent areas void of experimental data.

Utilization of Figs. 19.2.13 to 19.2.15 requires knowledge of the

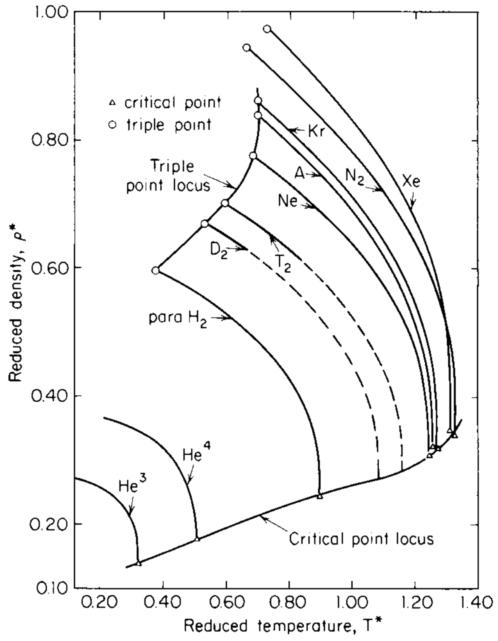

Fig. 19.2.13 Reduced density versus reduced temperature along the saturation curve for cryogenic liquids. Values of T^* are obtained by dividing the desired temperature (°R) by the value of T/T^* given in Table 19.2.5. Reduced density ρ^* must be multiplied by ρ/ρ^* given in Table 19.2.5 to obtain density in lbm/ft^3. Multiply lbm/ft^3 by 16.01 to obtain kg/m^3 and °R by 0.556 to obtain K.

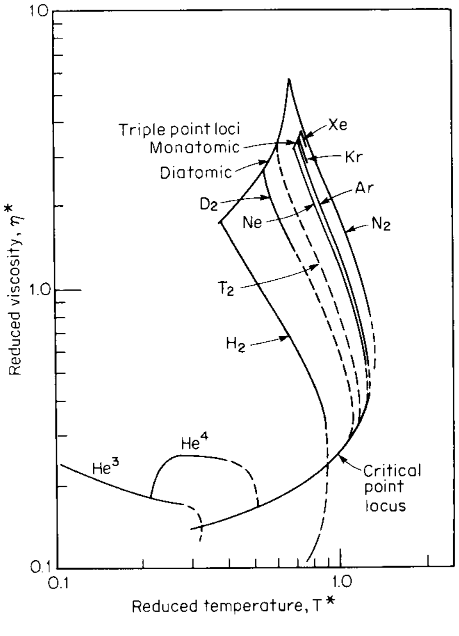

Fig. 19.2.14 Reduced viscosity versus reduced temperature along the saturation curve for cryogenic liquids. Values of T^* are obtained by dividing the desired temperature (°R) by the value of T/T^* given in Table 19.2.5. Reduced viscosity η^* must be multiplied by η/η^* given in Table 19.2.5 to obtain viscosity in lbm/ft · h. Multiply lbm/ft · h by 0.000413 to obtain Pa · s.

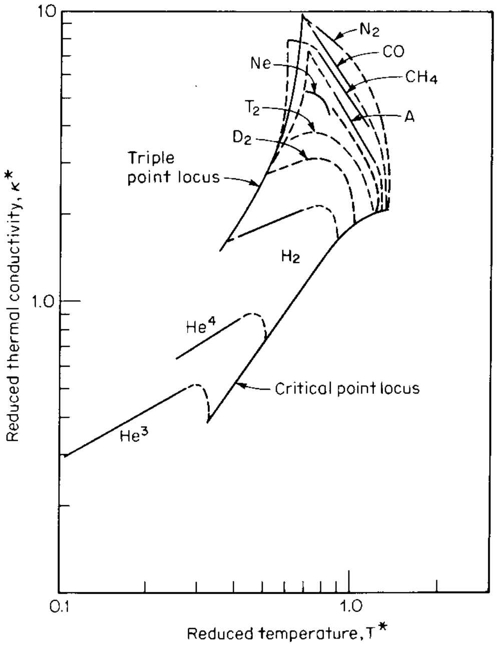

Fig. 19.2.15 Reduced thermal conductivity versus reduced temperature along the saturation curve for cryogenic liquids. Values of T^* are obtained by dividing the desired temperature (°R) by the value of T/T^* given in Table 19.2.5. Reduced thermal conductivity κ^* must be multiplied by κ/κ^* given in Table 19.2.5 to obtain thermal conductivity in Btu/h · ft · °R. Multiply by 1.73 to obtain W/m · K.

constants ε and σ for the Lennard-Jones function for the intermolecular potential energy $\varphi(r)$ of a pair of molecules separated by a distance r, so that $\varphi(r) = 4\varepsilon[(\sigma/r)^{12} - (\sigma/r)^{6}]$. Here ε is the depth of the minimum in the potential energy of a pair of molecules, and σ is the separation of the pair at which $\varphi = 0$. To facilitate use of Figs. 19.2.13 to 19.2.15, all required constants and conversion factors have been combined and are listed in Table 19.2.5.

Table 19.2.5 Multiplication Factors for Use with Figs. 19.2.13 to 19.2.15

Substance	T/T^*, °R	ρ/ρ^*, lbm/ft^3	η/η^*, lb/ft-hr	κ/κ^*, Btu/(h·ft·°R)
Helium 3	18.4	18.71	0.0311	0.0204
Helium 4	18.4	24.84	0.0359	0.0178
Para-hydrogen	66.1	8.06	0.0360	0.0363
Deuterium	63.4	16.22	0.0500	0.0257
Tritium	62.1	24.37	0.0607	0.0210
Neon	64.1	100.66	0.1300	0.0128
Nitrogen	171.1	57.39	0.1382	0.0098
Argon	215.6	104.84	0.2183	0.0109
Krypton	297.4	177.1		0.0080
Xenon	397.8	197.4		

Thermal-expansion coefficient $(1/V)(\partial V/\partial T)_p$ and **compressibility** $(1/V)(\partial V/\partial P)_T$ data for the cryogenic liquids are obtained as derivatives of *PVT* data. Liquids hydrogen and helium have large thermal-expansion and compressibility coefficients (see Table 19.2.6).

Table 19.2.6 Compressibility and Thermal-Expansion Coefficients. Coefficients for Liquid Nitrogen, Hydrogen, and Helium at 14.7 lb/in² abs (0.101 MPa)

Cryogen	T, °R (K)	Compressibility, $(1/V)(\partial V/\partial P)_T$, 1/(lb/in^2 abs) (1/MPa)	Thermal expansion, $(1/V)(\partial V/\partial T)_p$, 1/°R (1/K)
Helium 4	7 (4)	0.0028 (0.4)	0.085 (0.153)
Nitrogen	139 (77)	0.000020 (0.0029)	0.0032 (0.0058)
Para-hydrogen	36 (20)	0.000134 (0.0195)	0.0090 (0.0162)
Water	540 (300)	0.0000036 (0.000526)	0.00015 (0.00027)

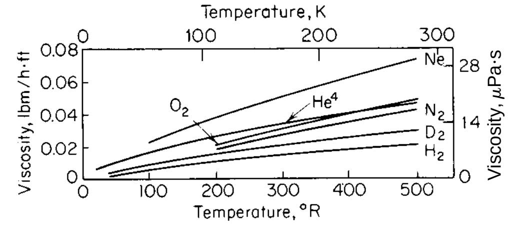

Fig. 19.2.16 Viscosity of selected gases at low temperatures and 14.7 lb/in^2 abs (0.1013 MPa).

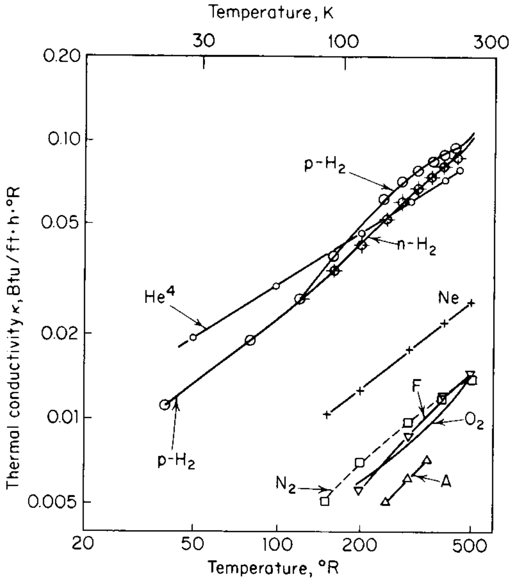

Fig. 19.2.17 Thermal conductivity of gases at low temperatures and 14.7 lb/in^2 abs (0.1013 MPa).

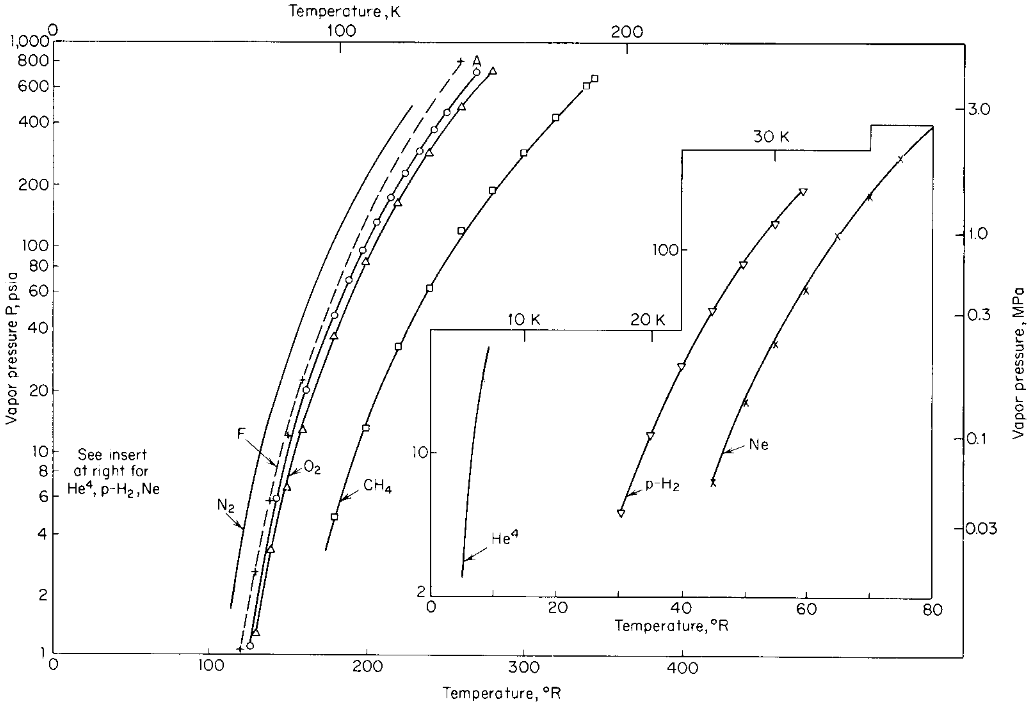

Fig. 19.2.18 Vapor pressures of the common cryogens.

For gases, kinetic theory predicts an increase of **viscosity** with rising temperature as shown in Fig. 19.2.16 ($\eta \sim \sqrt{T}$). Normally, the **thermal conductivities** of liquids decrease with increasing temperature, whereas the thermal conductivities of gases increase. Fig. 19.2.17 presents thermal conductivity for several gases as a function of temperature at 14.7 psia (0.1013 MPa).

Figure 19.2.18 is a graph of **vapor pressures** of the common cryogens.

Figure 19.2.19 presents **specific heats** at constant pressure for several of the cryogenic liquids, while Fig. 19.2.20 presents the specific heats for liquid hydrogen and helium along the saturation curve. Specific heats of gases are given in Fig. 19.2.21.

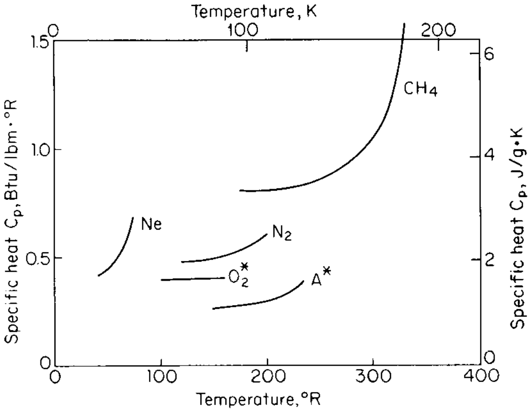

Fig. 19.2.19 Specific heat at constant pressure (C_p) for liquid cryogens at saturation pressure. Asterisk means C_p evaluated along an isobar at 14.7 lb/in^2 abs (0.1013 MPa) instead of saturation pressure.

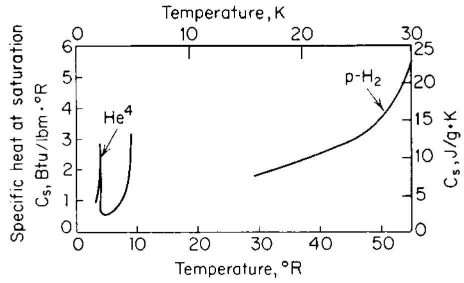

Fig. 19.2.20 Specific heats at saturation C_s along the saturation curve versus temperature for ^{4}He and para hydrogen.

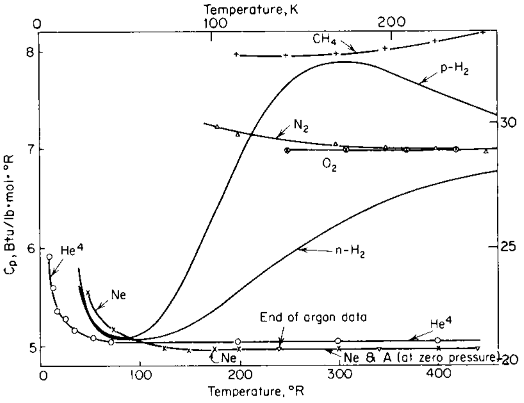

Fig. 19.2.21 Specific heat C_p of cryogen gases at 14.7 lb/in^2 abs (0.1013 MPa) versus temperature.

INSTRUMENTATION

(See also Secs. 15 and 16.)

The usual instrumentation problems of engineering practice are further complicated by low-temperature problems which require special calibration procedures.

Pressures are measured with normally used apparatus, e.g., bourdon gages, transducers, and manometers. If the measuring device is located outside the cryogenic environment and insulated from it by a thermal barrier, no special problems are introduced. Occasionally response time requirements necessitate the placing of the transducer in the cryogen space. Then the problems of temperature compensation and calibration must be considered.

Level measurements currently utilize several methods. The differential-pressure method can in principle be used with all cryogens. With cryogens of low densities and low heats of vaporization (e.g., H_2 and He) pressure oscillation occurs in the liquid leg. In the case of LH_2 these can be eliminated by the proper design of the liquid leg and the introduction of He gas which does not condense.

Direct weighing is also used to determine ''levels,'' although difficulties are encountered owing to the large tare-to-cryogen weight ratios (as large as 10 to 1 for a full container in the case of H_2) and extraneous loadings introduced by permanently connected piping, frost, and wind. Weighing systems have nevertheless been successfully used on 50,000-gal (190 m^3) LH_2 dewars with an accuracy in level changes of 250 gal (1 m^3).

With the **capacitance gage,** the sensing element consists of a concentric tube capacitor which can be calibrated to give liquid-level measurements accurate to a few tenths of a percent. An inexpensive, accurate **point-level sensor** is easily fabricated from a carbon resistor, typically 1⁄10 W, 1,500 Ω. Its use depends upon the fact that its resistance is a strong function of temperature (Fig. 19.2.22). A small current (50 to 80 mA) is passed through the resistor to heat it. Since heat from the resistor is dissipated more readily in the liquid than in the gas, the

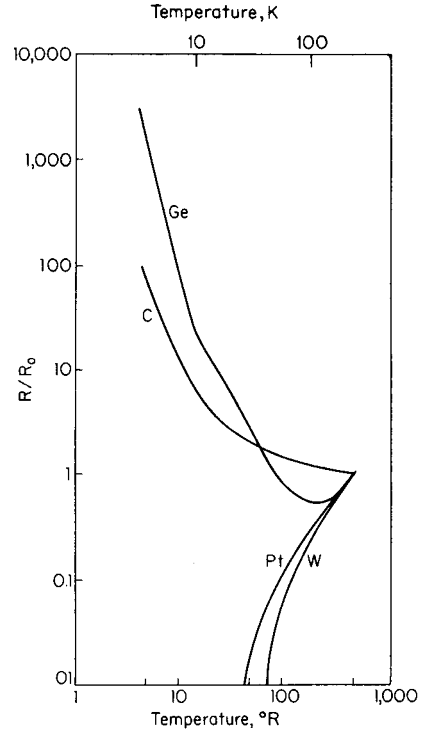

Fig. 19.2.22 Ratio of resistance R to the resistance R_0 at 492°R (273 K) versus temperature for several resistance thermometers. (Values for germanium and carbon are representative only.)

temperature of the resistor undergoes a step increase when the resistor environment changes from liquid to gas, whereas the resistance decreases stepwise.

Flow measurements utilize orifices, venturis, and turbine-type meters. Various devices for direct mass-flow measurement exist, but none are widely used. ''Quality'' meters have been built to determine the percentage liquid in two-phase flow. The determination of quality is complicated by nonequilibrium of temperature in the gas phase.

Temperature measurements are usually made with thermocouples, resistance thermometers, and vapor-pressure thermometers.

The **vapor-pressure thermometer** depends upon a vapor-pressure-temperature relationship (see Fig. 19.2.18). In general, a small cavity is filled with a gas which has a condensation temperature in the vicinity of the temperature to be measured. If sufficient gas is present, the pressure will be that of the liquid vapor pressure at the coldest part of the measuring system. Maximum speed of response is obtained if the total quantity of gas is minimized to permit only a thin film of liquid to form. In the case of H_2, a catalyst (e.g., iron hydroxide) to promote ortho-para conversion should be included in the bulb for an accurate measurement since the vapor pressure of H_2 is dependent upon its ortho-para composition (see above).

The pressure at the surface of a liquid cryogen in a dewar may, under conditions of thermal equilibrium, be used to measure the temperature of the cryogen and of apparatus immersed in it. Corrections for the pressure of the hydrostatic head of liquid may be required. If using this method, it should be realized that large vertical temperature gradients can exist in an unstirred liquid cryogen in a vessel with good insulation.

Thermocouples are favored for the measurement of temperatures because of their low cost, ease of application, and rapid response. The thermoelectric powers (temperature sensitivities) of the thermocouples commonly used at higher *T*'s decrease with decreasing temperature, and spurious emfs generated in wires of non-uniform composition are troublesome. The proper use of a reference junction at a known fixed temperature close to the temperature being measured is often advantageous for accuracy. Variability in composition of thermocouple alloys makes individual calibrations necessary for accurate results. Figure 19.2.23 gives typical thermocouple emf-versus-temperature curves for some common thermocouples. A thermocouple consisting of gold with 0.07-at % iron versus chromel has been found useful over the entire temperature range from 2 to 300 K. Its sensitivity increases from 11 μV/K at 4 K to 17 μV/K at 77 K to 22 μV/K at 300 K.

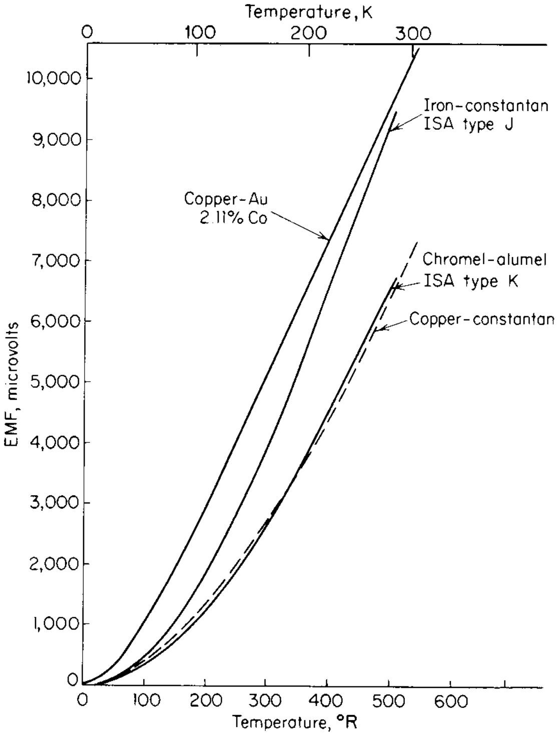

Fig. 19.2.23 EMF versus temperature for several thermocouples.

Radio carbon resistors (*Allen Bradley* for 2 to 100 K and *Speer Carbon* for 0.01 to 4.2 K) are much used as temperature sensors because of their low cost and high sensitivity. However, the permanence and stability of their calibrations are not very good, and germanium (doped) resistance thermometers are preferred (0.01 to 100 K) for their good reproducibility of calibration, although they are considerably more expensive.

The carbon and germanium resistors (thermometers) in magnetic fields *H* exhibit a magnetoresistance (Hall) effect, proportional to H^2, which complicates the determination of temperatures. Recent developments include: (1) a carbon-glass electrical resistor which has a stable calibration; and (2) the electrical capacitance thermometer (dielectric: $SrTiO_3$, a perovskite in solution in a solid SiO_2 or Al_2O_3 glass) with very small (approaching zero) magnetic effect and a stable calibration. The capacitance thermometer is preferred when magnetic fields are present.

The Rh-0.5-at % Fe, wire-coil thermometers, usable in the range of 0.1 to 300 K, have a high sensitivity especially below 10 K.

INSULATION

(See also Sec. 6.)

The degree of thermal isolation required at low *T*'s is normally greater than at elevated *T*'s because it is more costly to remove heat leaking into a low-temperature system than to replace heat lost at elevated *T*'s, as demonstrated by the Carnot cycle. Other differences between insulating for low and elevated temperatures are:

1. Condensation of moisture in low-temperature insulation is possible. When this occurs, the conductance of the insulation is significantly increased. This problem is avoided by a vapor barrier on the outside of the insulation.

2. Condensation of the atmosphere in the insulation and surface washing by the condensed liquid are possible when the insulating surfaces are below 150°R (83 K). This can result in a large added transfer of heat. It is avoided by using an outer cover or surface impermeable to air, and evacuating the insulating space or replacing the atmosphere in it with a noncondensable gas.

At low temperatures, as at other temperatures, the fundamental modes of heat transfer are conduction and radiation, and it is against these that insulation is used. Convection of heat is practically eliminated by the insulation.

Low-T insulation categories are (1) high vacuum, with or without multiple radiation shields, (2) powders, (3) rigid foams, and (4) low-conductivity solids, such as balsa wood and corkboard.

1. (i) **Heat is transferred across an evacuated space** by radiation and by conduction through the residual gas. The conductivity of a gas is almost independent of its density (pressure) as the pressure is reduced in an evacuated space until the free paths of the molecules are increased to an appreciable fraction of the separation of the containing walls.

The molecular *mean* free path at 1 atm (0.1013 MPa) and 492°R (273 K) in air is 4×10^{-6} in (10^{-7} m); in H_2, 6×10^{-6} in (1.5×10^{-7} m); and in He, 10×10^{-6} in (2.5×10^{-7} m). At pressures lower than about 10^{-3} torr (0.133 Pa), the rate of heat transfer Q by residual gas between parallel surfaces at T_1 and T_2 less than 1 in (0.0254 m) apart is approximately $(Q/A) \approx (\text{const})\, \alpha P(T_2 - T_1)$, where P is pressure *measured* in torr with a gage at *ambient T* and α is an *overall* accommodation coefficient of gas molecules in collision with the walls. Thus, $\alpha = \alpha_1\alpha_2/[\alpha_2 + \alpha_1(1 - \alpha_2)]$, where α_1 and α_2 are accommodation coefficients of gas molecules at the two walls (see Table 19.2.7). Table 19.2.8 gives the values of the *(const)* in the equation for *Q*/*A*.

The **rate of radiative heat transfer** between parallel surfaces at T_1 and $T_2 > T_1$, where emissivities are e_1 and e_2, respectively, is $Q = e_1e_2s/[e_2 + (1 - e_2)e_1]\,[T_2^4 - T_1^4]$, where s is the Stefan-Boltzmann constant whose value is 1.712×10^{-9} Btu/ft$^2 \cdot$ h $\cdot$ °R^4

Table 19.2.7 Approximate Values of Accomodation Coefficients, α

Temp, °R (K)	He	H_2	Air
540 (300)	0.3	0.3	0.8–0.9
140 (78)	0.6	0.5	1
35 (19)	0.6	1	1

Table 19.2.8 Constant in the Gas-Conduction Equation

$Q/A \approx (\text{const})\ \alpha P(T_2 - T_1)$

Gas	T_2 and T_1, °R (K)	Const, Btu/h · ft² · torr · °R (J/N · s · K)
N_2	<700 (389)	28 (1.19)
O_2	<540 (300)	26 (1.11)
H_2	540 and 140 (300 and 78)	93 (3.96)
H_2	140 and 36 (78 and 20)	70 (2.98)
He	Any	49 (2.09)

NOTE: T_1—inner wall (cold). T_2—outer wall (hot).

(5.67×10^{-8} W/m²K⁴) (see also Sec. 4). Table 19.2.9 contains emissivities for 540°R (300 K) thermal radiation; these for most engineering purposes may be considered equivalent to minimal values of the *e*'s for very carefully prepared and cleaned surfaces. Organic coatings have high emissivities approaching unity (>0.9). Mechanically polished surfaces have higher *e*'s than the minimal values. Handling a surface transfers an absorbing (high *e*) coating to it.

(ii) **Refrigerated Radiation Shield** An LN_2 (140°R) (78 K) cooled surface interposed in the insulating vacuum space of an LH_2 or an LHe container reduces the heat transferred by radiation to the LH_2 or LHe by a factor of about 250 when compared with that transferred by a surface of the same emissivity at room temperature without the interposition of the refrigerated shield. Nearly all the heat radiated by the surface at room temperature is absorbed by the LN_2, and this results in evaporation of the relatively inexpensive LN_2.

(iii) **A floating radiation shield** is an opaque layer (e.g., a metal sheet), having surfaces of low emissivity suspended in the vacuum space with a minimum of thermal contacts with surfaces that are warmer or cooler. If the emissivity of the surfaces of the floating shield is the same as the emissivity of the vacuum-space walls, the floating shield decreases the *radiative* heat transfer by a factor of 2. For *m* floating shields arranged in series between the inner cold and outer warm surfaces, the rate of *radiative* heat transfer becomes $\dot{Q}_m$ (*m* floating shields) $= [1/(m+1)]\dot{Q}_0$ where $\dot{Q}_0$ = heat transfer for $m = 0$.

(iv) **Multilayer (Super) Insulation** The principle of multiple floating radiation shields has been extended to the use of many thin metal layers (up to 75 layers per inch or 3,000 layers per meter) separated by thin thermal insulation. Best results have been obtained with Al foil separated by glass-fiber paper. Other materials have been used successfully, e.g., nylon nets as separators and aluminized Mylar with no spacer material. The *apparent* thermal *conductivity* of Al foil multilayer insulation spaced with glass fiber paper is variable depending on the number of layers of foil, the thickness of the paper, and the thickness and compacting of the insulating layer. For $\Delta T = (540 - 36)$°R or $(300 - 20)$ K, apparent mean conductivities range from 2.0 to 4.0×10^{-5} Btu/h · ft · °R (3.5 to 7×10^{-5} J/s · m · K). For $\Delta T = [540 - 140$°R $(300 - 78)$ K], the values are about one-third larger. Using nylon nets, glass fabric or glass fibers for separating the Al foil increases the conductance from three to seven times. The above conductivities are for the direction normal to the foil surfaces. Lateral conductivities are many thousands of times larger. An advantage of aluminized Mylar is its reduced lateral conductivity. Another advantage is its relatively low density (one-third to one-half of other multilayer insulations). If the residual gas pressure in multilayer insulation is less than 10^{-4} torr, the conductance is practically independent of the residual gas pressure. At 10^{-3} torr (0.133 Pa), the conductance is increased by the order of 50 percent.

Table 19.2.9 Minimal Values of Emissivity of Metal Surfaces at Various Temperatures for 540°R (300 K) Thermal Radiation

	Surface temp, °R (K)		
Surfaces	7 (3.9)	140 (78)	540 (300)
Copper	0.005	0.008	0.018
Silver	0.0044	0.008	0.02
Aluminum	0.011	0.018	0.03
Chromium		0.08	0.08
Nickel		0.022	0.04
Brass	0.018		0.035
Stainless steel 18-8		0.048	0.08
50 Pb-50 Sn solder		0.032	
Glass, paints, carbon			>0.9
Nickel plate on copper		0.033	

2. (i) **Powder insulation** consists of finely divided materials. Conductances of evacuated powders may vary by as much as 100 percent with variations of particle size and *apparent* density (packing) of the powder. The most commonly used powders are perlite, expanded SiO_2 (aerogel), calcium-silicate, diatomaceous earth, and carbon black. The conductance of powder insulation is practically independent of gas pressure down to 10 torr (1.333 kPa), at which pressure it begins to decrease rapidly with decreasing pressure as the free path of the gas molecules becomes comparable with the space between powder particles. The most rapid decrease of conductance occurs between 1 and 10^{-1} torr (133 and 13.3 Pa). At 10^{-3} torr (0.133 Pa) the conductance reaches practically the lower limit. Even at 10^{-2} torr (1.33 Pa) the conductance is quite low. For this reason, good mechanical pumps are adequate for the evacuation of powder insulation (fine mesh filters are needed to protect the pump from being damaged by the powder). When evacuated to 10^{-4} torr (1.33×10^{-2} Pa) or less, the apparent mean thermal conductivity for perlite [540°R (300 K) to 140°R (78 K)] ranges from 0.6 to 1.2 mW/m · K. For nonevacuated perlite filled with He or H_2 an apparent mean thermal conductivity of 115 mW/m · K can be used. With N_2 a typical value is 30 mW/m · K.

The principal function of the powder is to impede the transfer of heat by radiation. Powders are used when radiation would constitute an important heat leak. If an insulating vacuum is bounded by highly reflecting walls, such as clean Cu, Ag, or Al, adding powder may result in an increase of heat transfer because of the paths of solid conduction through the powder.

(ii) **Al powder or flakes added to a powder insulation** reduces the radiative transfer through the insulation. Powdered Al is more effective than flaked Al. The conductance of highly evacuated perlite insulation [$\Delta T = 540 - 140$°R $(300 - 78$ K)] may be reduced 40 percent by the addition of 25 percent of Al powder. A similar addition of Al powder to Santocel and Cab-O-Sil (diatomaceous earth) reduces the conductance by 70 percent. Metal powder additions are most effective with those powders that are partially transparent to infrared radiation of wave lengths longer than 3 μm.

3. **Rigid-Foam insulation** The foams most used in cryogenic applications are the more or less closed-cell foams. Table 19.2.10 gives thermal conductivities of selected samples of insulating foams. The conductivity of a foam is dependent on the conduction through the intracellular gas, on the transfer by thermal radiation, and on solid conduction. Temperatures below the condensing *T* of the encased gas condense it and improve the insulation. Gases encased when polymer foams are blown gradually diffuse out of the cells and are replaced by ambient gases. The conductivities of many Freon and CO_2 blown foams increase in time as much as 30 percent because of the diffusion of air into the foam. Conductivities may increase by a factor of 3 or 4 when cells are permeated with H_2 or He. Glass and silica foams appear to be the only ones having fully closed cells. The **structural rigidity** of foams may be used to eliminate the need for mechanical supports. It is possible to use rigid foams without inner liners and outer casings. Polymer foams have relatively high **thermal expansions,** even greater than the same polymer without voids. This makes cracking of the insulation a problem

Table 19.2.10 Apparent Mean Thermal Conductivities of Selected Foams, Balsa Wood, and Corkboard

Material	Density lbm/ft^3 (kg/m^3)	Thermal conductivity,* Btu/h · ft · °R (W/K · m)
Polystyrene foam†	2.4 (38)	0.019 (0.033)
	2.9 (46)	0.015 (0.026)
Epoxy resin foam†	5.0 (80)	0.019 (0.033)
Polyurethane foam†	5.0–8.7 (80–139)	0.019 (0.033)
Rubber foam†	5.0 (80)	0.021 (0.036)
Silica foam†	10.0 (160)	0.032 (0.055)
Glass foam†	8.7 (139)	0.020 (0.035)
Balsa wood, across grain	7.3 (117)	0.027 (0.047)
	8.8 (141)	0.032 (0.055)
	20.0 (320)	0.048 (0.083)
Corkboard, no added binder	5.4 (86)	0.021 (0.036)
	7.0 (112)	0.0225 (0.039)
	10.6 (170)	0.025 (0.043)
	14.0 (224)	0.028 (0.048)
Corkboard with asphalt binder	14.5 (232)	0.027 (0.047)

* Test space pressure = 14.7 lb/in^2 abs (0.101 MPa).

† ΔT = 540 to 140°R (300 to 78 K).

when it is bonded to metal. Sliding expansion joints have been used to overcome this. Mylar film bags (0.001 in or 0.025 mm thick) have been used to hold cryogenic liquids inside rigid-foam insulation and plastic films outside to keep the ambient atmosphere from the insulation.

4. **Balsa wood and corkboard** have been used for insulating cryogens that are not very cold, e.g., liquid methane (201°R or 112 K) and higher-boiling cryogens. The conductivities of these insulators (Table 19.2.10) are considerably higher than for the other types of low-*T* insulation discussed above. Their use, therefore, is restricted to special applications, as where their ability to support internal structures mechanically is important.

Conduction of heat along structural supports passing through the insulation merits special attention for an adequate realization of the benefits of superior insulation. Ideal materials of construction would have high mechanical strength and low thermal conductivity. In Table 19.2.11, the yield strengths, in tension, of construction materials are compared with their thermal conductivities. This kind of comparison favors the nonmetallic materials in Table 19.2.11. When large quantities of cryogens are handled in the field, metals are commonly used for structural members because glass and plastics (Teflon excepted) are brittle at low *T*'s.

Minimizing the transfer of heat through the supporting structure offers opportunities for ingenuity and inventiveness in design. The transfer is minimized by lengthening the supporting members. A stack of thin metal disks or washers, utilizing the contact resistance between adjacent disks, will increase thermal resistance without increase in length of support. A stack of 0.0008-in (0.02-mm) stainless steel disks under a compressive stress of 1,000 lb/in^2 (6.895 MPa), in vacuum, has the same thermal resistance as a solid rod 50 times the length of the stack.

Insulation Selection Because of their smaller heats of vaporization and greater cost of production, the lowest-*T* cryogens (LHe and LH_2) ordinarily merit more effective insulation than the higher-boiling cryogens like LO_2 and LN_2, and these in turn are ordinarily better insulated than other cryogens boiling at still more elevated *T*'s. The choice of insulation ordinarily involves the conditions of use of the equipment, as well as considerations of the tolerable loss of refrigeration and the cost of the insulation and its installation. If equipment is to have intermittent, rather than long, uninterrupted operating periods, the insulation heat capacity and time lag in reaching steady state are important. An insulation with large heat capacity (e.g., powders) will evaporate large quantities of refrigerant and take a long time to reach steady state. Powders are compacted by mechanical shocks and vibrations. Continued mechanical loading will compact both multilayer insulation and powders. Weight and bulkiness of the insulation can be important. The heat transported by the mechanical supports and vents connecting the cold interior with the warm exterior becomes the minimum refrigeration loss, attainable only with perfect insulation of the rest of the system. A large loss of refrigeration through supports and vents ordinarily reduces the attractiveness of costly insulation having the lowest conductance.

SAFETY

Precautions are needed for the safe handling and storage of cryogens because (1) air can contaminate the cryogen, creating a potential explosive, either directly as when air mixes with H_2 or CH_4 or indirectly by transforming an inert cryogen like LN_2 into an oxidant (a potential hazard for combustible insulation), and (2) moisture and air can freeze on cold surfaces, clogging vents and preventing normal operation of valves.

Vent systems should be sized for sufficient exhaust velocities to prevent back-diffusion of air into the cryogen space. Storage at slightly

Table 19.2.11 Comparison of Materials for Support Members

Material	E_y,* ¶ kpsi (MPa)	κ,* Btu/h · ft · °R (W/K · m)	E_y/κ kpsi · h · ft · °R/Btu (MN · K/W · m)
Aluminum 2024	55 (379)	47 (81)	1.17 (4.68)
Aluminum 7075	70 (483)	50 (87)	1.4 (5.55)
Copper, ann	12 (83)	274 (474)	0.044 (0.18)
Hastelloy† B	65 (448)	5.4 (9.3)	12 (48)
Hastelloy† C	48 (331)	5.9 (10.2)	8.1 (32)
K Monel‡	100 (690)	9.9 (17.1)	10.1 (40)
Stainless steel 304, ann	35 (241)	5.9 (10.2)	5.9 (24)
Stainless steel (drawn 210,000 lb/in^2)	150 (1034)	5.2 (9.0)	29 (115)
Titanium, pure	85 (586)	21 (36.3)	4.0 (16)
Titanium alloy (4A1–4Mn)	145 (1000)	3.5 (6.1)	41.4 (164)
Dacron§	20 (138)	0.088 (0.15)¶	227 (920)
Mylar§	10 (69)	0.088 (0.15)¶	113 (460)
Nylon§	20 (138)	0.18 (0.31)	111 (445)
Teflon§	2 (14)	0.14 (0.24)	14.3 (58)

* E_y is the yield stress, and κ is the average thermal conductivity between 36 and 540°R (20 and 300 K).

† Haynes Stellite Co.

‡ International Nickel Co.

§ E.I. du Pont de Nemours & Company

¶ Room-temperature value.

elevated pressure is preferable to prevent in-leakage of contaminants. Condensation can also be a problem on poorly insulated lines carrying low-boiling cryogens; frost is not a hazard but the oxygen-enriched condensed air is hazardous in the presence of combustibles. The insulation of liquid vent lines is a necessary precaution. Where this is not possible, appropriately placed guttering can prevent the contact of any condensed air with combustible objects.

Any space, either containing a cryogen or being refrigerated by a cryogen, should be protected with proper **safety-relief mechanisms,** i.e., relief valves or rupture disks. Such space includes less obvious ones such as the insulating vacuum space surrounding a cryogenic fluid or a section of line which can trap liquid between two valves. A heat leak to trapped liquid confined without a vent for escape of vapor may develop pressures sufficient to burst the containing vessel. Gas pressures necessary to maintain liquid density at ambient temperature are for helium, 18,000 lb/in^2 (124 MPa); for hydrogen, 28,000 lb/in^2 (193 MPa); and for nitrogen, 43,000 lb/in^2 (296 MPa).

The **selection of structural** materials calls for consideration of the effects of low temperatures on the properties of those materials. A number of otherwise suitable materials become brittle at low temperature (Fig. 19.2.10). Materials must perform satisfactorily over the complete range from cryogenic to room temperature, with hydrogen embrittlement particularly significant. While hydrogen embrittlement is not believed to be a problem at liquid-hydrogen temperatures, it has been shown to occur at room temperature. The 300-series stainless steels, aluminum alloys and BeCu, among other alloys, are more resistant to hydrogen embrittlement.

In calculating allowable stresses, room temperature yield and tensile strengths should be used because (1) pressure testing of containers at ambient temperature is frequently necessary, and (2) in the case of large vessels, large unknown temperature gradients can exist above the liquid level both in the ullage space and in the walls of the vessel. The 300-series stainless steels are normally austenitic (f.c.c.). However, in some 300-series stainless steels austenite is partially transformed by cold-working and possibly by temperature cycling to martensite. As martensite is brittle at low T's, the ductility of these steels is reduced by this kind of treatment. For this reason, it is recommended as an added safety factor that room temperature strength of the 300 series stainless steels be used for the design of structural members.

Careful **stress analysis** is essential and must include stresses due to (1) thermal contraction of equipment, and (2) radial, axial, and circumferential temperature gradients caused by non-uniform cooling rates. The latter are frequently encountered in the cool-down of cryogenic transfer lines.

Cryogenic equipment should be **purged** before being placed in service to remove unwanted condensables and gases that could form explosive mixtures. Commonly used methods are (1) evacuating and back-filling with purge gas and (2) flowing of purge gas through the system.

The cryogen can be a direct **hazard to personnel**. Cryogen ''burns'' can result from direct contact with either the cryogen or uninsulated equipment containing the cryogen. The large evolution of gases associated with cryogenic spills can result in asphyxiation. Asphyxiation is a hazard in entering a warmed cryogenic vessel. Further safety details are found in the references.

19.3 OPTICS

by Michael J. Clark

REFERENCES: AIP Handbook, McGraw-Hill. Born and Wolf, ''Principles of Optics,'' Pergamon Press. Hecht and Zajak, ''Optics,'' Addison-Wesley. Jenkins and White, ''Fundamentals of Optics,'' McGraw-Hill. Kingslake, ''Lens Design Fundamentals,'' Academic Press. Malacara, ''Optical Shop Testing,'' Wiley. Photonics Handbook, Laurin Publishing Co.

FUNDAMENTAL THEORIES OF LIGHT

Electromagnetic radiation spans a very large spectrum (Fig. 19.3.1). **Visible light** is a small band in the electromagnetic spectrum with wavelengths ranging between 0.390 and 0.770 μm where approximately 0.560 μm (green-yellow) seems the brightest to the average person. Units of measurement for electromagnetic radiation are wavelengths of light. Wavelength is the distance between peaks of a light wave. **Wavelength** λ has a reciprocal relationship to frequency of vibration of a wave, f, and for a traveling light wave that relationship is $\lambda = c/f$, where c is the speed of light. The **speed of light** in a vacuum for electromagnetic waves of all wavelengths is taken as 2.99793×10^8 m/s (for calculations made to four significant digits or less it is practical to use the estimation of 3.0×10^8 m/s for the speed of light). The speed of light is reduced when the light is traveling in any medium. The ratio of the speed of light in a vacuum c to the speed of light in a medium c' is defined as the **index of refraction,** $n = c/c'$. The index of refraction of air at standard conditions (0°C and 760 mmHg) is taken as 1.00292; it is often assumed to be unity. Most optical substances; gases, liquids, and solids (glasses, crystals, etc.) have refractive indexes between 1 and 4 (see AIP Handbook or Photonics Handbook). The index of refraction of most substances (including optical glass) is not constant but varies with the color of light. The **dispersion** ν of glass is normally taken as $\nu = (n_D - 1)/(n_F - n_C)$, where n_C, n_D, and n_F are the indexes of refraction for three spectral lines at 0.6563 μm (red), 0.5893 μm (yellow), and 0.4861 μm (blue), respectively. These lines are obtained from hydrogen (n_C and n_F) and sodium (n_D) discharge tube light sources. Most optical glasses have dispersions between 20 and 65.

FUNDAMENTALS OF GEOMETRICAL OPTICS

At an interface of two media with different refractive indexes, n and n' (Fig. 19.3.2), light rays at an incident angle I measured from the direction normal to the interface will be partially reflected back into the incident medium and partially refracted into the second medium. Two fundamental laws of physics describe these occurrences: The **law of reflection** states that the angle of incidence I equals the angle of reflection I'', and the angle of reflection is on the opposite side of the normal in the incident medium. The **law of refraction** (also referred to as **Snell's law**) states that the relationship between the incident angle and the refracted angle I' obeys $n \sin I = n' \sin I$. Given a spherical interface on an optical axis (Fig. 19.3.3), the relationship between object distance l, image distance l', and radius of curvature of the spherical interface r, is given by $n'/l' = (n' - n)/r + n/l$. This relationship holds only for rays that are close to the optical axis (paraxial rays).

The **magnification** m for such a surface is $m = h'/h = nl'/n'l$. If the image is inverted with respect to its object, the magnification is a negative number. Fractional magnifications indicate an actual reduction in size. The **focal length** f of a thin lens is the distance from the lens when the object is infinitely far away. The **power** Φ of a lens is taken as $\Phi = n/f$ (see Jenkins and White). The focal lengths of typical commercial lenses for use in air vary between 3 mm for microscope objectives to 30.5 m (100 ft) for large observatory telescope objectives.

The larger the **aperture** of an optical element, the brighter will be the image and the better the resolving power, if the rest of the system (including the eye) is capable of accommodating all of the light. The aperture of telescope objectives and camera projection lenses is usually

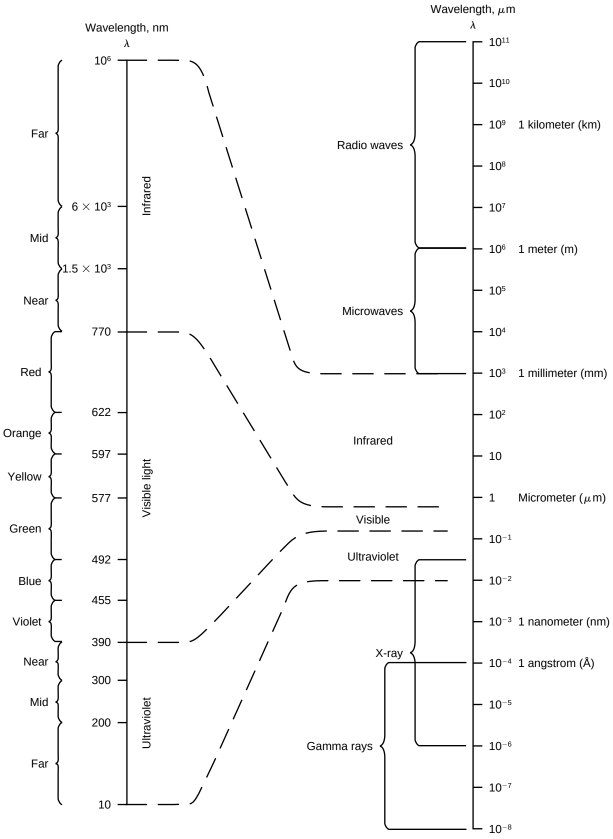

Fig. 19.3.1 Electromagnetic spectrum.

described by the **f number,** which is defined as the ratio of the focal length to the clear diameter. Commercially available lenses usually fall between *f*/1.2 and *f*/15. The aperture of microscope objectives is usually described by the **numerical aperture** NA. The numerical aperture is defined as *n* sin *U*, where *n* is the index of refraction of the object space and *U* is the half angle of the vertex of the cone of light from the object that passes through the lens. Commercially available lenses usually fall between NA/1.40 and NA/0.08.

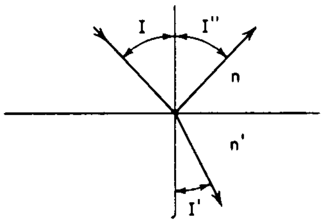

Fig. 19.3.2 Reflection and refraction at a plane surface.

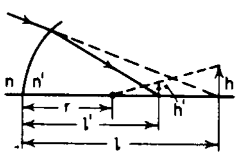

Fig. 19.3.3 Refraction at a spherical surface; all parameters are shown in their positive sense.

Photographic film speeds are usually given as an ASA exposure index. Black-and-white films are available commercially in the ASA range from 25 to 3,000. Color films fall in the slower end of this range. A rule of thumb for exposing films on a bright, sunny day is to set the aperture of the camera at *f*/16 and the shutter speed at 1/ASA exposure index.

OPTICAL DESIGN

The first step in developing a cost-effective optical system is to investigate what components are commercially available that would meet your specific needs. If commercially available components do not sufficiently meet the requirements of the application, then a customized design would need to be developed. The majority of optical designs are performed through the use of computer-aided design (CAD) packages. Originally, early in the 1970s, the optical CAD market was dominated by mainframe packages. Due to recent advances in computer hardware technology, the availability of PC-based optical design packages has expanded (see Photonics Handbook for more information on optical design packages). The optical design package is used as a tool by designers to develop an optical system. The designer generally enters initial design parameters and specifications. The CAD software then optimizes the design parameters and outputs a design proposal. Several iterations may be required to develop an appropriate system. When developing an optical design consideration should be made for *f* number, clear aperture, field of view, spectral range, packaging constraints, environmental parameters, transmittance, and performance specification.

LASERS

The name **laser** is an acronym of light amplification by the stimulated emission of radiation. A laser is a device that generates a highly coherent monochromatic source of intense radiation. The laser contains a cavity that is excited to stimulate emissions of photons that are repeatedly reflected between highly reflective mirrors that are precisely aligned at either end of the cavity. The intensity of the emission increases as it oscillates within the cavity. One mirror is made to partially transmit the intense monochromatic source. The wavelength of the source is determined by the type of material used to make up the cavity. The cavity may be made up of any material, gas, liquid, crystal or glass, whose atoms are capable of being excited from the ground state to a semistable state. Some typical laser applications are bar code scanning, optical discs, optical fiber communication, photolithography, ablation, micromachining, microsurgery, fusion, spectroscopy, interferometry, and distance measurement, to name a few.

INTERFERENCE AND INTERFEROMETRY

Optical **interference** is the combination of two or more waves of light. Constructive interference occurs when the peaks and valleys of the waves align with each other; these waves are said to be *in phase.* If the waves are out of phase they will destructively interfere. For wavefronts with equal amplitudes and wavelengths, and the appropriate phase relationship, destructive interference will cause the waves to cancel each other and therefore create dark bands called **fringes**. Optical surfaces, plano or spherical, may be measured through the relative interference of the wavefront of a test surface with the wavefront of a known highly precise reference surface. **Test glasses** and/or **test/plates** are frequently used as a reference surface in measuring an optical surface. The test glass is brought into contact with the surface under test and placed under a monochromatic source of light so that interference fringes become visible. The interference fringes are then evaluated to determine the accuracy of the surface under test relative to the reference surface (see Photonics Handbook for additional information on fringe interpretation). An **interferometer** is a measurement instrument that utilizes interference theory to determine the accuracy of optical surfaces and/or systems. Interferometers typically generate a precise monochromatic wavefront that is split to generate a test wavefront and reference wavefront. These wavefronts are passed through the test and reference optical systems and then recombined to create interference fringes. There are several different configurations of interferometers with variable applications (see Malacara). Some of the most common configurations are the Fizeau interferometer, Michelson interferometer, and the Twyman-Green interferometer. Many of the commercially available interferometers have incorporated sophisticated computer systems to capture surface data and enhance the analysis of optical surfaces and systems.

Defects of an optical surface or system that cause deviations in the image accuracy are called **aberrations**. There are several primary geometrical aberrations that are evaluated to develop optimum performance of an optical design. **Spherical aberration** is an axial error where rays across the aperture have different focal lengths. **Astigmatism** occurs when light in the tangential plane focuses to a different point than light in the sagittal plane. In the absence of astigmatism, the image is formed on a curved surface referred to as the Petzval surface. The curvature of the Petzval image, or **field curvature,** is 1/(radius of the Petzval surface). **Coma** occurs when rays through the outer portion of a lens produce a higher or lower magnification than the central rays. **Distortion** is a change of magnification as a function of the field of view. There are two **chromatic aberrations:** axial and lateral. Axial chromatic aberration is the effect of different wavelengths of light focusing to a different point on the optical axis due to the effects of dispersion. Lateral color is a change or difference in the off-axis image focus for different wavelengths of light.

FIBER OPTICS

Optical fibers consist of an optically transparent central fiber, surrounded by a less dense cladding. Bundles of such fibers may be used to transmit images over long distances and/or around curves. Optical fibers utilize the principle of total internal reflection where the incident angle is greater than or equal to the critical angle. The **critical angle** Φ_C can be computed by the expression $\sin \Phi_C = n/n'$, where n is the refractive index of the fiber and n' is the refractive index of the cladding surrounding the fiber. To achieve total internal reflection, the index of the fiber material must be greater than the index of the cladding. Optical fibers are very effective in communication applications—video, voice, and data—with expanding applications in industrial facilities and military systems. Significant advantages to optical fibers over electrical wires are electrical isolation, low signal degradation, broad signal bandwidth, high reliability, light weight, and low cost.

19.4 MISCELLANEOUS

(Staff Contribution)

SIZES OF TYPE

The unit of height of a line of printer's type is the ''point.'' One point = $\frac{1}{72}$ in. Six-point type is consequently $\frac{6}{72} = \frac{1}{12}$ in high. The sizes generally employed in books and periodicals are 6-, 7-, 8-, 9-, 10-, 11-, and 12-point. These are shown in Fig. 19.4.1.

6-point
7-point
8-point
9-point
10-point
11-point
12-point

Fig. 19.4.1 Examples of type sizes.

COPY PREPARATION AND COPY FITTING

It is recommended that manuscript material be printed double-spaced on 8½- by 11-in paper with 1½-in margins at the top and left side, 1-in margins on the bottom and left side, and with each page containing 25 lines, double-spaced. Thus a printed line will be 60 characters long. Nearly all printers can print at 10 characters to the inch and double-spaced (three lines to the inch). For convenience in copyediting, printed copy should be printed double-spaced. One manuscript page so printed is equivalent to about 250 words (assuming that each word averages six characters).

One way to determine approximately how many lines of manuscript copy are equivalent to how many lines of finished book type is to print, to the specifications given above, a number of lines from a book which has been printed in the desired type font. This will show the relationship between the number of manuscript-copy lines and lines set in type.

Nearly all word-processing programs will produce satisfactory manuscript. Authors working directly with a publisher will usually make arrangements to submit their manuscripts in computer-readable form. Some publishers require it. Publishers have very comprehensive format-conversion facilities. Authors working in groups or with intermediate editors, however, face the problem that others of their associates or editors may not have much conversion capability. In such cases, the editors or group leaders will make arrangements to accept such computer-readable copy as they can. In all cases, a printed copy should accompany all computer-readable material.

PROOFREADING

When correcting proof, use ink or pencil of a different color from the marks already on the proof. Put your marks in the right or left margin, whichever is nearer to the word corrected. If there are several corrections in a single line, place them in order from left to right, separated by a slant line (e.g., tr/cap/,). If the same correction is made several times in the same line with no intervening correction, make your correction once in the margin, followed by an appropriate number of slant lines. For instance, if you add an *s* to three words in a line, write in the margin s///. When you wish to insert words, put a caret at the point of insertion and write the additional words in the margin. To delete material without substituting anything, cross it out and put a delete sign in the margin. When you delete words and substitute other words, cross out the unwanted material, use a caret within the line, and write the new material in the margin; the delete sign is then unnecessary. Standard proofreader's marks are shown in Fig. 19.4.2.

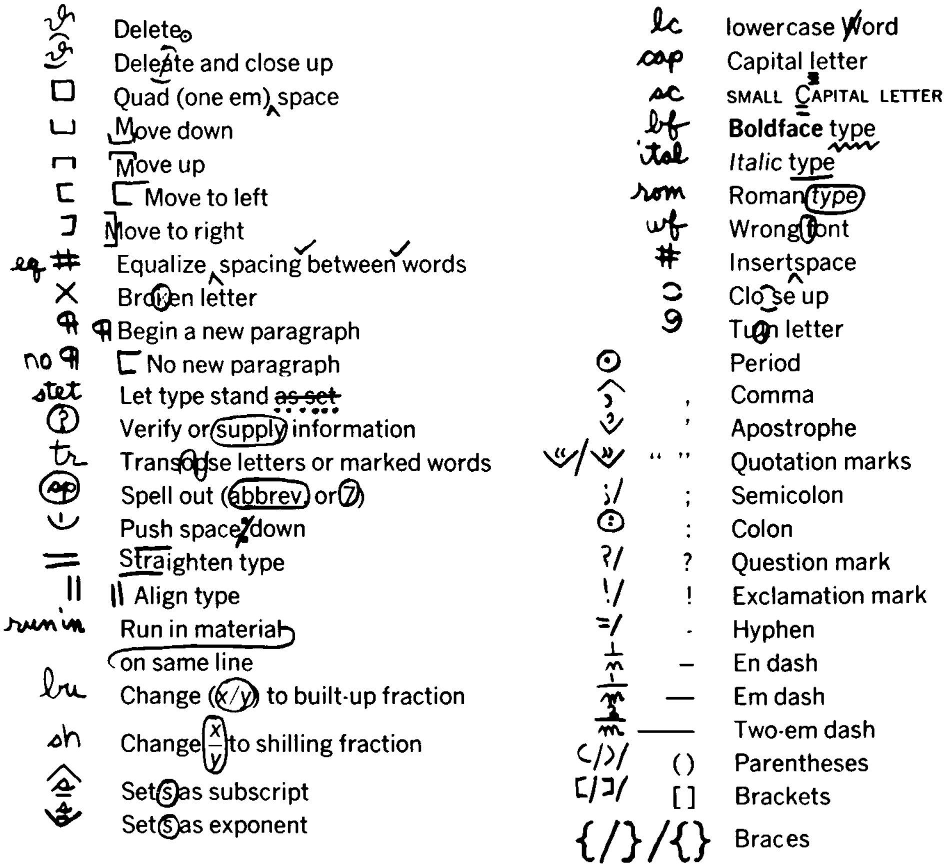

Fig. 19.4.2 Proofreader's marks.

Occasionally you may decide that material you have deleted should be restored. Place a row of dots under the crossed-out material, cross off the delete sign in the margin, and write ‘‘stet’’ (let it stand).

SPECIAL ALPHABETS

The primary alphabets used as symbols are Greek, script, and German. Examples of these are shown in Figs. 19.4.3 through 19.4.6.

Α Β Γ Δ Ε Ζ Η Θ Ι Κ Λ Μ Ν Ξ Ο Π Ρ Σ Τ Υ ϕ Χ Ψ Ω
α β γ δ ϵ ζ η θ ι κ λ μ ν ξ ο π ρ σ τ υ ϕ χ ψ ω

Fig. 19.4.3 Greek alphabet.

Α Β Γ Δ Ε Ζ Η Θ Ι Κ Λ Μ Ν Ξ Ο Π Ρ Σ Τ Υ Φ Χ Ψ Ω
α β γ δ ϵ ζ η θ ι κ λ μ ν ξ ο π ρ σ τ υ ϕ χ ψ ω

Fig. 19.4.4 Boldface Greek alphabet.

Note that the capital Greek letters used by the printer are easily mistaken for the Roman letters A, B, E, Z, H, I, K, M, N, O, P, T, and X and may not be suitable as symbols. If they are used in a manuscript, they should be identified with a marginal note so they will not be improperly set. The form of many Greek boldface symbols is markedly different from the style of the lightface characters.

Lightface: 𝒜 ℬ 𝒞 𝒟 ℰ ℱ 𝒢 ℋ ℐ 𝒥 𝒦 ℒ
ℳ 𝒩 𝒪 𝒫 𝒬 ℛ 𝒮 𝒯 𝒰 𝒱 𝒲 𝒳 𝒴 𝒵
Boldface: 𝓑 𝓔 𝓔 𝓕 𝓗 𝓘 𝓜

Fig. 19.4.5 Script alphabet.

𝔄 𝔅 ℭ 𝔇 𝔈 𝔉 𝔊 ℌ ℑ 𝔍 𝔎 𝔏 𝔐 𝔑 𝔒 𝔓 𝔔 ℜ 𝔖 𝔗 𝔘 𝔙 𝔚 𝔛 𝔜 ℨ
𝔞 𝔟 𝔠 𝔡 𝔢 𝔣 𝔤 𝔥 𝔦 𝔧 𝔨 𝔩 𝔪 𝔫 𝔬 𝔭 𝔮 𝔯 ſ 𝔱 𝔲 𝔳 𝔴 𝔵 𝔶 𝔷
ä ü ö 𝔄̈ 𝔘̈ 𝔒̈ ch ck ß ſt tz

Fig. 19.4.6 German Fraktur alphabet.

Script and German symbols (Figs. 19.4.5 and 19.4.6) may be indicated by circling in a distinctive color the corresponding English letter.